PIPING HANDBOOK

BY

SABIN CROCKER, M. E.

Senior Engineer, Engineering Division, The Detroit Edison Company; Member, American Society of Mechanical Engineers, American Society for Testing Materials, A. S. A. Sectional Committee on Standardization of Pipe Flanges and Fittings, A. S. A. Sectional Committee on Standardization of Wrought Iron and Wrought Steel Pipe and Tubing, A. S. A. Sectional Committee on Code for Pressure Piping and Chairman of Its Subcommittee on Scope and Intent

FOURTH EDITION
SECOND IMPRESSION

McGRAW-HILL BOOK COMPANY, INC.
NEW YORK AND LONDON
1945

THE MAPLE PRESS COMPANY, YORK, PA.

PREFACE

In prefacing the First Edition in 1930 it was pointed out that the purpose of this handbook is to provide authoritative and accessible data for the engineer interested in piping design. The author and publisher sensed that the increasing importance of piping systems in modern power plants, in distribution systems, and in industrial operations, together with the specialized nature of the problems involved, justified a handbook devoted exclusively to the subject. That such a need existed has been attested by the interest accorded the first three editions.

Many sources have been drawn upon freely with the idea of assembling and coordinating all appropriate material. Due credit to the original sources of extracted material has been given throughout the book. Owing to the increasing number of standards, specifications, and codes for materials and construction published on all phases of piping, it is growing more difficult to digest all of this wealth of information so that it will fit between the covers of one handbook. In some cases, therefore, it has been necessary to stop with citing published references on subjects too extensive or too involved for abstracting, such, for instance, as the Hardy Cross method for computing flow in distribution networks. On the other hand, the need has been seen for making this handbook *self-contained*, insofar as practicable, for the benefit of those who have to deal with design problems without ready access to a reference library.

In the Fourth Edition, the scope of Piping Handbook has been extended to include chapters on "Gas Piping," "Refrigeration Piping," "Hydraulic Power Transmission Piping," and "Corrosion," which also covers the subject of protective coatings. In addition to this the chapter on "Water-supply Piping" has been considerably augmented, and supplementary material of interest to hydraulic engineers has been added in the section on "Flow of Water in Pipes," which now includes the Scobey, Williams-Hazen, and Kutter-Manning formulas. In making these extensions, the author has been assisted and counseled in the respective fields by

the outstanding experts whose names are mentioned in the List of Collaborators.

Growing interest in the "rational formula" or Reynolds number type of solution for fluid-flow problems (see pages 107 to 137) has led to an elaboration of this subject and to providing a greater variety of rational formulas in which all constants have been adjusted to suit using a common friction factor f which can be determined from Fig. 15*a* or *b*. In order to obviate the necessity for making cut-and-try solutions in solving certain problems by this method, a new system for making direct rational solutions in such cases has been provided in Fig. 15*b* and Table XV. A special effort has been made to define on pages 82 to 84 all the symbols used in the flow formulas of Chap. II and to follow this code uniformly throughout the chapter. This has involved some changes in terminology from the earlier editions.

The area-moment or grapho-analytical method for solving piping flexibility problems (see pages 783 to 816), which was introduced with the First and Second Editions, has found extensive use and is generally recognized as giving as refined a solution as is worth while attempting in practical applications. Although a solution by this method can be made as rigorously exact as circumstances warrant, it is advisable to ignore certain minor rotations in any method of solution in order to keep the work within reasonable bounds. Accordingly, the original method has been continued unchanged, but simplified solution charts for certain frequently encountered shapes of bends have been added from time to time.

All existing codes, dimensional standards, and materials specifications for piping abstracted in the earlier editions have been revised to date and abstracts added for those formulated since. Abstracts and references embrace standards and codes of the American Standards Association, the American Society of Mechanical Engineers, the American Society for Testing Materials, the American Petroleum Institute, the American Gas Association, the American Water Works Association, the National Board of Fire Underwriters, the American Society of Heating and Ventilating Engineers, and others. For the convenience of those desiring further information, it is indicated where unabridged versions of the latest drafts of such standards may be purchased.

In addition to changes required to keep the handbook abreast of technical developments, a continual effort has been made to increase its usefulness through improving tables and charts, citing

additional authorities, providing more cross references, and augmenting the index. Any suggestions toward accomplishing these ends will be welcomed, as will criticism of the material presented and information regarding omissions or typographical errors.

Owing to the preoccupation of J. H. Walker with other affairs, he was unable to participate fully in preparing this Fourth Edition and his name no longer appears as co-author. Fortunately Mr. Walker was able to take part in revamping the chapters on "Heat Insulation," "Building Heating Systems," and "Underground Steam Piping," in which fields he is an expert of long standing. In recognition of this contribution, the authorship of these chapters has been credited to him.

The chapter on "Water-supply Piping," originally contributed by George H. Fenkell, has been expanded under his direction and continues with his name as author. Mr. Fenkell is a well-known authority on water-works practice, having been General Manager and Chief Engineer of the Department of Water Supply, City of Detroit, Michigan, until his retirement in 1938.

Much of the credit for improvements in the Fourth Edition is due the author's associates Max W. Benjamin, Arthur McCutchan, and Harvey A. Wagner who relieved him of a great deal of the burden through taking responsibility for assignments in fields where they specialize. To all those whose names appear in the accompanying list of collaborators or elsewhere in the book, grateful acknowledgment is made for having contributed helpful suggestions, specialized information, or copy for illustrations.

SABIN CROCKER

DETROIT, MICH.,
April, 1945.

LIST OF COLLABORATORS

R. P. Anderson, Secretary, Division of Refining, American Petroleum Institute

Russell E. Barnard, Chairman Technical Standards Committee, Steel Pipe Manufacturers Technical Advisory Committee

John W. Batten, President, Michigan Consolidated Gas Company

Max W. Benjamin, Engineering Division, The Detroit Edison Company

Joseph A. Blecki, Piping Engineer, The Detroit Edison Company

C. M. Burnam, Jr., Editor, *Heating, Piping and Air Conditioning*

Leslie D. Burritt, Chairman Subcommittee on Oil Piping, American Standard Code for Pressure Piping, ASA B31

H. M. Chadwick, Welded Pipe Division, California Corrugated Pipe Company

Frank J. Egan, Vice President, James B. Clow and Sons

George H. Fenkell, retired, formerly Chief Engineer and General Manager, Department of Water Supply, City of Detroit

Carl H. Flink, Technical Secretary, American Society Heating and Ventilating Engineers

Herbert A. Freeman, Underground Lines Department, The Detroit Edison Company

John Gaillard, Mechanical Engineer, American Standards Association

Joseph C. Groff, Development Engineer, The Aldrich Pump Company

Arthur Gould, Piping Designer, The Detroit Edison Company

John J. Harman, Secretary, Manufacturers Standardization Society of the Valve and Fittings Industry

F. Theodore Harrington, Vice President and Sales Manager, Vickers, Inc.

Charles A. Hill, Streamline (Solder) Fittings Division, Mueller Brass Company

J. E. Holveck, Hydraulic Systems, Worthington Pump and Machinery Corporation

LIST OF COLLABORATORS

WILLIAM W. HURLBUT, Assistant Chief Engineer and General Manager, Bureau of Water Works and Supply, City of Los Angeles, Calif.

HARRY E. JORDAN, Secretary, American Water Works Association

JEROME J. KANTER, Materials Research Engineer, Crane Company

WATSON F. KINNEY, Research Department, The Detroit Edison Company

CLIFFORD B. LEPAGE, Assistant Secretary—Standards, American Society of Mechanical Engineers

FRANCIS F. LONGLEY, Vice President, Lock Joint Pipe Company

A. J. MAAHS, Chief Engineer, Transite Pipe Division, Johns-Manville Corporation

ARTHUR MCCUTCHAN, Engineering Division, The Detroit Edison Company

ROBERT J. PAINTER, Assistant to the Secretary, American Society for Testing Materials

A. L. PARKER, President, The Parker Appliance Company

CHARLES M. PARKER, Secretary, General Technical Committee, American Iron and Steel Institute

LAWRENCE K. RICHEY, Superintendent of Distribution, Michigan Consolidated Gas Company

C. GEORGE SEGELER, Engineer of Utilization, American Gas Association

ARTHUR M. SELVEY, Chief Engineer, Shell Chemical Division, Shell Union Oil Corporation

J. H. SHEEHAN, Cone Valve Division, Chapman Valve Manufacturing Company

JULIAN E. STARK, Manager, Engineering and Research Division, Crane Company

A. B. STICKNEY, Chairman, Subcommittee on Refrigeration Piping, American Standard Code for Pressure Piping, ASA B31

J. HALL TAYLOR, President, Taylor Forge and Pipe Works

HARVEY A. WAGNER, Engineering Division, The Detroit Edison Company

J. HERBERT WALKER, Superintendent of Central Heating, The Detroit Edison Company

EDWIN A. WERT, Vice President, Power Piping Division, Blaw Knox Company

THOMAS F. WOLFE, Engineer, Cast Iron Pipe Research Association

C. A. YOUNG, Secretary, Division of Production, American Petroleum Institute

CONTENTS

LIST OF ABBREVIATIONS[1]

abs	Absolute
AGA	American Gas Association
AISI	American Iron and Steel Institute
Amer Std	American Standard
API	American Petroleum Institute
ASA	American Standards Association
ASHVE	American Society of Heating and Ventil Engineers
ASME	American Society of Mechanical Engineers
ASTM	American Society for Testing Materials
avg	Average
AWWA	American Water Works Association
B or Bé	Baumé
B & S	Bell and spigot *or* Brown & Sharpe (gauge)
bbl	Barrel
Btu	British thermal unit (s)
C	Centigrade
Cat.	Catalogue
cfm	Cubic feet per minute
cfs	Cubic feet per second
C.I.	Cast iron
C.S.	Cast steel (not recommended for abbreviation)
Comp	Companion
cu ft	Cubic feet
cu in.	Cubic inch(es)
C to F	Center to face
deg *or* °	Degree(s)
C	Degrees Centigrade
F	Degrees Fahrenheit
diam	Diameter
dwg	Drawing
ex.-hv.	Extra-heavy

[1] Abbreviations conform to the practice of the American Standard Abbreviations for Scientific and Engineering Terms, ASA Z10.1. In the interests of economy, this standard recommends that the period following abbreviations be omitted except where such omission results in an English word.

F. & D.	Faced and drilled
F	Fahrenheit
F. to F.	Face to face
Fig.	Figure
flg	Flange or flanges
flgd	Flanged
g	Gage *or* gauge
gal	Gallon
galv	Galvanized
gal per min	Gallons per minute, also gpm
hex	Hexagonal
hr	Hour
I.B.B.M.	Iron body bronze (or brass) mounted
I.D.	Inside diameter
kw	Kilowatt(s)
lb	Pound(s)
M.I.	Malleable iron
max	Maximum
min	Minimum
Mfr	Manufacturer
mtd	Mounted
MSS	Manufacturers Standardization Society (of Valve and Fittings Industry)
NEWWA	New England Water Works Association
N.P.S.	Nominal pipe size (formerly I.P.S. for iron pipe size)
O.D.	Outside diameter
O.S. & Y.	Outside screw and yoke
psi	Pounds per square inch
red.	Reducing
sched	Schedule
scd	Screwed
sec	Second
S.F.	Semifinished
Spec	Specification
sq	Square
SSP	Steam service pressure
SSU	Seconds Saybolt Universal
Std	Standard
Trans	Transactions
wt	Weight
WWP	Working water pressure

GREEK ALPHABET

Α, α	Alpha	Ν. ν.	Nu
Β, β	Beta		
Γ, γ	Gamma	Ο, ο	Omicron
Δ, δ	Delta	Π, π	Pi
Ε, ε	Epsilon	Ρ, ρ	Rho
Ζ, ζ	Zeta	Σ, σ, ς	Sigma
Η, η	Eta	Τ, τ	Tau
Θ, θ	Theta	Υ, τ	Upsilon
Ι, ι	Iota		
Κ, κ	Kappa	Χ, χ.	Chi
Λ, λ	Lambda		Psi
Μ, μ	Mu	Ω, ω	Omega

PIPING HANDBOOK

CHAPTER I

DEFINITIONS, FORMULAS, AND TABLES

LENGTHS, AREAS, SURFACES, AND VOLUMES

LIST OF SYMBOLS

A = angle in degrees.[1]
C = length of chord.
d = diameter of circle or sphere = $2r$.
h = height of segment, altitude of cone, etc., as explained in context.
π = ratio of circumference to diameter of circle = 3.1416.
Θ = angle in radian measure.[1]
S = length of arc, slant height, etc., as explained in context.
r = radius of circle or sphere = $d/2$.
R = mean radius of curvature for pipe bends.

Areas are expressed in square units, and volumes in cubical units, of the same system in which lengths are measured.

Triangle.—Area = one-half base × altitude.

Circle.—(Fig. 1):

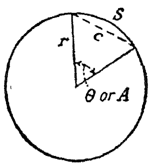


FIG. 1.—Length of arc and chord.

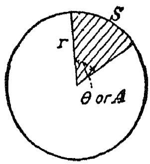


FIG. 2.—Area of sector.

Circumference = $\pi d = 2\pi r$. Area = $\pi r^2 = \pi d^2/4$.

Length of arc, $S = \Theta r = 0.0175Ar$.

Length of chord, $C = 2r \sin \Theta/2 = 2r \sin A/2$.

[1] Degrees can be converted to radian measure by multiplying by 0.0175, since 2π radians = 360 deg. Hence, $\Theta = 0.0175A$.

Area of Sector.—(Fig. 2):

Area $= \frac{1}{2}rS = \frac{1}{2}{}^2\Theta = \pi r^2 A/360 = 0.008727r^2A.$

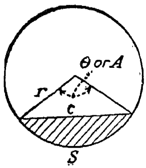


FIG. 3.—Area of segment, method I.

Area of Segment (Method I, Fig. 3).—Find area of sector having same arc, and area of triangle formed by chord and radii of sector. Area of segment equals sum of these two areas if segment is greater than a semicircle, and it equals their difference if segment is less than a semicircle.

$$\text{Area} = \tfrac{1}{2}r^2(\Theta \pm \sin\Theta)$$
$$= \tfrac{1}{2}r^2(0.0175A + \sin A).$$

Area of Segment [Method II (approximate)]:[1]

When $h = 0$ to $\frac{1}{4}d$, area $\quad h\sqrt{1.766dh - h^2}$.

When $h = \frac{1}{4}d$ to $\frac{1}{2}d$, area $= h\sqrt{0.017d^2 + 1.7dh - h^2}$.

When $h =$! to d, subtract area of empty sector from area of entire circle.

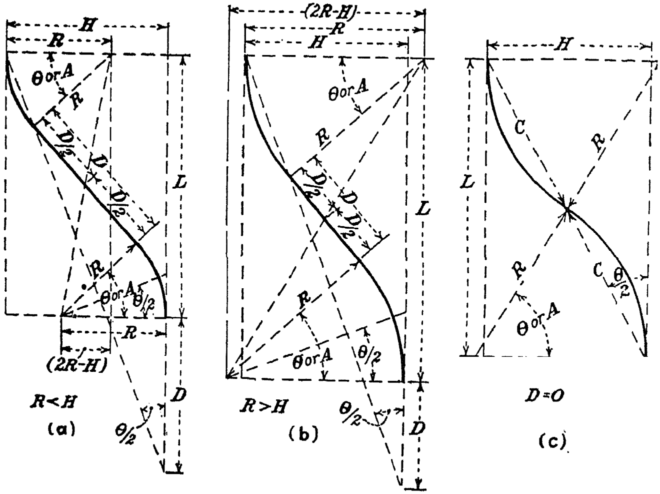


FIG. 4.—Offset bends.

Offset Bends (Fig. 4) (of interest in connection with pipe offset 7 shown in Fig. 8 on page 774).—The relation of D, R, H, and L is determined by geometry for the general case shown in Figs. 4a

[1] For sketch and table of volumes in partly full horizontal tanks, see Table V on pp. 24–26. The greatest error possible by this method is 0.23 per cent.

and b as follows: Consider the diagonal line joining the centers of curvature of the two arcs in either figure as forming the hypotenuse of three right-angled triangles and write an equation between the squares of the two other sides, thus,

$$2\sqrt{(D/2)^2 + R^2} = \sqrt{(2R - H)^2 + L^2}$$

squaring both sides and solving for each term in turn,

$$D = \sqrt{H^2 - 4HR + L^2}$$
$$L = \sqrt{D^2 + 4HR - H^2}$$
$$H = 2R - \sqrt{4R^2 - L^2 + D^2}$$
$$R = \frac{L^2 + H^2 - D^2}{4H}$$

$\theta = 2 \tan^{-1}\left(\frac{H}{L + D}\right)$ (from similarity of triangles, see Fig. 4a).

When $D = 0$ (Fig. 4c):

$$L = \sqrt{(4R - H)H}$$
$$R = \frac{L^2 + H^2}{4H}$$
$$C = \sqrt{RH}.$$

Length of pipe in offset:

$$\begin{aligned} L &= 2R\theta + D \\ &= 0.035RA + D \\ &= 4R \tan^{-1}\left(\frac{H}{L + D}\right) + D \end{aligned}$$ (where the angle is expressed in radians).

Cylinder:

Area $= 2\pi rh + 2\pi r^2 = 2\pi r(h + r)$.
r = radius of base. h = height.
Volume $= \pi r^2 h$.

Pyramid.—Right pyramid (*i.e.*, vertex directly above center of base):

Lateral area = one-half slant height × perimeter of base.
Volume = one-third altitude × area of base.

Cone:

Volume = one-third area of base × perpendicular distance from vertex to plane of base.

Right circular cone:

Lateral area $= \pi rs$.

Volume $= \frac{1}{3}\pi r^2 h$.

where s = slant height.

r = radius of base.

h = perpendicular distance from vertex to plane of base.

Frustum of right circular cone:

Lateral area $= \pi s(r + r')s = \sqrt{(r - r')^2 + h^2}$

where r = radius of lower base.

r' = radius of upper base.

h = height of frustum.

s = slant height of frustum.

Volume $= \frac{1}{3}\pi h(r^2 + rr' + r'^2)$.

Sphere:

Area $= 4\pi r^2 = \pi d^2$.

Volume $= \frac{4}{3}\pi r^3 = \frac{1}{6}\pi d^3$.

Lift of a Valve.—The lift of a valve to give full opening should be one-fourth the diameter (of seat). *Example:* If the diameter of valve seat is 6 in., then the lift should be $1\frac{1}{2}$ in.

Proof.—Area of 6-in.-diameter port $= \pi 6^2/4 = 28.27$ sq in. Area of ring opened by $1\frac{1}{2}$-in. lift $= \pi 6 \times 1\frac{1}{2} = 28.27$ sq. in.

Number of Holes of Small Diameter Equal in Area to One Hole of Large Diameter.—RULE: Divide the large diameter by the small diameter and square the quotient. *Example:* When making a screen out of a 6-in. diameter pipe, it is desired to know the number of $\frac{1}{4}$-in. holes equal in area to the cross section of the 6-in. pipe. *Solution:* $6 \div \frac{1}{4} = 24$. $24 \times 24 = 576$ $\frac{1}{4}$-in. holes equal the area of a 6-in. pipe. *Proof:*

Area of a 6-in. pipe $= \pi 6^2/4 = 28.27$ sq in.

Area of 576 $\frac{1}{4}$-in. holes $= [\pi (\frac{1}{4})^2/4] \times 576 = 28.27$ sq in.

CENTER OF GRAVITY

General.—The center of gravity of a body or of a system of bodies connected rigidly is that point about which there would be no tendency for the body to rotate if it were used as the point of suspension. In bodies of homogeneous material the center of gravity and the center of magnitude are identical. The center of gravity of a regular figure is the same as its geometrical center, as in a circle, cylinder, circular ring, etc.

Center of Gravity of a Triangle.—The intersection of the lines joining the vertices with the mid-points of the sides, and at a perpendicular distance from any side equal to one-third of the corresponding altitude.

Of a Trapezium.—Draw a diagonal dividing it into two triangles. Draw a line joining their centers of gravity. Draw the other diagonal, making two other triangles and a line joining their centers. The intersection of these two lines is the center of gravity.

Of a Sector of a Circle.—On the radius bisecting the arc, a distance $2cr \div 3l$ from the center, c being the chord, r the radius, and l the length of arc.

Of a Semicircle.—On the middle radius $0.4244r$ from the center.

Of a Quadrant.—On the middle radius $0.6002r$ from the center.

Of a Segment of a Circle.—$c^3/12a$ from the center, c being the chord, a the area of the segment.

Of a Cone or Pyramid.—In the axis one-fourth of its length from the base.

The common center of gravity for two bodies is that point which divides the distance between their respective centers of gravity in inverse proportion to their weights. For more than two bodies, find the common center of gravity of any two as above. Then find the common center of these two jointly with a third, and so on.

STATICS

Simple Forces.—When two or more forces act upon a body at one point, they may be considered as being combined into a resultant force. Conversely, any force may be resolved into component forces. In Fig. 5 let the lines f_1, f_2 represent by their directions and lengths the directions and magnitudes, respectively, of two forces acting on a point O. The resultant force F is represented in direction and magnitude by the diagonal of the parallelogram of which f_1 and f_2 are the sides.

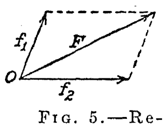


Fig. 5.—Resultant of simple forces.

Conversely, any force F may be resolved into component forces by a reverse of the above operation.

Moments.—The moment of a force with respect to a given point is the tendency of that force to produce rotation around it. The magnitude of the moment is represented by the product of the force and the perpendicular distance from its line of action to the point or center of moment. In the English system of weights

and measures, moments are expressed as the product of the force in pounds and the length of the moment arm in feet or inches, the unit of the moment being termed the "foot-pound" or the "inch-pound" (not to be confused with the unit of work of the same name). Moments acting in a clockwise direction are designated as positive and those acting in a counterclockwise direction are negative. They may be added and subtracted algebraically, as moments, regardless of the direction of the forces themselves.

Couples.—Two parallel forces of equal magnitude acting in opposite directions constitute a couple. The moment of the couple is the product of one of the forces and the perpendicular distance between the two. A couple has no single resultant and can be balanced only by another couple of equal moment of opposite sign.

Law of Equilibrium.—When a body is at rest, the external forces acting upon it must be in equilibrium. This means that (1) the algebraic sums of the components of all forces with reference to any three axes of reference at right angles with each other must each be zero; and (2) the algebraic sum of all moments with reference to any three such axes must be zero. When the forces all lie in the same plane the algebraic sums of their components with respect to any two axes must be equal to zero, and the algebraic sum of all moments with respect to any point in the plane must be zero.

WORK, POWER, AND ENERGY

Work.—When a body is moved against a resistance, or lifted against gravity, or when its motion is accelerated, work must be done upon it. The amount of work done is the product of the force and the distance through which it acts. The unit of work in the English system is the foot-pound, which is the amount of work done by a force of 1 lb acting through a distance of 1 ft. The following symbols are used in this section in defining the interrelation of work, power, and energy:

A = area, sq in. or sq ft as noted.
F = force, lb.
g = acceleration of gravity = 32.2 ft per sec.^{2}
h = vertical distance, ft.
H = thermal energy, Btu.
hp = horsepower.
kw = kilowatts.

$K.E.$ = kinetic energy, ft-lb.
$P.E.$ = potential energy, ft-lb.
p = pressure, psi.
l = distance, ft.
T = time, sec.
v = velocity, ft per sec.
V = volume change, cu ft.
w = weight, lb.
W = work, ft-lb.

According to the above definition of work, the following expressions may be written to represent work:

$$W = Fl = wh = pAl = pV, \text{ etc.}$$

The above expressions contain no term involving time, since the measure of work is independent of the time interval during which it is performed.

Power.—Power is the time rate of performing work. The English unit of power is the horsepower, which is defined as 33,000 ft-lb per min or 550 ft-lb per sec. Electrical power is commonly expressed in watts or kilowatts, 1 kw being equivalent to 1.34 hp and 1 hp to 0.746 kw. The expressions for horsepower corresponding to those given above for work are

$$hp = \frac{W}{550T} = \frac{Fl}{550T}, \text{ etc.}$$

Electrical power is the product of volts × amperes, *i.e.*,

$$kw = \frac{\text{volts} \times \text{amperes}}{1{,}000}.$$

Energy.—Energy is the capacity for doing work possessed by a system through virtue of work having previously been done upon it. Whenever work has been done upon a system in producing a *change* in either its *motion*, its *position*, or its *molecular condition*, the system has acquired the capacity for doing work in turn. Energy may be that due to motion, termed "kinetic energy"; that due to position, termed "potential energy"; or that due to molecular energy which is manifest usually as heat or as chemical stress. These three forms of energy are mutually convertible, one into another, and are possessed by a system as a result of work which has been done upon it. In the English system, the units of energy are the foot-pound and the Btu which are mutually convertible, according to the first law of thermodynamics, *i.e.*,

1 Btu is equivalent to 778 ft-lb. Some of the more common expressions for energy are as follows:

a. The potential energy of a body of weight w lb which has been raised h ft against gravity is $P.E. = wh$.

b. The kinetic energy possessed by a body of weight w lb moving at a velocity v ft per second is $K.E. = wv^2/2g$.

c. If the body of a were to fall freely through the distance h, its potential energy would be converted to kinetic energy and it would acquire a velocity v determined as follows:

$$P.E. = wh = K.E. = \frac{wv^2}{2g}, \text{ hence } wh \quad \frac{wv^2}{2h}, \text{ and } v = \sqrt{2gh}.$$

d. The molecular energy contained in a gas, as heat, is also available to produce velocity as follows, assuming 100 per cent efficiency in the conversion:

$$K.E. = 778H \text{ (since 1 Btu} = 778 \text{ ft-lb), and } K.E. = \frac{wv^2}{2g}.$$

Hence, $H = \dfrac{wv^2}{2g \times 778}$ and $v = \sqrt{\dfrac{50{,}000H}{w}}$ (see paragraph on Velocity Produced by a Given Change in Heat Content on page 70).

e. Energy is measured in the English system in horsepower-hours, kilowatt-hours, Btu, and foot-pounds. The relation between these units at 100 per cent efficiency is as follows:

1 hp-hr = 0.746 kw-hr = 2,546 Btu = 1,980,788 ft-lb.
1 kw-hr = 1.34 hp-hr = 3,412 Btu = 2,654,536 ft-lb.

HEAT AND TEMPERATURE

Units of Heat.—In the English system, the unit of heat is the mean British thermal unit (Btu) which is the $1/180$ part of the heat required to raise the temperature of 1 lb of water from 32 to 212 in the Fahrenheit scale. For approximate calculations, the Btu may be considered as the heat required to raise the temperature of 1 lb of water 1 F. In the metric system, the unit of heat is the mean (gram) calorie which is the one one-hundredth part of the heat required to raise the temperature of 1 g of water from 0 to 100 C. The large calorie, or kilogram calorie, is more convenient for engineering work. It is equal to 1,000 g cal; 1 kg cal = 3,968 Btu; and 1 Btu = 0.252 kg cal. Heat is a form of energy which is mutually convertible with work as explained under "Energy."

Units of Temperature.—The intensity of heat is denoted by the term "temperature." The temperature of a substance is measured by noting its effect upon a thermometer or pyrometer whose thermal properties are known. The mercury thermometer is suitable for measuring temperatures from −39 to about 600 F. This limit may be extended to 1000 F if the capillary tube above the mercury is filled with nitrogen or carbon dioxide under pressure. High temperatures must be measured with pyrometers, usually of the thermocouple or of the optical type. The most commonly used thermometer scales are the Fahrenheit and the Centigrade. Thermometer scales have as their bases the melting and boiling points of water. The relation of the Fahrenheit and Centigrade scales is as follows:

	Absolute zero	Freezing point of water	Boiling point of water
Degrees Fahrenheit.	−459.6	32	212
Degrees Centigrade.	−273	0	100

Hence, the relation of the two scales is

$$C = \tfrac{5}{9}(F - 32) \text{ and } F = \tfrac{9}{5}\,C + 32,$$

since the interval between the freezing and boiling of water is 180 on the Fahrenheit scale and 100 on the Centigrade. C = reading on Centigrade scale; F = reading on Fahrenheit scale.

A convenient rule for converting Centigrade readings to Fahrenheit is: multiply by 2, subtract one-tenth, and add 32. Example: $100\text{ C} \times 2 = 200$; $200 - 20 = 180$; $180 + 32 = 212$ F.

Specific Heat.—The specific heat of any substance may be defined as the amount of heat in Btu required to change the temperature of 1 lb of the substance 1 F. The name "specific heat" is derived from the ratio which the quantity of heat required to raise the temperature of 1 lb of a substance 1 deg bears to the quantity of heat required to raise the temperature of 1 lb of water 1 deg, the latter quantity being assumed as unity. Since this ratio has no exact significance, due to the change of specific heats with temperature, the first definition given above is preferred.

The specific heats of most substances change with change in temperature. Hence, in exact calculations, it is necessary to use the specific heat which applies for the temperature range in

question. The specific heat of any substance at a particular temperature is called its "instantaneous specific heat" at that temperature. The average specific heat over a range in temperature is termed its "mean specific heat" for that range. The use of a mean specific heat for a given temperature range will give practically the same result as an integration of the instantaneous specific heats for the same temperature range.

Gases have two specific heats, one at *constant volume* and one at *constant pressure*, as explained on page 145 in the section on Properties of Air.

Numerical values for the mean specific heats of common piping metals are given in Table I on page 344. Extensive data on both instantaneous and mean specific heats of gases and liquids are given in Chap. II on Fluids.

DEFINITIONS OF PIPING TERMS

Alloy Steel.—A steel which owes its distinctive properties to elements other than carbon. Alloy-steel billets are rolled into stud-bolt stock and pipe and forged into large-headed bolts, pipe flanges, etc. Cast alloy steel is used in valve bodies and fittings where high tensile strength at elevated temperatures is desired, or where resistance to corrosion is a factor (see page 322).

Backing Ring.—A strip of metal used to prevent weld splatter from entering a pipe when making a butt-welded joint and to ensure complete penetration of the weld to the inside of the pipe wall (see Fig. 15, Chap. IV).

Back-pressure Valve.—A valve similar to a low-pressure safety valve which is set to maintain a certain back pressure on feed heaters, oiling systems, or other devices requiring a constant operating pressure irrespective of pressure variations of the supply. The back-pressure valve is arranged to relieve any excess supply to atmosphere or elsewhere, and it opens and closes, automatically, as required to produce this result.

Blank Flange.—A flange that is not drilled but is otherwise complete. Compare Blind Flange.

Bleeder.—A small cock or valve to draw off water of condensation from a run of piping. A small connection to obtain circulation in warming up a line.

Blind Flange.—A flange used to close the end of a pipe. It produces a blind end which is also known as a dead end. Compare "Blank Flange."

Branch Tee.—A header or manifold having many side branches. A common example is the branch tee used in making up pipe coil radiators for building heating systems.

Brazed.—Connected by hard solder which usually is copper and zinc, half and half. Such solder requires a full red heat and is commonly used with borax flux. Pure copper is sometimes used for brazing.

Butt Weld.—Welded along a seam that is butted edge to edge and not scarfed or lapped. A term used to designate pipe made by this process. Also applied to circumferential pipe joints made by the fusion-welding process.

By-pass.—A small passage around a large valve for warming up a line. An emergency connection around a reducing valve, trap, etc., to use in case they are out of commission.

Carbon Steel.—A steel which owes its distinctive properties chiefly to the carbon (as distinguished from the other elements) which it contains. For physical and chemical properties of typical carbon steels for pipe fittings, see page 345.

Companion Flange.—A pipe flange suited to connect with another companion flange or with a flanged valve or fitting. A loose flange which is attached to a pipe by threading, Van Stoning, welding, or similar method as distinguished from a flange which is cast integral with a fitting or pipe.

Coupling.—A threaded sleeve used to connect two pipes. Commercial couplings have internal threads to fit external threads on pipe.

Creep or Plastic Flow of Metals.—At sufficiently high temperatures all metals flow under stress. The higher the temperature and stress, the greater the tendency to plastic flow for any given metal.

Double extra-strong refers to a schedule of wrought-pipe weights in common use (see Table III, page 359).

Extra-heavy.—The term "extra-heavy" was formerly used to designate cast-iron flanges, fittings, valves, etc., suitable for a maximum working steam pressure of 250 lb gauge. Owing to the general use of much higher pressures, the term "extra-heavy" in this sense has become inappropriate. The use of the term "extra-heavy" to denote "extra-strong" pipe is incorrect.

Extra-strong.—A schedule of wrought-pipe weights in common use. In sizes 8 in. and smaller, extra-strong pipe is identical with Schedule 80 pipe (see Tables II and XI, pages 358 and 367).

Forge Weld.—A method of manufacture similar to hammer welding. The term "forge welded" is applied more particularly to headers and large drums, while "hammer welded" usually refers to pipe.

Furnace Weld.—A term applied to the process of making butt-welded or lap-welded pipe in which the skelp is heated in a furnace preparatory to welding by passing through rolls.

Fusion Weld.—The union of metals by fusion, using an oxy-acetylene torch, the electric arc, or thermit reaction. With the first two methods, the edges to be joined usually are chamfered or beveled to give an included angle of 45 to 90 deg, which is filled in with fused metal from a welding rod. This is also known as an "autogenous weld."

Galvanizing.—A process by which the surface of iron or steel is covered with a layer of zinc.

Hammer Weld.—Method of manufacturing large pipe (usually 20 in. and larger) by bending a plate into circular form, heating the overlapped edges to a welding heat and welding the longitudinal seam with a power hammer applied to the outside of the weld while the inner side is supported on an overhung anvil.

Lapped Joint.—Same as Van Stone. A type of pipe joint made by using loose flanges on lengths of pipe whose ends are lapped over to give a bearing surface for a gasket or metal-to-metal joint (see Fig. 18, page 508 and page 521).

Lap Weld.—Welded along a scarfed longitudinal seam in which one part is overlapped by the other. A term used to designate pipe made by this process.

Malleable Iron.—Cast iron which has been heat-treated in a malleableizing oven to relieve its brittleness. The process somewhat improves the tensile strength and enables the material to stretch to a limited extent without breaking.

Mill Length.—Also known as "random length." The usual run-of-mill pipe is 16 to 20 ft in length. Line pipe and pipe for power-plant use are sometimes made in double lengths of 30 to 35 ft.

Nipple.—A piece of pipe less than 12 in. long and threaded on both ends. Pipe over 12 in. long is regarded as cut pipe. Common types of nipples are: close nipple, about twice the length of a standard pipe thread and without any shoulder; shoulder nipple, of any length and having a shoulder between the pipe threads; short nipple, a shoulder nipple slightly longer than a close nipple

and of a definite length for each pipe size which conforms to manufacturer's standard; long nipple, a shoulder nipple longer than a short nipple which is cut to specific length.

Nonreturn Valve.—A stop valve whose disk can move independently of the stem so that the valve can act as a check. Such valves are largely used between boilers and headers to prevent steam from the header entering the boiler in case of tube failure or other trouble necessitating shutdown. The name "stop and check valve" is often applied to this type (see Fig. 9, page 888).

Nozzle.—As applied to piping, this term refers to a flanged connection on a boiler, tank, or manifold consisting of a pipe flange, a short neck, and a riveted or welded attachment to the boiler or other vessel.

Pipe.—The name "pipe" is applied to tubular products of dimensions and materials commonly used for pipe lines and connections, formerly designated as "iron pipe size" (IPS). The outside diameter of all weights and kinds of IPS pipe is of necessity the same for a given pipe size on account of threading.

Relief Valve.—A valve arranged to provide an automatic relief in case of excess pressure. It may be either spring loaded or of the dead-weight type.

Resistance Weld.—Method of manufacturing pipe by bending a plate into circular form and passing electric current through the material to obtain a welding heat.

Reynolds Number.—The Reynolds number is a dimensionless factor or coefficient used in correlating pressure drop with the inside diameter of the pipe, the velocity of flow, and the mass density and absolute viscosity of the fluid.

Run.—The portion of a fitting having its end in line, or nearly so, as distinguished from branch connections, side outlets, etc.

Saddle Flange.—Also known as "tank flange" or "boiler flange." A curved flange shaped to fit a boiler, tank, or other vessel, and receive a threaded pipe. A saddle flange is usually riveted or welded to the vessel.

Safety Valve.—A relief valve for expansive fluids provided with a huddling ring and chamber to control the amount of blowback before the valve reseats.

Sargol.—A special type of joint in which a lip is provided for welding to make the joint fluid tight, while mechanical strength is provided by bolted flanges. The Sargol joint is used with both Van Stone pipe and fittings (see illustration on page 508).

Sarlun.—An improved type of Sargol joint.

Schedule Numbers.—Schedule numbers indicate approximate values of the expression $1{,}000 \times P/S$ where P is the service pressure and S is the allowable stress, both expressed in pounds per square inch. See Tables V and VI, pages 361 and 362, for complete list of schedule numbers of the American Standard for wrought-iron and wrought-steel pipe (ASA B36.10).

Seamless.—Pipe formed by piercing and rolling a solid billet or cupping from a plate is termed "seamless."

Semi-steel.—A high grade of cast iron made by the addition of steel scrap to pig iron in the cupola or electric furnace. More correctly described as "high-strength gray iron." It is used to some extent for valve bodies and fittings.

Service Fitting.—A street ell or street tee having a male thread at one end.

Skelp.—A piece of plate prepared by forming and bending, ready for welding into pipe. Flat plates when used for butt-welded pipe are called "skelp."

Source Nipple.—A short length of heavy-walled pipe between steam lines and the first valve of by-pass drain or instrument connections.

Spiral Riveted.—A method of manufacturing pipe by coiling a plate into a helix and riveting together the overlapped edges.

Spiral Welded.—A method of manufacturing pipe by coiling a plate into a helix and fusion-welding the overlapped or abutted edges.

Stainless Steel.—An alloy steel having unusual corrosion-resisting properties, usually imparted by nickel and chromium (see page 333).

Standard.—The term "standard" was formerly used to designate cast-iron flanges, fittings, valves, etc., suitable for a maximum working steam pressure of 125 lb gauge. The multiplicity of standards which have recently come into use in connection with high pressures makes the old use of this word confusing.

Standard Weight.—A schedule of wrought-pipe weights in common use. Standard-weight pipe in sizes 10 in. and smaller is identical with Schedule 40 pipe (see Tables I and VI, pages 357 and 362).

Stop and Check Valve.—See Nonreturn Valve.

Stop Valve.—A valve of the gate or globe type used to shut off a line.

Stress Relieving.—A term applied to the process of heating welded assemblies and pipe joints to a temperature of 1100 to 1300 F to permit locked-in stresses to relieve themselves through creep.

Van Stone.—See lapped joint.

Welding Fittings.—Wrought- or forged-steel elbows, tees reducers, heads, saddles, and the like, beveled for butt welding to pipe. Fittings with hubs or with ends counterbored for fillet welding to pipe are used to some extent for small pipe sizes.

Welding-end Valves.—Valves without end flanges and with ends tapered and beveled for butt welding to pipe. Small valves may be counterbored to provide sockets for fillet welding to pipe.

Wrought Iron.—Iron refined in a plastic state in a puddling furnace. It is characterized by the presence of about 3 per cent of slag irregularly mixed with pure iron and about 0.5 per cent carbon and other elements in solution.

Wrought Pipe.—The term "wrought pipe" refers to both wrought steel and wrought iron. Wrought in this sense means "worked" as in the process of forming furnace-welded pipe from skelp, or seamless pipe from plates or billets. The expression "wrought pipe" is thus used as a distinction from cast pipe. Wrought pipe in this sense should not be confused with "wrought-iron pipe" which is only one variety of wrought pipe. When "wrought-iron pipe" is referred to, it should be designated by its complete name.

TABLE I.—UNITS OF WEIGHT AND MEASURE WITH CONVERSION FACTORS

From Carl Hering's "Conversion Tables."

The values of units used in the U. S. are given unless otherwise indicated.

ap = apothecary; av = avoirdupois; Br = British; cm = centimeter; g = gram; kg = kilogram; km = kilometer; m = meter; ml = milliliter; mm = millimeter; U. S. = United States.

Acre = 208.710 ft sq (length of each side of square) = 43,560 sq ft = 4,840 sq yd = 0.00156250 sq mile = 0.404687 hectare = 4,046.87 sq m.

Acre-foot (irrigation) = 43,560 cu ft = 325,851 gal (U. S.) = 1,233.49 cu m.

Barrel (liquid, U. S.) = No legal value. The following values are customary to some extent:—31½ and 31 gal (U. S.); 42 gal (U. S.—Standard Oil Co.).

TABLE I.—(*Continued*)

Butt or pipe, see pipe.

Centigram = 0.01 g = 0.154324 grain.

Centimeter = 0.01 m = 0.0328083 ft = 0.393700 in. = 393.700 mils.

Centimeter, circular = 100 cir mm = 78.5398 sq. mm = 0.785398 sq cm = 0.155000 cir in. = 0.121736 sq. in. = 155,000 cir mils.

Centimeter² (square centimeter) = 127.324 cir mm = 100 sq mm = 1.27324 cir cm = 0.01 sq decimeter = 197,352 cir mils = 155,000 sq mils = 0.197352 cir in. = 0.155000 sq in.

Centimeter³ (cubic centimeter) = 0.999973 milliliter = 0.000-999973 liter = 0.001 cu decimeter = 0.0610234 cu in.

Circular mil, circular inch, circular centimeter, etc., see mil, inch, centimeter, etc.

Cubic inch, cubic centimeter, etc., see inch, centimeter, etc.

Decigram = 0.1 g = 1.54324 grains

Deciliter = 0.1 liter = 0.1000027 cu decimeter = 6.10250 cu in.

Decimeter² (square decimeter) = 0.01 sq m = 15.5000 sq in. = 0.1076387 sq ft = 0.0119599 sq yd.

Decimeter³ (cubic decimeter) = 0.999973 liter.

Dekameter = 10 m = 10.9361 yd.

Dram (av) = 1/16 oz (av) = 27.34375 grains = 0.455729 dram (ap) = 1.77185 g.

Dram (ʒ) (ap) = 1/8 oz (troy or ap) = 3 scruples = 60 grains = 2.19429 drams (av) = 3.8879351 g.

Foot (Br) = 0.9999971 ft (U. S.) = 30.4800 cm.

Foot (U. S.) = 12,000 mils = 12 in. = 1/3 yd = 1/5,280 or 0.000189394 statute mile (U. S.) = 30.4801 cm = 1.0000029 ft (Br).

Foot, circular = 144 cir in. = 113.097 sq in. = 0.785398 sq ft = 929.034 cir cm = 729.662 sq cm.

Foot² (square foot) (Br) = 0.9999942 sq ft (U. S.) = 0.0929029 sq m.

Foot² (square foot) (U. S.) = 144 sq in. = 1/9 or 0.111111 sq yd = 183.346 cir in. = 1.27324 cir ft = 929.034 sq cm = 1.000-0057 sq ft (Br).

Foot³ (cubic foot) = 1,728 cu in. = 0.0370370 cu yd = 28.-316265 liter = 28.3170 cu decimeters = 7.48052 gal (U. S.) = 0.803564 bu (U. S.).

Gallon, Imperial (Br) = 32 gills (Br) = 8 pt (Br) = 4 qt (Br) = 0.5 pk (Br) = 1/8 or 0.125 bu (Br) = 4.5458269 liter = 4.5459631

TABLE I.—(*Continued*)

cu decimeters = 1.20091 gal (U. S.) = 277.410 cu in. = 4,545.9631 cu cm.

Gallon (liquid; U. S.) = 231 cu in. = 0.133681 cu ft = 3.78533 liter = 3.78543 cu decimeters = 3,785.43 cu cm = 32 gills (U. S.) = 8 pt (liquid; U. S.) = 4 qt (liquid; U. S.) = 0.8327024 gal (Br.)

Gill (Br) = ¼ pt (Br) = 1/32 or 0.03125 gal (Br).

Gill (liquid; U. S.) = ¼ pt (liquid; U. S.) = 1/32 gal (U. S.).

Grain (same in av, troy, and ap weights) = 1/7,000 lb (av) = 0.00228571 oz (av) = 1/5,760 lb (troy or ap) = 0.0647989 g.

Gram = 0.001 kg = 15.43235639 grains = 0.564383 dram (av) = 0.0352740 oz (av) = 0.00220462 lb (av) = 0.771618 scruple = 0.257206 dram (ap) = 0.0321507 oz (troy or ap).

Gram-molecule of any gas at 0 C and 760 mm of mercury pressure has a volume of 22,380 cu cm.

Hectare = 100 ares = 2.47104 acres = 10,000 sq m = 11,959.9 sq yd.

Hectoliter = 100 liter = 3.53145 cu ft.

Hectometer = 100 m = 109.361 yd.

Hogshead (liquid; U. S.) = 63 gal (U. S.) = 2 bbl of 31½ gal (U. S.) = 238.4759 liter.

Hundredweight, short = 100 lb (av) = 1/20 or 0.05 short or net ton = 45.35924 kg.

Hundredweight, long = 112 lb (av) = 1/20 or 0.05 long or gross ton = 50.8024 kg.

Inch = 1,000 mils = 1/12 ft = 1/36 yd = 2.540005 cm.

Inch, circular = 1,000,000 cir mils = 1/144 or 0.00694444 cir ft = 785.398 sq mils = 0.785398 sq in. = 0.00545415 sq ft = 645.163 cir mm = 6.45163 cir cm = 506.709 sq mm = 5.06709 sq cm.

Inch^2 (square inch) = 1/144 or 0.00694444 sq ft = 1,000,000 sq mils = 1,273,240 cir mils = 1.27324 cir in. = 6.45163 sq cm = 8.21447 cir cm.

Inch^3 (cubic inch) = 1/1728 or 0.000578704 cu ft = 16.38716 cu cm = 16.3867 ml = 0.0163867 liter = 0.01638716 cu decimeter.

Kilogram or **kilo** = 1,000 g = 0.001 metric ton = 15,432.4 grains = 35.2740 oz (av) = 2.20462 lb (av) = 0.0220462 cwt (short) = 0.0196841 cwt (long) = 0.00110231 short or net ton = 0.000984206 long or gross ton = 32.1507 oz (troy or ap).

Kilometer = 1,000 m = 3,280.83 ft = 1,093.61 yd = 0.621370 statute mile (U. S.) = 0.539593 nautical mile (U. S.).

TABLE I.—(*Continued*)

Kilometer² (**square kilometer**) = 100 hectares = 247.104 acres = 1,195,985 sq yd = 0.386101 sq mile.

Liter = 1.000027 cu decimeters = 10 deciliters = 1,000.027 cu cm = 0.01 hectoliter = 0.001000027 cu m = 61.0250 cu in. = 0.0353134 cu ft = 2.11342 pt (liquid; U. S.) = 1.05671 qt (liquid; U. S.) = 0.264178 gal (U. S.) = 1.81620 pt (dry; U. S.) = 0.908102 qt (dry; U. S.) = 0.113513 pk (U. S.) = 0.028378 bu (U. S.) = 1.7598475 pt (Br) = 0.8798475 qt (Br) = 0.2199809 gal (Br) = 0.10999097 pk (Br) = 0.02749764 bu (Br).

Meter (international) = 0.001 km = 0.01 hectometer = 0.1 dekameter = 10 decimeters = 100 cm = 1,000 mm = 1,000,000 micro-meters = 39.370113 in. (Br) = 39.37 in. exact legal value (U. S.) = 3.28083 ft (U. S.) = 1.09361 yd (U. S.) = 0.000621370 statute mile (U. S.).

Meter² (**square meter**) = 10,000 sq cm = 100 sq decimeters = 0.01 are = 1,550 sq in. = 10.76387 sq ft (U. S.) = 1.19599 sq yd (U. S.) = 10.76393 sq ft (Br) = 1.19599 sq yd (Br) = 0.000247104 acre.

Meter³ (**cubic meter**) = 999.973 liter = 1.30794 cu yd.

Micro-meter, micron, or microne = 0.001 mm.

Mil = 0.001 in. = 0.02540005 mm.

Mil, circular = 0.000001 cir in. = 0.785398 sq mil = 0.000000785398 sq in. = 0.000645163 cir mm = 0.000506709 sq mm.

Mil² (**square mil**) = 0.000001 sq in. = 0.00000127324 cir in. = 1.27324 cir mils = 0.000821447 cir mm = 0.000645163 sq mm.

Mile (Br) = 5,280 ft = 1,760 yd = practically the same as statute mile (U. S.).

Mile, international geographical = 24,350.3 ft = 8,116.77 yd = 7.422 km = 7,422 m = 4.6118 statute miles (U. S.) = $\frac{1}{15}$ of 1 deg at the equator.

Mile, international nautical = 6,076.10 ft = 2,025.37 yd = 1.852 km = 1,852 m = $\frac{1}{60}$ of 1 deg of meridian at the equator.

Mile, nautical (Br) = 6,080 ft = 2,026.67 yd = 0.999966 nautical mile (U. S.) = 1.15152 statute miles (U. S.) = 1.85319 km.

Mile, nautical (U. S.) same as sea mile and geographical mile (all U. S.) = 6,080.20 ft = 1.85325 km = 1.15155 statute miles (U. S.) = 1.000034 nautical miles (Br) = 1 minute of earth's circumference.

TABLE I.—(*Continued*)

Mile, statute or land (U. S.) = 5,280 ft = 1.60935 km.

Mile² **(square mile)** = 640 acres = 3,097,600 sq yd = 2.59000 sq km.

Milligram = 0.001 g = 0.0154324 grain.

Millimeter = 0.001 m = 39.370 mils = 0.039370 in.

Millimeter, circular = 0.01 cir cm = 0.785398 sq mm = 1,550 cir mils = 1,217.36 sq mils = 0.00121736 sq in.

Millimeter² **(square millimeter)** = 0.01 sq cm = 1.27324 cir mm = 1,973.52 cir mils = 1,550 sq mils = 0.001550 sq in.

Ounce (℥) same as troy ounce = 480 grains = 24 scruples = 8 drams (ap) = 1/12 or 0.0833333 lb (troy or ap) = 3.1035 g.

Ounce (av) = 16 drams (av) = 1/16 or 0.062500 lb (av) = 437.500 grains = 28.3495 g = 0.911458 oz (troy or ap).

Ounce (troy, gold and silver) same as ap ounce = 480 grains = 20 dwt = 1/12 or 0.0833333 lb (troy or ap) = 0.0684714 lb (av) = 31.1035 g = 1.09714 oz (av).

Ounce, fluid (ap U. S.) = 480 minims (ap U. S.) = 8 fluid drams (ap U. S.) = 1/16 pt (ap U. S.) = 1.80469 cu in = 0.0295729 liter.

Pennyweight (troy) = 24 grains = 1.55517 g = 1/20 oz (troy or ap).

Pint (ap U. S.) same as ordinary U. S. liquid pint = 128 fluid drams (ap U. S.) = 16 fl oz (ap U. S.) = 0.473167 liter.

Pint (Br) = 0.125 gal (Br) = 0.015625 bu (Br) = 0.568230047 liter.

Pint (dry; U. S.) = 0.5 qt (dry; U. S.) = 0.550599 liter.

Pint (liquid; U. S.) = 0.125 gal (U. S.) = 0.473167 liter.

Pipe or **butt** (liquid; U. S.) = 126 gal (U. S.) = 2 hogsheads (U. S.) = 476.9471 liters.

Pound (av) = 7,000 grains = 16 oz (av) = 256 drams (av) = 14,5833 oz (troy or ap) = 453.5924277 g = 0.4535924277 kg = 7,000/5,760 or 1.21528 lb (troy or ap).

Pound (troy or ap) = 5,760 grains = 12 oz (troy or ap) = 0.373242 kg = 5,760/7,000 or 0.822857 lb (av).

Quart (Br) = 0.25 gal (Br) = 1.03202 qt (dry; U. S.) 1.20091 qt (liquid; U. S.) = 1.1364593 liter.

Quart (dry; U. S.) = 2 pt (dry; U. S.) = 1/8 or 0.125 pk (U. S.) = 1/32 or 0.031250 bu (U. S.) = 67.200625 cu in. = 0.0388893 cu ft = 1.101198 liter = 1,101.23 cu cm = 11.01198 deciliters = 1.16365 qt (liquid; U. S.) = 0.968972 qt (Br) = 0.242243 gal (Br).

Quart (liquid; U. S.) = 0.25 gal (U. S.) = 0.946333 liter.

TABLE I.—(*Concluded*)

Scruple (℈) (ap) = 20 grains = 1/3 dr (ap) = 1/24 oz (troy or ap) = 1.295978 gr.

Section (of land) = 1 mile sq = 640 acres.

Square (building) = 100 sq ft.

Square inch, square centimeter, square mil, etc., see inch, mil, centimeter, etc.

Stone (Br) = 14 lb (av) = 6.35029 kg.

Ton (gross) **displacement of water** = 35.8813 cu ft = 1.01605 cu m.

Ton, register (shipping, for whole vessels) = 100 cu ft = 2.8317 cu m.

Ton, long or gross = 2,240 lb (av) = 1.12 short or net tons = 1016.05 kg = 1.01605 metric tons.

Ton, short or **net** = 2,000 lb (av) = 20 cwt (short) = 907.185 kg = 0.907185 metric ton = 17.8571 cwt (long) = 0.892857 long or gross ton.

Ton, metric, tonne, tonneau, millier, or bar = 2,204.62 lb (av) = 1.10231 short or net tons = 0.984206 long or gross ton = 1,000 kilograms.

Yard (Br) = 0.9999971 yd (U. S.) = 0.9143992 m.

Yard (U. S.) = 36 in. = 3 ft = 1.0000029 yd (Br) = 0.914402 m.

Yard² (**square yard**) (Br) = 0.9999943 sq yd (U. S.) = 0.836126 sq m.

Yard² (**square yard**) (U. S.) = 1,296 sq in. = 9 sq ft = 1/4,840 or 0.000206612 acre = 0.836131 sq m = 1.0000057 sq yd (Br).

Yard³ (**cubic yard**) = 27 cu ft = 46,656 cu in. = 0.764559 cu m.

TABLE II.—DECIMAL EQUIVALENTS OF EIGHTHS, SIXTEENTHS, THIRTY-SECONDS, AND SIXTY-FOURTHS OF AN INCH

Eighths	9/32 = 0.28125	19/64 = 0.296875
1/8 = 0.125	11/32 = 0.34375	21/64 = 0.328125
1/4 = 0.250	13/32 = 0.40625	23/64 = 0.359375
3/8 = 0.375	15/32 = 0.46875	25/64 = 0.390625
1/2 = 0.500	17/32 = 0.53125	27/64 = 0.421875
5/8 = 0.625	19/32 = 0.59375	29/64 = 0.453125
3/4 = 0.750	21/32 = 0.65625	31/64 = 0.484375
7/8 = 0.875	23/32 = 0.71875	33/64 = 0.515625
Sixteenths	25/32 = 0.78125	35/64 = 0.546875
1/16 = 0.0625	27/32 = 0.84375	37/64 = 0.578125
3/16 = 0.1875	29/32 = 0.90625	39/64 = 0.609375
5/16 = 0.3125	31/32 = 0.96875	41/64 = 0.640625
7/16 = 0.4375	Sixty-fourths	43/64 = 0.671875
9/16 = 0.5625	1/64 = 0.015625	45/64 = 0.703125
11/16 = 0.6875	3/64 = 0.046875	47/64 = 0.734375
13/16 = 0.8125	5/64 = 0.078125	49/64 = 0.765625
15/16 = 0.9375	7/64 = 0.109375	51/64 = 0.796875
Thirty-seconds	9/64 = 0.140625	53/64 = 0.828125
1/32 = 0.03125	11/64 = 0.171875	55/64 = 0.859375
3/32 = 0.09375	13/64 = 0.203125	57/64 = 0.890625
5/32 = 0.15625	15/64 = 0.234375	59/64 = 0.921875
7/32 = 0.21875	17/64 = 0.265625	61/64 = 0.953125
		63/64 = 0.984375

Table III.—Wire and Sheet-metal Gauges
(Diameters and thicknesses in decimal parts of an inch)

Gauge No.	American wire gauge, or Brown and Sharpe (for copper wire)	Steel wire gauge, or Washburn and Moen or Roebling (for steel wire)	Birmingham wire gauge (B.W.G.) or Stubs' iron wire (for steel wire or sheets)	Stubs steel wire gauge	British Imperial standard wire gauge (S.W.G.)	U. S. standard gauge for sheet metal (iron and steel) 480 lb per cu ft	AISI inch equivalent for U. S. steel sheet thickness	British standard for iron and steel, sheets and hoops 1914 (B.G.)
0000000	......	0.4900	.....	.....	0.500	0.500	......	0.6666
000000	......	0.4615	.....	.....	0.464	0.469	......	0.625
00000	......	0.4305	.....	.....	0.432	0.438	......	0.5883
0000	0.460	0.3938	0.454	.....	0.400	0.406	......	0.5416
000	0.410	0.3625	0.425	.....	0.372	0.375	......	0.5000
00	0.365	0.3310	0.380	.....	0.348	0.344	......	0.4452
0	0.325	0.3065	0.340	.....	0.324	0.312	......	0.3964
1	0.289	0.2830	0.300	0.227	0.300	0.281	......	0.3532
2	0.258	0.2625	0.284	0.219	0.276	0.266	......	0.3147
3	0.229	0.2437	0.259	0.212	0.252	0.250	0.2391	0.2804
4	0.204	0.2253	0.238	0.207	0.232	0.234	0.2242	0.2500
5	0.182	0.2070	0.220	0.204	0.212	0.219	0.2092	0.2225
6	0.162	0.1920	0.203	0.201	0.192	0.203	0.1943	0.1981
7	0.144	0.1770	0.180	0.199	0.176	0.188	0.1793	0.1764
8	0.128	0.1620	0.165	0.197	0.160	0.172	0.1644	0.1570
9	0.114	0.1483	0.148	0.194	0.144	0.156	0.1495	0.1398
10	0.102	0.1350	0.134	0.191	0.128	0.141	0.1345	0.1250
11	0.091	0.1205	0.120	0.188	0.116	0.125	0.1196	0.1113
12	0.081	0.1055	0.109	0.185	0.104	0.109	0.1046	0.0991
13	0.072	0.0915	0.095	0.182	0.092	0.094	0.0897	0.0882
14	0.064	0.0800	0.083	0.180	0.080	0.078	0.0747	0.0785
15	0.057	0.0720	0.072	0.178	0.072	0.070	0.0673	0.0699
16	0.051	0.0625	0.065	0.175	0.064	0.062	0.0598	0.0625
17	0.045	0.0540	0.058	0.172	0.056	0.056	0.0538	0.0556
18	0.040	0.0475	0.049	0.168	0.048	0.050	0.0478	0.0495
19	0.036	0.0410	0.042	0.164	0.040	0.0438	0.0418	0.0440
20	0.032	0.0348	0.035	0.161	0.036	0.0375	0.0359	0.0392
21	0.0285	0.0317	0.032	0.157	0.032	0.0344	0.0329	0.0349
22	0.0253	0.0286	0.028	0.155	0.028	0.0312	0.0299	0.0313
23	0.0226	0.0258	0.025	0.153	0.024	0.0281	0.0269	0.0278
24	0.0201	0.0230	0.022	0.151	0.022	0.0250	0.0239	0.0248
25	0.0179	0.0204	0.020	0.148	0.020	0.0219	0.0209	0.0220
26	0.0159	0.0181	0.018	0.146	0.018	0.0188	0.0179	0.0196
27	0.0142	0.0173	0.016	0.143	0.0164	0.0172	0.1064	0.0175
28	0.0126	0.0162	0.014	0.139	0.0148	0.0156	0.0149	0.0156
29	0.0113	0.0150	0.013	0.134	0.0136	0.0141	0.0135	0.0139
30	0.0100	0.0140	0.012	0.127	0.0124	0.0125	0.0120	0.0123
31	0.0089	0.0132	0.010	0.120	0.0116	0.0109	0.0105	0.0110
32	0.0080	0.0128	0.009	0.115	0.0108	0.0102	0.0097	0.0098
33	0.0071	0.0118	0.008	0.112	0.0100	0.0094	0.0090	0.0087
34	0.0063	0.0104	0.007	0.110	0.0092	0.0086	0.0082	0.0077
35	0.0056	0.0095	0.005	0.108	0.0084	0.0078	0.0075	0.0069
36	0.0050	0.0090	0.004	0.106	0.0076	0.0070	0.0067	0.0061
37	0.0045	0.0085	.....	0.103	0.0068	0.0066	0.0064	0.0054
38	0.0040	0.0080	.....	0.101	0.0060	0.0062	0.0060	0.0048
39	0.0035	0.0075	.....	0.099	0.0052	......	......	0.0043
40	0.0031	0.0070	.....	0.097	0.0048	......	......	0.0039
41	......	0.0066	.....	0.095	0.0044	......	......	0.0034
42	......	0.0062	.....	0.092	0.0040	......	......	0.0031
43	......	0.0060	.....	0.088	0.0036	......	......	0.0027
44	......	0.0058	.....	0.085	0.0032	......	......	0.0024
45	......	0.0055	.....	0.081	0.0028	......	......	0.0022
46	......	0.0052	.....	0.079	0.0024	......	......	0.0019
47	......	0.0050	.....	0.077	0.0020	......	......	0.0017
48	......	0.0048	.....	0.075	0.0016	......	......	0.0015
49	......	0.0046	.....	0.072	0.0012	......	......	0.0014
50	......	0.0044	.....	0.069	0.0010	......	......	0.0012

TABLE IV.—CONTENTS, IN CUBIC FEET AND U. S. GALLONS, OF PIPES AND CYLINDRICAL TANKS OF VARIOUS DIAMETERS AND 1 FT. IN LENGTH, WHEN COMPLETELY FILLED[1]

(1 gal. = 231 cu. in. 1 cu. ft. = 7.4805 gal.)

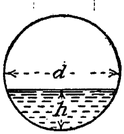
Diameter, in inches	For 1 ft. in length		Length, in inches, of cylinder of 1 cu. ft. capacity	Diameter, in inches	For 1 ft. in length		Length, in inches, of cylinder of 1 cu. ft. capacity
	Cubic feet; also, area in square feet	U. S. gal., 231 cu. in.			Cubic feet; also, area in square feet	U. S. gal., 231 cu. in.	
12½	0.8522	6.375	14.080	21¼	2.463	18.42	4.872
12⅝	0.8693	6.503	13.800	21½	2.521	18.86	4.760
12¾	0.8866	6.632	13.530	21¾	2.580	19.30	4.651
12⅞	0.9041	6.763	13.270	22	2.640	19.75	4.545
13	0.9218	6.895	13.020	22¼	2.700	20.20	4.445
13⅛	0.9395	7.028	12.780	22½	2.761	20.66	4.347
13¼	0.9575	7.163	12.530	22¾	2.823	21.12	4.251
13⅜	0.9757	7.299	12.300	23	2.885	21.58	4.160
13½	0.994	7.436	12.070	23¼	2.948	22.05	4.070
13⅝	1.013	7.578	11.850	23½	3.012	22.53	3.990
13¾	1.031	7.712	11.640	23¾	3.076	23.01	3.901
13⅞	1.051	7.855	11.420	24	3.142	23.50	3.819
14	1.069	7.997	11.230	25	3.409	25.50	3.520
14⅛	1.088	8.139	11.030	26	3.678	27.58	3.263
14¼	1.107	8.281	10.840	27	3.976	29.74	3.018
14⅜	1.127	8.431	10.650	28	4.276	31.99	2.806
14½	1.147	8.578	10.460	29	4.587	34.31	2.616
14⅝	1.167	8.730	10.280	30	4.909	36.72	2.444
14¾	1.187	8.879	10.110	31	5.241	39.21	2.290
14⅞	1.207	9.029	9.940	32	5.585	41.78	2.149
15	1.227	9.180	9.780	33	5.940	44.43	2.020
15⅛	1.248	9.336	9.620	34	6.305	47.16	1.903
15¼	1.268	9.485	9.460	35	6.681	49.98	1.796
15⅜	1.289	9.642	9.310	36	7.069	52.88	1.698
15½	1.310	9.801	9.160	37	7.467	55.86	1.607
15⅝	1.332	9.964	9.010	38	7.876	58.92	1.527
15¾	1.353	10.121	8.870	39	8.296	62.06	1.446
15⅞	1.374	10.278	8.730	40	8.727	65.28	1.375
16	1.396	10.440	8.600	41	9.168	68.58	1.309
16¼	1.440	10.772	8.330	42	9.621	71.91	1.247
16½	1.485	11.11	8.081	43	10.085	75.44	1.190
16¾	1.530	11.45	7.843	44	10.559	78.99	1.136
17	1.576	11.79	7.511	45	11.045	82.62	1.087
17¼	1.623	12.14	7.394	46	11.541	86.33	1.040
17½	1.670	12.49	7.186	47	12.048	90.13	0.996
17¾	1.718	12.85	6.985	48	12.566	94.00	0.955
18	1.768	13.22	6.787	49	13.095	97.96	0.916
18¼	1.817	13.59	6.604	50	13.635	102.00	0.880
18½	1.867	13.96	6.427	51	14.186	106.12	0.846
18¾	1.917	14.34	6.259	52	14.748	110.32	0.814
19	1.969	14.73	6.094	53	15.320	114.60	0.783
19¼	2.021	15.12	5.938	54	15.904	118.97	0.755
19½	2.074	15.51	5.786	55	16.499	122.82	0.727
19¾	2.128	15.92	5.639	56	17.104	127.95	0.702
20	2.182	16.32	5.500	57	17.720	132.55	0.677
20¼	2.237	16.73	5.365	58	18.347	137.24	0.654
20½	2.292	17.15	5.236	59	18.985	142.02	0.632
20¾	2.348	17.56	5.110	60	19.637	146.89	0.611
21	2.405	17.99	4.989				

[1] For contents of pipes 12 in. nominal size and smaller see Tables I to XII on pp. 357–368.

To find the capacity of pipes greater than the largest given in the table, look in the table for a pipe one-half the given size and multiply its capacity by 4, or one of one-third its size, and multiply its capacity by 9, etc.

TABLE V.—CONTENTS OF PIPES AND CYLINDRICAL TANKS[1]—AXIS HORIZONTAL—FLAT ENDS—CONTENTS PER FOOT OF LENGTH FOR ANY DEPTH OF LIQUID

h = Depth of liquid inches	d = diameter of tank, inches									
	12		18		24		30		36	
	Gal.	Cu. ft.	Gal.	Cu. ft.	Gal.	Cu. ft.	Gal.	Cu. ft.	Gal.	Cu. ft.
2	0.64	0.0860	0.80	0.1072	0.93	0.1244	1.05	0.1400	1.15	0.154
4	1.73	0.2317	2.18	0.2920	2.57	0.3440	2.90	0.3878	3.21	0.429
6	2.94	0.3927	3.85	0.5149	4.59	0.6140	5.23	0.6988	5.80	0.775
8	4.14	0.5537	5.67	0.7578	6.85	0.9152	7.85	1.049	8.75	1.17
10	5.23	0.6994	7.55	1.009	9.26	1.238	10.72	1.432	12.0	1.60
12	5.87	0.7854	9.38	1.252	11.75	1.571	13.72	1.833	15.4	2.03
14	.....	.....	11.04	1.476	14.24	1.903	16.82	2.248	19.0	2.54
16	.....	.....	12.43	1.659	16.65	2.226	19.90	2.660	22.6	3.02
18	.....	.....	13.22	1.767	18.91	2.527	23.00	3.075	26.4	3.53
20	.....	.....	.....	.....	20.93	2.797	26.00	3.476	29.6	3.95
22	.....	.....	.....	.....	22.57	3.017	28.85	3.859	33.4	4.46
24	.....	.....	.....	.....	23.50	3.1416	31.49	4.209	37.4	5.00
26	.....	.....	.....	.....	.....	.....	33.82	4.521	40.4	5.40
28	.....	.....	.....	.....	.....	.....	35.67	4.768	43.7	5.84
30	.....	.....	.....	.....	.....	.....	36.72	4.908	46.6	6.23
32	.....	.....	.....	.....	.....	.....	.....	.....	49.1	6.55
34	.....	.....	.....	.....	.....	.....	.....	.....	51.2	6.85
36	.....	.....	.....	.....	.....	.....	.....	.....	52.9	7.07
38										
40										
42										
44										
46										
48										
50										
52										
54										
56										
58										
60										
64										
68										
72										
76										
80										
84										



Formulas for determination of approximate capacity of horizontal cylindrical tanks for any depth. (Given: diameter of tank d and height of segment h.)

To find area of segment

when $h = 0$ to $\frac{1}{4}d$; area $= h\sqrt{1.766dh - h^2}$

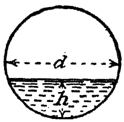
when $h = \frac{1}{4}d$ to $\frac{1}{2}d$; area $= h\sqrt{0.017d^2 + 1.7dh - h^2}$

1 cu. ft. = 7.4805 U. S. gal.

[1] Continued on pp. 25 and 26.

TABLE V.—(*Continued*)

h = depth of liquid inches	d = diameter of tank, inches							
	42		48		54		60	
	Gal.	Cu. ft.	Gal.	Cu. ft.	Gal.	Cu. ft.	Gal.	Cu. ft.
2	1.25	0.167	1.36	0.182	1.43	0.191	1.47	0.197
4	3.49	0.465	3.72	0.496	3.98	0.531	4.19	0.560
6	6.31	0.843	6.90	0.921	7.25	0.967	7.48	1.00
8	9.57	1.28	10.3	1.37	11.0	1.47	11.6	1.55
10	13.3	1.77	14.2	1.89	15.2	2.02	16.2	2.16
12	16.9	2.26	18.6	2.48	19.7	2.63	21.0	2.81
14	21.0	2.80	22.8	3.04	24.2	3.23	26.3	3.52
16	25.2	3.36	27.4	3.66	29.4	3.92	31.4	4.19
18	29.4	3.92	32.3	4.31	34.8	4.64	36.9	4.93
20	33.8	4.51	37.0	4.94	40.1	5.35	42.8	5.72
22	38.2	5.10	42.0	5.61	45.6	6.08	48.8	6.53
24	42.5	5.67	47.0	6.27	51.0	6.80	54.7	7.30
26	46.8	6.25	52.0	6.94	56.7	7.56	61.0	8.15
28	50.9	6.80	57.0	7.61	62.3	8.33	66.9	8.94
30	55.0	7.34	61.7	8.23	67.8	9.05	73.4	9.81
32	58.8	7.86	66.6	8.89	73.4	9.79	79.7	10.7
34	62.4	8.19	71.3	9.52	78.8	10.5	85.9	11.5
36	65.6	8.75	75.7	10.1	84.2	11.2	92.6	12.4
38	68.4	9.13	79.9	10.7	89.4	11.9	98.0	13.1
40	70.7	9.44	83.7	11.2	94.4	12.6	105	14.0
42	72.0	9.61	87.4	11.7	99.5	13.3	110	14.7
44	.....	.....	90.3	12.1	104	13.9	115	15.4
46	.....	.....	92.7	12.4	108	14.4	121	16.2
48	.....	.....	94.0	12.6	112	14.9	126	16.8
50	.....	.....	.....	.....	115	15.4	131	17.5
52	.....	.....	.....	.....	117	15.6	135	18.0
54	.....	.....	.....	.....	119	15.9	139	18.6
56	.....	.....	.....	.....	.....	.....	143	19.1
58	.....	.....	.....	.....	.....	.....	145	19.4
60	.....	.....	.....	.....	.....	.....	147	19.6
64								
68								
72								
76								
80								
84								



Formulas for determination of approximate capacity of horizontal cylindrical tanks for any depth. Given:—diameter of tank d and height of segment h.
To find area of segment

when $h = 0$ to $\frac{1}{4}d$; area =

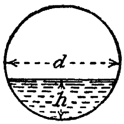
when $h = \frac{1}{4}d$ to $\frac{1}{2}d$; area =

1 cu. ft. = 7.4805 U. S. gal.

(*Continued on p. 26*)

TABLE V.—(*Concluded*)

h = depth of liquid, inches	d = diameter of tank, inches							
	66		72		78		84	
	Gal.	Cu. ft.	Gal.	Cu. ft.	Gal.	Cu. ft.	Gal.	Cu. ft.
2	1.57	0.210	1.65	0.220	1.73	0.229	1.77	0.236
4	4.42	0.580	4.64	0.618	4.81	0.641	4.95	0.661
6	8.04	1.07	8.10	1.16	8.78	1.17	9.13	1.22
8	12.2	1.63	12.8	1.71	13.4	1.78	13.9	1.86
10	17.0	2.27	17.7	2.36	18.6	2.48	19.7	2.63
12	22.1	2.96	23.6	3.14	24.2	3.22	25.7	3.43
14	27.6	3.68	28.9	3.85	30.2	4.03	31.5	4.21
16	33.3	4.45	35.0	4.66	36.1	4.81	38.2	5.10
18	39.3	5.25	41.7	5.55	43.3	5.78	45.8	6.12
20	45.5	6.08	48.0	6.40	50.3	6.72	52.5	7.01
22	51.8	6.93	54.7	7.29	57.6	7.69	60.0	8.02
24	58.3	7.79	61.9	8.25	64.8	8.65	68.8	9.19
26	65.0	8.68	68.7	9.15	72.1	9.63	75.7	10.1
28	71.8	9.59	76.0	10.2	80.0	10.7	84.1	11.2
30	78.6	10.5	83.5	11.1	88.0	11.8	91.6	12.3
32	85.4	11.4	90.7	12.1	96.0	12.8	101	13.5
34	92.3	12.3	98.2	13.1	104	13.9	109	14.5
36	99.1	13.2	106	14.1	112	14.9	117	15.6
38	106	14.2	113	15.1	120	16.0	126	16.8
40	113	15.1	121	16.1	128	17.1	135	18.0
42	119	15.9	128	17.1	136	18.2	144	19.2
44	126	16.8	136	18.2	144	19.2	153	20.4
46	132	17.6	143	19.1	152	20.3	162	21.6
48	138	18.4	150	20.0	160	21.4	171	22.8
50	144	19.2	157	21.0	168	22.4	179	23.9
52	150	20.0	164	21.9	176	23.5	187	25.0
54	156	20.8	170	22.7	183	24.4	196	26.2
56	161	21.5	176	23.5	191	25.5	204	27.2
58	165	22.0	182	24.3	198	26.4	212	28.3
60	169	22.6	188	25.1	205	27.4	219	29.2
64	176	23.5	198	26.4	218	29.1	235	31.4
68	.....	.....	207	27.6	230	30.7	250	33.4
72	.....	.....	211	28.2	239	31.9	262	35.0
76	.....	.....	.....	.....	246	32.8	274	36.6
80	.....	.....	.....	.....	.....	.....	283	37.8
84	.....	.....	.....	.....	.....	.....	288	38.5



Formulas for determination of approximate capacity of horizontal cylindrical tanks for any depth. Given:—diameter of tank d and height of segment h.
To find area of segment

when $h = 0$ to $\frac{1}{4}d$; area $= h\sqrt{1.766dh - h^2}$

when $h = \frac{1}{4}d$ to $\frac{1}{2}d$; area $= h\sqrt{0.017d^2 + 1.7dh - h^2}$

1 cu. ft. = 7.4805 U. S. gal.

CHAPTER II

FLUIDS

PROPERTIES OF FLUIDS

PROPERTIES COMMON TO ALL FLUIDS

Definition of a Fluid.—A fluid is a substance in a liquid, gaseous, or vapor state which offers little or no resistance to distortion of form. The most common examples of these three states are: water as a liquid; air as a gas; and saturated or partially saturated steam as a vapor.

Definition of a Liquid.—A liquid is a fluid which offers a high resistance to compression, *i.e.*, it changes volume very slightly with considerable variation in pressure and when the pressure is removed does not sensibly dilate. Like all elastic substances, it is actually compressible to a limited extent (see discussion of the compressibility of water on page 266).

Definition of a Gas.—A gas is a compressible fluid, *i.e.*, it responds to a change in pressure with a considerable change in volume. The properties and laws of gases are treated on pages 142 to 146.

Definition of a Vapor.—A liquid may be changed to a gas by the application of heat; and a gas may be changed to liquid form by the abstraction of heat. A gas at the point of becoming a liquid, or a liquid on the point of becoming a gas, is called a "vapor."

Free Surface of a Liquid.—That surface of a liquid at rest which is in contact with a gas or vapor is termed the "free surface." It is a surface of equal pressure normal to the line of action of the force of gravity at any point on the surface of the earth, and in ordinary engineering work may be considered as horizontal.

FLUID PRESSURE

Pascal's Law.—The basic theorem of fluid pressure known as "Pascal's law" may be stated as follows: *The pressure at any given*

point in a fluid at rest is normal to the surface on which it acts and is of equal intensity in all directions. For illustration see Fig. 1, page 1287.

Intensity of Pressure.—The intensity of fluid pressure p is the pressure per unit of area between contiguous fluid surfaces, or between a fluid and the face of a solid, and is the weight of a column of fluid supported by a unit area. The intensity at any point is measured, therefore, by the product of the vertical distance h from that point to the free surface of the liquid and the weight w of a cubic unit of the fluid. Intensity of pressure is independent of the shape, area, or direction of the surface presented to the fluid. If all factors are given in corresponding units of the same system, the intensity of pressure may be expressed by the following formula

$$p = hw \text{ and } h = \frac{p}{w}.$$

In the case of two or more fluids of different densities which occupy different strata in the same vessel, the intensity of fluid pressure at any point is the sum of the pressures due to each. For instance, if there were above this point h_1 ft of the first liquid of density w_1 and h_2 ft of the second liquid of density w_2, the total pressure p in pounds per square foot at that point would be

$$p = h_1w_1 + h_2w_2 = p_1 + p_2. \tag{1}$$

A similar condition exists in a closed vessel, such, for instance, as a hydropneumatic pressure tank in which a compressible gas (air) is above a liquid (water). Here an air pressure p_0 is due to compression, and the pressure at any point below the free surface of the water is

$$p = p_0 + h_1w_1 \tag{2}$$

For all ordinary purposes, however, the value of [illegible]$_1$ is negligible, and the equation may be written

$$p = p_0 + h_2w_2 = p_0 + p_2. \tag{3}$$

Conversion Formula for Intensity of Pressure.—The terms most commonly used to express intensity of pressure are feet or inches of water or mercury; pounds per square inch or square foot; and kilograms per square meter or square centimeter. The following formulas serve to convert various expressions for head and intensity of pressure into units of any system desired:

CONVERSION OF HEAD TO PRESSURE

$$p = yh \quad \text{or} \quad h = \frac{p}{y}. \qquad (4)$$

h = the head of the given fluid in linear units corresponding to an intensity of pressure p.

p = intensity of pressure per unit of area produced by a head h. p may be stated in any units of weight and area.

y = the weight of a column of the given fluid expressed in the same unit of weight as p, this column having for its height one linear unit in which h is measured, and for its cross-sectional area, the unit area in which p is given.

NOTE.—w as used in the preceding section is a specific value of y.

CONVERSION OF PRESSURE FROM UNITS OF ONE SYSTEM TO THOSE OF ANOTHER

$$p_2 = xp_1 \quad \text{or} \quad p_1 = \frac{p_2}{x}. \qquad (5)$$

p_1 = pressure per unit area in first system.

p_2 = pressure per unit area in second system.

$$x = \frac{\text{number of area units of first system in one area unit of second}}{\text{number of weight units of first system in one weight unit of second}}.$$

CONVERSION OF HEAD OF FLUID *A* EXPRESSED IN LINEAR UNITS OF ONE SYSTEM INTO HEAD OF FLUID *B* EXPRESSED IN LINEAR UNITS OF ANOTHER SYSTEM

$$h_2 = zh_1 \quad \text{or} \quad h_1 = \frac{h_2}{z}. \qquad (6)$$

h_1 = head of fluid A expressed in linear units of first system.

h_2 = head of fluid B expressed in linear units of second system.

$$z = \frac{\text{specific gravity or density of fluid } A}{(\text{specific gravity or density of fluid } B) \times (\text{number of linear units of first system in one linear unit of second system})}.$$

Numerical Values of the Conversion Factors x, y, and z.—Numerical values for these conversion factors, which have been calculated for a number of cases frequently encountered in engineering work, are given in Table I.

Definition of Heads.—The term "head" has various special meanings in connection with fluids [illegible] these conceptions of head may be

TABLE I.—CONVERSION FACTORS FOR UNITS OF PRESSURE AND HEAD

To convert from \ to	Pounds per square inch	Ounces per square inch	Pounds per square foot	Kilograms per square meter	Kilograms per square centimeter	Atmospheres	Feet of fresh water	Feet of sea water	Inches of fresh water	Meters of fresh water	Inches of mercury	Millimeters of mercury
	Multiply by											
Pounds per square inch	1.000	16.000	144.00	703.067	0.07031	0.06804	2.3067	2.2504	27.68	0.70307	2.036	51.712
Ounces per square inch	0.0625	1.000	9.000	43.942	0.004394	0.004253	0.14417	0.14065	1.7300	0.04394	0.1272	3.2320
Pounds per square foot	0.006945	0.1111	1.000	4.8824	0.0004882	0.0004725	0.01602	0.01563	0.1922	0.004882	0.01414	0.3591
Kilograms per square meter	0.001422	0.02275	0.2048	1.000	0.0001	0.00009676	0.003281	0.003201	0.03937	0.0010	0.002896	0.07355
Kilograms per square centimeter	14.223	227.57	2,048.2	10,000	1.000	0.9676	32.81	32.01	393.72	10.000	28.96	735.51
Atmospheres	14.696	235.136	2,116.35	10,332.9	1.0333	1.000	33.90	33.07	406.80	10.3329	29.921	760.00
Feet in height of fresh water	0.4335	6.9360	62.428	304.801	0.03048	0.02949	1.000	0.9756	12.000	0.3048	0.8826	22.419
Feet in height of sea water	0.4443	7.1094	63.9887	312.420	0.03124	0.03024	1.0250	1.000	12.300	0.3124	0.9047	22.979
Inches in height of fresh water	0.03613	0.5780	5.2023	25.400	0.002540	0.002458	0.08333	0.08130	1.000	0.0254	0.07355	1.8682
Meters in height of fresh water	1.422	22.756	204.82	1,000	0.1000	0.09678	3.2808	3.2008	39.3696	1.000	2.8957	73.5514
Inches in height of mercury	0.4912	7.859	70.731	345.34	0.03453	0.03342	1.1329	1.1053	13.596	0.3453	1.000	25.400
Millimeters in height of mercury	0.01934	0.3094	2.7847	13.596	0.001359	0.001316	0.04461	0.04352	0.5353	0.01359	0.03937	1.000

Example.—To convert from atmospheric pressure in pounds per square inch (14.696) to inches of mercury or feet of water: $14.696 \times 2.036 = 29.921$ in. Hg and $14.696 \times 2.3067 = 33.90$ ft water.

NOTE.—The following values were used in calculating this table: Specific gravity of mercury = 13.596; specific gravity of sea water = 1.025 (average); density of fresh water = 62.428 lb per cu ft.

The *velocity head* of a fluid moving with a given velocity is the equivalent height through which a body must fall to acquire the same velocity. It may be expressed in symbols as

$$h = \frac{v^2}{2g} \tag{7}$$

in which v represents velocity and g the acceleration due to gravity, both expressed in the same units of space and time. In common engineering practice, h is the vertical feet of head of the particular fluid; v is its velocity in feet per second; g is the acceleration due to gravity of 32.2 ft per sec per sec. Velocity head is that head which represents kinetic energy due to directional motion of the fluid through a pipe, flume, or similar conduit.

The *pressure head* or *static head* of a fluid is that head which is due to the application of external force, such as is produced by a pump or a compressible fluid (gas or vapor) with which it is in contact, or by the weight of a column of fluid which it supports.

The *elevation head* is that head representing vertical distance above the elevation taken as datum.

The *friction head* represents that loss of head in a moving fluid expended in overcoming frictional resistance to flow.

Heads or pressures are measured from some reference point or datum fixed by conditions of the problem. Intensity of pressure, unless otherwise stated, is measured from atmospheric pressure as zero and is called "gauge pressure." *Absolute pressure* is gauge pressure plus atmospheric pressure. The use of the terms "head" and "pressure" is interchangeable when considered in connection with the conversion formula given above, *i.e.*, $p = yh$.

STRESS DUE TO FLUID PRESSURE

Stress Reactions Produced in Cylinders by Fluid Pressure.— In analyzing the stress conditions set up in the walls of a vessel by internal or external fluid pressure, consideration must be given to the way the walls of the vessel react to support the pressure. The most common form of vessel supporting fluid pressure has, in general, the properties of a right circular hollow cylinder. Pipes, tubes, and other circular conduits come under this classification and, for convenience, the term "cylinder" will be here considered as embracing all such conduits. It should be clearly understood at the start, in order to avoid needless repetition, that the stress formulas here derived apply to sections through the cylinder walls so far removed from closed ends, end flanges, or other stiffening

elements as not to receive any reinforcement from them. Ordinarily, this condition is met at a distance of four to six cylinder diameters from the reinforcement.

Internal or external fluid pressure may give rise (1) to a tangential or hoop stress within the wall of the cylinder, (2) to both a hoop and a longitudinal stress acting jointly. In either case the wall is also under a direct radial compressive stress whose magnitude is usually so small with regard to the other two that it may be neglected. Case 1 is illustrated in Fig. 1, and Case 2, which represents the usual condition, is illustrated in Fig. 2.

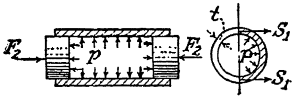


FIG. 1.—Stress conditions in cylinder with open ends.

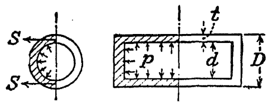


FIG. 2.—Stress conditions in cylinder with closed ends.

The following symbols are used throughout this section to designate the corresponding items in the same system of measure:

d = internal diameter of cylinder (or pipe), linear units.
D = external diameter of cylinder, linear units.
t = thickness of cylinder wall, same linear units.
p = internal fluid pressure, weight units per area unit.
P = external fluid pressure, weight units per area unit.
S = fiber stress, same weight unit per area unit.
λ = Poisson's ratio (coefficient of lateral contraction) for values see Table I, page 344, and Table XLIX, page 296.
F = force, same weight units produced by a pressure p acting on a given area.

Where additional symbols or subscripts are used, their significance is explained in the immediate context.

Figure 1 illustrates a cylinder with frictionless plungers fitted into its ends, the plungers being kept in place by external forces F_2 which exactly balance the internal fluid pressure tending to force them outward. In this case the tube wall is subjected to only the internal forces shown as acting at right angles to its inner surface. It is obvious that these forces can give rise to radial and hoop stresses only in the tube wall.

Figure 2 illustrates the ordinary case of a cylinder with both ends closed. In this case the tube wall is subjected not only to the hoop and radial stresses mentioned above but, at the same

time, to a longitudinal stress at right angles to the tangential or hoop stress.

Derivation of Formula for Hoop Stress.—The derivation of the formula for hoop or tangential stress is as follows: Let Fig. 3 represent one-half portion of a cylinder of length l. Consider the half cylinder as a free body and resolve all the elementary radial forces due to internal pressure into their components perpendicular to the cutting plane. The component of these elementary forces perpendicular to the cutting plane is represented by $plrd\theta \sin \theta$. The total force normal to this plane is the summation of these elementary forces over one-half circumference, or from 0 to π in radian measure. This summation may be written as follows:

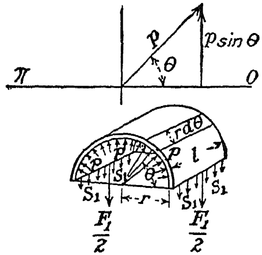


FIG. 3.—Derivation of hoop stress.

$$F_1 = \int_0^{\pi} plrd\theta \sin \theta = plr \int_0^{\pi} \sin \theta d\theta = 2plr = pld.$$

This normal force is opposed by an equal force on the other half of the cylinder, acting in the opposite direction 180 deg from the first force. These opposing forces F_1 produce a hoop stress S_1 in the cylinder walls equal to the force F_1 divided by the area over which the stress is distributed. This area is $2tl$, and $S_1 = F_1/2tl$ or $F_1 = 2tlS_1$. Equating the two values obtained for F_1, $pld = 2tlS_1$, canceling l, and transposing,

$$S_1 = \frac{pd}{2t}. \tag{8}$$

These relations are independent of the length of the cylinder, provided the section is far enough removed from the flanges or other reinforcements to receive no support from them. Equation (8) is generally known as the "common formula for bursting pressure" and is frequently used in calculations of bursting strength for thin-walled cylinders without taking into account the effect of lateral contraction described on page 38. In equation (8) it is assumed that the hoop stress is uniformly distributed across the cylinder wall. This condition does not exist, except in the case of cylinders having walls of infinitesimal thickness.

Empirical formulas such as Barlow's formula on page 41 and modifications of Barlow's formula used in various codes differ

from the common formula for bursting pressure in the substitution of the outside diameter for the inside diameter.

Derivation of Formula for Longitudinal Stress.—The derivation of the formula for longitudinal stress in a cylinder with closed ends is as follows: Let Fig. 4 represent a section of a cylinder with closed ends. The total force F_2 acting against each end of the cylinder is $F_2 = \frac{p\pi d^2}{4}$. This force is distributed over an area of metal in the cylinder wall equal to $\pi(d + t)t$ and the unit stress is $S_2 = F_2 \div \pi(d + t)t$. Substituting the value above,

$$S_2 = \frac{p\pi d^2}{4} \div \pi(d + t)t.$$

In the case of thin-walled cylinders the value of $(d + t)$ approaches d, and an approximate equation may be written

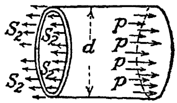


FIG. 4.—Derivation of longitudinal stress.

$$S_2 = \frac{p\pi d^2}{4} \div \pi dt = \frac{pd}{4t} \qquad (9)$$

which is equal to one-half the hoop stress of equation (8).

Radial Stress.—There is a direct compression on the inside surface of a pipe equal to the internal pressure. As this stress is small in relation to the hoop and longitudinal stresses, it is usually neglected although it has some bearing on the theories of bursting strength applicable to pipe. It is designated as S_3 for convenience in combining with the hoop and longitudinal stresses in equation (10).

Combination of Hoop, Longitudinal, and Radial Stresses.—When a material is subjected to mutually perpendicular stresses, as in the case of pipe under internal pressure, it is necessary to employ some theory of strength to relate failure of material under the action of such stresses to failure of the same material under simple tensile stress in a testing machine. Four principal theories have been evolved based on different concepts of the manner in which material fails under a combination of stresses. This failure may be due to exceeding a given value of: (1) the maximum tensile stress; (2) the maximum strain; (3) the maximum shear stress: or (4) a function of the differences of the principal stresses.

In the first, or maximum-stress theory, only hoop tension is considered to be related to failure of the pipe, the longitudinal and radial stresses having no effect.

In the second, or maximum-strain theory, the longitudinal stress is considered to strengthen the material, the amount of the strengthening depending upon the value of Poisson's ratio for the particular material. Clavarino's formula on page 38 is a direct application of the maximum-strain theory. The third theory, that of maximum shear, holds that failure is dependent only on the greatest and least stresses acting. Since the longitudinal tension is intermediate in value between the hoop tension and the radial compression, it does not, according to this theory, affect the stress at which failure will occur.

The fourth theory, known as the Mises-Hencky maximum-energy-of-distortion theory, holds that yielding failure takes place in accordance with a relation given by the differences of the principal stresses. This relation is expressed by the following equation:

$$(S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_3 - S_1)^2 = 2S_0^2 \qquad (10)$$

in which S_1, S_2, and S_3 are the principal stresses, hoop, longitudinal, and radial, respectively, and S_0 is the yield strength of the material determined from simple tensile tests.

Collapsing Pressure.—If a thin-walled ring or short section of pipe is subjected to external fluid pressure, a compressive stress is set up in the walls. By an analysis similar to that used in the case of internal pressure, it can be shown that here again $S = pD/2t$, S being a compressive stress and D the outside diameter.

Taking the effect of lateral contraction into account, the collapsing pressure of a long circular tube having a perfectly circular form is given by the following equation:[1]

3

(11)

where P = critical value of external pressure, E = modulus of elasticity, λ = Poisson's ratio, and t and D are wall thickness and mean diameter of tube, respectively, all dimensions being expressed in inches. For usual pipe-wall thicknesses, D may be taken as the outside diameter. This formula can be used for calculating the critical value of the collapsing pressure providing the corre-

[1] See "Collapse by Instability of Thin Cylindrical Shells under External Pressure," by D. F. Windenburg and Charles Trieling, *Trans. ASME*, Vol. 6, 1934; "Vessels under External Pressure," by D. F. Windenburg, *Mechanical Engineering*, Vol. 59, pp. 601–607, August, 1937; and "Study of Collapsing Pressures of Thin-walled Cylinders," by R. G. Sturm, *Univ. Illinois Bull.* 39 (12), p. 77, 1941.

sponding compressive stress does not exceed the yield strength of the material. The limiting value of the ratio t/D may be found from the relation

$$S = \frac{E}{1 - \lambda^2}\left(\frac{t}{D}\right)^2. \tag{12}$$

The above equations apply to ideal tubes of perfectly circular

TABLE II.—MAXIMUM UNIFORM EXTERNAL PRESSURE FOR STEEL PIPE, PSI

$\frac{D}{t}$	Critical[1] pressure, round pipe	Maximum pressure, pipe with 1% initial eccentricity[2] — Yield stress, psi			$\frac{D}{t}$	Critical[1] pressure, round pipe	Maximum pressure, pipe with 1% initial eccentricity[2] — Yield stress, psi		
		30,000	40,000	60,000			30,000	40,000	60,000
10	8,000	4,600	6,000	6,980	160	16.0	13.2	13.7	15.3
20	4,000	1,800	2,200	2,850	170	13.6	11.4	11.7	12.9
30	2,130	910	1,060	1,280	180	11.5	9.7	10.0	11.0
40	1,000	500	550	630	190	9.75	8.32	8.55	9.28
50	528	284	320	380	200	8.25	7.12	7.32	7.88
60	322	190	200	240	210	7.02	6.16	6.30	6.70
70	200	130	143	160	220	6.10	5.40	5.55	5.90
80	129	90	100	110	230	5.36	4.80	4.90	5.20
90	91	65	66	80	240	4.76	4.30	4.40	4.62
100	66	48	53	56	250	4.22	3.80	3.90	4.15
110	50	38	41	43	260	3.80	3.40	3.50	3.80
120	38	30	33	35	270	3.40	3.04	3.22	3.40
130	31	24	26	28	280	3.05	2.70	2.80	2.95
140	25	20	21	23	290	2.72	2.40	2.50	2.60
150	20	15	16	19	300	2.44	2.18	2.20	2.30

[1] Computed by equation (11) for values of D/t above the limits given by equation (12). For values of D/t below 20 the critical stress is taken as equal to the yield stress. Intermediate values are based on straight-line interpolation between the yield stress and the critical stress corresponding to the limiting D/t ratio given by equation (12).
[2] Computed by the quadratic equation

$$P_e^2 - \left[2S\left(\frac{t}{D}\right) + \left(1 + 0.03\frac{D}{t}\right)\cdot P\right]P_e + 2S\left(\frac{t}{D}\right)P = 0$$

where P_e = maximum external pressure for pipe having initial eccentricity of 1 per cent, psi. For other eccentricities adjust the factor 0.03 in the ratio of the percentages.
S = yield stress, psi.
P = critical value of collapsing pressure for perfectly circular pipe from equation (11), psi.
For definition of other symbols see p. 32.
From "Theory of Elastic Stability," by S. Timoshenko, *op. cit.*

form. Since commercial pipe may deviate as much as 1 per cent over or under the specified outside diameter,[1] and since the failure of pipe under external pressure is profoundly affected by initial

[1] For tolerances on pipe outside-diameter variations see ASTM Spec A120, p. 272, A139 p. 401, and A106 p. 382.

ellipticity, formulas have been developed[1] which enable maximum values of external pressure to be calculated on the basis of the initial ellipticity.

Values of critical stress and the *maximum* external pressures which steel pipe having an initial ellipticity of 1 per cent can support without collapsing are given in Table II. Depending on the likelihood of shock and other considerations, a factor of safety of from 2 to 4 should be allowed in setting *safe* external pressures. For example, if the *maximum* external pressure difference given in Table II is 20 psi, the *safe* external pressure difference for design purposes would be 5 to 10 psi. The figures in this table are for steel having a modulus of elasticity, $E = 30 \times 10^6$ psi. The safe external pressure for brass, copper, and other materials with a lower modulus of elasticity and yield strength may be obtained for values of D/t greater than 50 by multiplying by the ratio of their modulus of elasticity to that of steel. For values of D/t less than 50 the external pressure for the lower strength material should be adjusted in the ratio of the yield strengths of the materials.

For comparison with the above values of maximum working pressure for an assumed initial ellipticity of 1 per cent, the collapsing pressures shown in Table III have been computed from empirical formulas deduced by Prof. A. P. Carman[2] and by Prof. Reid T. Stewart[3] from tests on steel tubes.

TABLE III.—COLLAPSING PRESSURE OF SEAMLESS-STEEL TUBES DEDUCED FROM COLLAPSING TESTS

D/t	10	20	30	40	50	60	70	80	90	100
Carman	7,462	2,690	1,100	785	402	232	146	78	69	50
Stewart	7,281	2,954	1,504	782	402	232	146	78	69	50

The collapsing pressures deduced from tests are higher than the safe working external pressures computed for an initial ellipticity of 1 per cent for the thicker pipes and slightly lower for pipe thinner than those having a D/t ratio greater than 70. The values of safe working pressure based on the formulas developed by Dr. S. Timoshenko represent a more rational solution of the problem.

Thick-walled Cylinders—Clavarino's Formula.—In the case of a thick-walled pipe or cylinder subjected to internal or external

[1] "Theory of Elastic Stability," by S. Timoshenko, McGraw-Hill Book Company, Inc., New York, 1936, p. 222.

[2] "Tests at University of Illinois Experiment Station," *Bull.* 17, June, 1906.

[3] "Tests on Lap-welded Bessemer Steel Tubes," by Reid T. Stewart, *Trans. ASME*, Vol. 27.

fluid pressure, it can no longer be assumed that the stress due to bursting or collapsing pressure is uniformly distributed across the wall. In either case the stress will be greatest at the inner surface and decrease to a minimum at the outer surface. As with thin-walled cylinders, the stress is tensile with internal pressure, and compressive with external. According to Clavarino,[1] for either thin- or thick-walled cylinders, if

p = internal unit pressure, psi.
P = external unit pressure, psi.
r = the radius to any point in the wall, in.
d = internal diameter of cylinder, in.
D = external diameter of cylinder, in.
S = unit stress at any radius r, psi.
S_m = unit stress at the inner surface (point of maximum stress).
λ = Poisson's ratio (coefficient of lateral contraction) for values see Table I, page 344, and Table XLIX, page 296.

For simultaneous internal and external pressure

$$S = \frac{[(1 - 2\lambda)(d^2p - D^2P)] + \left[(1 + \lambda)\frac{d^2D^2}{4r^2}(p - P)\right]}{D^2 - d^2} \tag{13}$$

In the case of internal pressure alone where $P = 0$ and $r = d/2$, the maximum stress may be written

$$S_m = p\frac{d^2(1 - 2\lambda) + D^2(1 + \lambda)}{D^2 - d^2} \tag{14}$$

or

$$p = S_m \frac{D^2 - d^2}{d^2(1 - 2\lambda) + D^2(1 + \lambda)}.$$

For convenience in the solution of the above formula, the expression $\frac{D^2 - d^2}{d^2(1 - 2\lambda) + D^2(1 + \lambda)}$ may be termed k and the formula written $p = kS_m$ or $S_m = p/k$. In the case of steel pipe or cylinders where λ is approximately 0.3, the expression for k becomes $k = \frac{10(D^2 - d^2)}{4d^2 + 13D^2}$. Since the value of k depends on the ratio of t/D, a schedule of approximate values for k based on this relation may be worked out, as has been done for steel cylinders in Table IV. Through the use of this table, the solution of Clavarino's formula for the case of internal pressure in steel pipes or cylinders with closed ends is greatly simplified.

[1] For a derivation of Clavarino's formula see "Mechanics of Materials," by M. and T. Merriman, John Wiley & Sons, Inc., New York, 1928, p. 393.

TABLE IV.—INTERNAL FLUID PRESSURE FACTORS k FOR CONDITIONS SHOWN IN FIG. 2[1]

[Calculated by Clavarino's formula, assuming for steel a "coefficient of lateral contraction" (Poisson's ratio) equal 0.3]

Rule.—Divide thickness of tube or pipe by its outside diameter, both being expressed in inches, then multiply the tabular value corresponding to this quotient by the working-fiber stress in pounds per square inch. The result will be the safe internal pressure in pounds per square inch.

For further use of table, see examples, page 40.

t/D	0.000	0.001	0.002	0.003	0.004	0.005	0.006	0.007	0.008	0.009
0.01	0.0235	0.0259	0.0282	0.0306	0.0329	0.0352	0.0376	0.0399	0.0423	0.0446
0.02	0.0470	0.0493	0.0517	0.0540	0.0564	0.0587	0.0610	0.0634	0.0657	0.0681
0.03	0.0704	0.0727	0.0751	0.0774	0.0797	0.0821	0.0844	0.0867	0.0891	0.0914
0.04	0.0937	0.0961	0.0984	0.1007	0.1031	0.1054	0.1077	0.1100	0.1123	0.1147
0.05	0.1170	0.1193	0.1216	0.1239	0.1263	0.1286	0.1309	0.1332	0.1355	0.1378
0.06	0.1401	0.1424	0.1448	0.1471	0.1494	0.1517	0.1540	0.1563	0.1586	0.1609
0.07	0.1632	0.1655	0.1678	0.1700	0.1723	0.1746	0.1769	0.1792	0.1815	0.1838
0.08	0.1861	0.1883	0.1906	0.1929	0.1952	0.1974	0.1997	0.2020	0.2043	0.2065
0.09	0.2088	0.2111	0.2133	0.2156	0.2178	0.2201	0.2223	0.2246	0.2269	0.2291
0.10	0.2314	0.2336	0.2358	0.2381	0.2403	0.2425	0.2448	0.2470	0.2493	0.2515
0.11	0.2537	0.2559	0.2582	0.2604	0.2626	0.2648	0.2670	0.2692	0.2715	0.2737
0.12	0.2759	0.2781	0.2803	0.2825	0.2847	0.2869	0.2890	0.2912	0.2934	0.2956
0.13	0.2978	0.3000	0.3022	0.3043	0.3065	0.3087	0.3108	0.3130	0.3152	0.3173
0.14	0.3195	0.3216	0.3238	0.3259	0.3281	0.3302	0.3323	0.3345	0.3366	0.3388
0.15	0.3409	0.3430	0.3451	0.3472	0.3494	0.3515	0.3536	0.3557	0.3578	0.3599
0.16	0.3620	0.3641	0.3662	0.3683	0.3704	0.3724	0.3745	0.3766	0.3787	0.3808
0.17	0.3828	0.3849	0.3869	0.3890	0.3910	0.3931	0.3951	0.3972	0.3992	0.4013
0.18	0.4033	0.4053	0.4073	0.4094	0.4114	0.4134	0.4154	0.4174	0.4194	0.4214
0.19	0.4234	0.4254	0.4274	0.4294	0.4314	0.4333	0.4353	0.4373	0.4393	0.4412
0.20	0.4432	0.4452	0.4471	0.4490	0.4510	0.4529	0.4548	0.4568	0.4587	0.4606
0.21	0.4626	0.4645	0.4664	0.4683	0.4702	0.4721	0.4740	0.4758	0.4777	0.4796
0.22	0.4815	0.4834	0.4852	0.4871	0.4889	0.4908	0.4926	0.4945	0.4964	0.4982
0.23	0.5001	0.5019	0.5037	0.5055	0.5073	0.5091	0.5109	0.5127	0.5145	0.5163
0.24	0.5181	0.5199	0.5216	0.5234	0.5252	0.5269	0.5287	0.5304	0.5322	0.5340
0.25	0.5357	0.5374	0.5391	0.5408	0.5426	0.5443	0.5460	0.5477	0.5494	0.5511
0.26	0.5528	0.5545	0.5561	0.5578	0.5594	0.5611	0.5628	0.5644	0.5661	0.5677
0.27	0.5694	0.5710	0.5726	0.5742	0.5758	0.5774	0.5790	0.5806	0.5822	0.5838
0.28	0.5854	0.5870	0.5885	0.5901	0.5916	0.5932	0.5947	0.5963	0.5978	0.5994
0.29	0.6009	0.6024	0.6039	0.6054	0.6069	0.6084	0.6099	0.6114	0.6129	0.6143
0.30	0.6158	0.6173	0.6187	0.6201	0.6216	0.6230	0.6244	0.6259	0.6273	0.6287

NOTE.—Curves for ready solution of Clavarino's formula for internal pressures of 1,000 lb and above were given in an article on "Design of Thick Walled Tubes and Cylinders," by F. E. Wertheim, *Heating, Piping and Air Conditioning*, October, 1931.

[1] Reproduced from "National Pipe Standards," by permission of The National Tube Co.

Example 1.—Required the safe working internal fluid pressure p when D is 5.0 in., wall thickness t is 0.625, maximum working fiber stress of steel S_m is 10,000 psi. *Solution:* $t/D = 0.625/5 = 0.125$; referring to Table IV, the value of k corresponding to $t/D = 0.125$ is 0.2869. Substituting this value in $p = kS_m$,

$$p = 0.2869 \times 10{,}000 = 2{,}869 \text{ psi.}$$

Example 2.—Required the fiber stress in a steel cylinder wall when D is 10.0 in., t is 0.75 and p is 350 psi. *Solution:* $t/D = 0.75/10.0 = 0.075$; referring to Table IV, the value of k corresponding to $t/D = 0.075$ is 0.1746. Substituting this value in $S_m = p/k$

$$S_m = \frac{350}{0.1746} = 2{,}000 \text{ psi.}$$

Example 3.—Required the thickness of the pipe wall t, when D is 6.0 in., S_m is 9,000 psi, and the working fluid pressure p is 540 psi.

Solution.—The factor $k = p/S_m = 540/9{,}000 = 0.060$; referring to Table IV, the ratio of t/D corresponding to 0.060 is 0.0256. Substituting: $t = D \times 0.0256 = 6.0 \times 0.0256 = 0.1536$ in.

In the case of *external pressure* alone where $p = 0$

$$S_m = -P\frac{D^2(2 - \lambda)}{D^2 - d^2}. \tag{15}$$

The minus sign before the right-hand member of this equation denotes compressive stress. While this equation offers a rational solution for cylinders under external pressure, the formulas for collapsing pressure given on pages 35 to 37 should be used in preference, to be on the side of safety.

Division between Thick- and Thin-walled Pipe.—The dividing line between the thick-walled and thin-walled pipe depends upon the ratio of thickness to outside diameter t/D. For pipes subjected to internal pressure, formulas for thin-walled pipe are applicable to nearly all commercial steel pipe as this ratio seldom exceeds 0.10. Formulas developed by Miller[1] based on the maximum-energy-of-distortion theory show that the correction for thickness is only 2½ per cent for a pipe with a t/D ratio of 0.10. These formulas yield relatively simple expressions for the case in which the pipe wall carries the load due to internal pressure on the closed ends.

$$p = 2.31KS(1 - K), \tag{16a}$$

where p = internal pressure.

S = stress in inner surface of pipe.

$$K = \frac{\text{wall thickness}}{\text{outside diameter}} = \frac{t}{D}.$$

[1] See discussion by Benjamin Miller of paper "Theories of Strength," by A. Nádai, *Trans. ASME*, July–September, 1933, Vol. 1, No. 3, APM 55-15, p. 24.

For a thin-walled pipe, the factor $(1 - K)$ differs by a negligible amount from unity so the expression for stress in a closed-end pipe reduces to

$$S = \frac{pD}{2.31t}. \tag{16b}$$

While based on different concepts of failure or yielding, these formulas show much the same reduction in hoop stress due to the presence of an axial tension as is given by Clavarino's formula, which is based on the maximum-strain theory. By either theory the computed stress in the inner skin of a thin-walled pipe, with closed ends free to move axially, is approximately 15 per cent less than if there is no longitudinal tension acting. If there is an axial compression acting equal to the longitudinal tension due to internal pressure, then a condition equivalent to an open-ended pipe obtains and the stress at the inner surface is correspondingly increased. The condition of an open-ended pipe is covered by Barlow's formula.

Barlow's Formula.—An empirical formula for internal fluid pressure which gives results on the side of safety for all practical thickness ratios is that known as *Barlow's formula.* This formula is similar to the common formula except that the outside diameter of the cylinder is used instead of the inside. Barlow's formula is

$$S = \frac{pD}{2t}. \tag{16c}$$

While Barlow's formula is widely used because of its convenience of solution it was not generally considered to have any theoretical justification until formulas based on the maximum-energy-of-distortion theory showed that for *thin-walled* pipe with no axial tension Barlow's formula actually is theoretically correct. Since most commercially important pipe has a ratio of wall thickness to outside diameter less than 0.10, Barlow's formula for *thin-walled* tubes is of great significance. Comprehensive bursting tests[1] on commercial steel pipe have demonstrated that Barlow's formula predicts the pressure at which the pipe will rupture with an accuracy well within the limits of uniformity of commercial pipe thickness. In general, failure occurred at a pressure about 3 per cent higher than predicted from Barlow's formula.

Barlow's formula with modifications has been employed in such well-recognized standards as the ASME Boiler Construction

[1] Tests by The National Tube Co.

Code, the ASA Standard for Steel Flanges and Flanged Fittings, the ASA Standard for Wrought Iron and Wrought Steel Pipe and the ASA Code for Pressure Piping.

Distinction between Cylinders with Open and Closed Ends.—According to the foregoing review of current theory thin-walled pipe, which is defined as having a t/D ratio of less than 0.10 and thus includes nearly all commercial steel pipe, can be designed reliably for bursting strength according to Barlow's formula since this formula is on the safe side for either open-end or closed-end conditions. While the assumption of closed ends, where justified, would indicate the possibility of using 15 per cent lighter wall thicknesses in some cases for thin-walled pipes, such an assumption seldom is warranted for pipe lines although it might be for cylinders with closed ends such as tanks or receivers. The uncertain conditions imposed on thin-walled pipe lines through the presence of valves, expansion joints, bends, anchors, supports, and attachment to equipment vitiate any consistent attempt to distinguish between open- and closed-end conditions and point out the need for designing all for the more severe assumption, *viz.*, as having open ends. In the case of thick-walled pipes the difference between open-end and closed-end conditions is not so great and either Clavarino's formula or the following expression known as "Lamé's formula" for maximum hoop stress[1] at the inner surface can be used

$$S = p\left(\frac{D^2 + d^2}{D^2 - d^2}\right) \tag{16d}$$

Code Formulas for Wall Thickness.—The following formulas which originated in the American Standard Code for Pressure Piping, ASA B31.1, are considered satisfactory for general design purposes. They have been adopted also for the ASME Boiler Construction Code and the two committees cooperate in setting allowable stresses (S values)[2] for various materials operating at different metal temperatures.

$$t_{min} = \frac{pD}{2S} + C \tag{16e}$$

[1] See "Strength of Materials," by S. Timoshenko, D. Van Nostrand Company, Inc., New York, 1941, p. 239.

[2] Allowable stresses vary between different code sections in addition to being revised from time to time. Hence care should be taken to refer to the latest published S values for the code section involved. Copies can be obtained from the American Society of Mechanical Engineers, 29 West 39th St., New York 18, N.Y.

TABLE V.—ALLOWABLE STRESSES (S VALUES) FOR PIPE IN POWER AND DISTRICT-HEATING SYSTEMS[1]
(Condensed from Tables 3 and 3a of American Standard Code for Pressure Piping, ASA B31.1-1942)

Material	ASTM specification	Minimum tensile strength, psi	S values, psi, for temperatures not to exceed 150 F	250 F	350 F	450 F	550 F	650 F	750 F	850 F	950 F	1000 F
Seamless carbon steel:												
Grade A[3]	A53, A106	48,000	12,000	11,520	11,040	10,560	10,080	9,600	8,250[2]	5,850	2,600	
Grade B	A53	60,000	15,000	14,400	13,800	13,200	12,600	12,000	9,950			
Grade B, silicon-killed	A106	60,000	15,000	14,400	13,800	13,200	12,600	12,000	10,400	7,400	3,800	2,000
Grade B, Bessemer-killed	A106	60,000										
Electric-fusion welded:												
Grade A[3]	A155	45,000	10,100	9,700	9,300	8,900	8,500	8,100	7,550	5,150	2,350	
Grade B[3]	A155	50,000	11,250	10,800	10,350	9,900	9,450	9,000	8,100	5,400	2,350	
Grade C[3]	A155	55,000	12,400	11,900	11,400	10,900	10,400	9,900	8,550	5,700	2,350	
Electric-resistance welded:												
Grade A	A135	48,000	10,200	9,800	9,400	8,950	8,550	8,150	7,000			
Grade B	A135	60,000	12,750	12,250	11,700	11,200	10,700	10,200	8,450			
Lap-welded steel	A106, A53	45,000	9,400	9,000	8,550	8,150	7,750	7,300	6,250			
	A120		8,800	8,400	8,000	7,600						
Butt-welded steel	A53	45,000	6,750	6,500	6,200	5,950	5,650					
	A120		6,500	6,250	6,000	5,700						
Seamless carbon molybdenum	A206	55,000	......	......	......	......	......	11,000	11,000	10,500	8,000	5,000
Seamless 4–6 chromium—½ molybdenum	A158	60,000	......	......	......	......	......	12,000	12,000	11,200	8,000	5,850
Seamless brass:												
Muntz metal or high brass	B43	......	5,000	4,000	2,500							
Admiralty	B43	......	7,000	6,500	6,000	4,500						
Red brass	B43	......	6,000	5,500	5,000	4,500[4]						
Seamless copper pipe or tubing	B42, B75, B88	30,000 to 40,000	6,000	5,000	4,500	4,000[4]						
Cast iron:												
Centrifugally cast	WW-P-421[5]	......	......	......	......	6,000						
Pit cast	A44(ASA A21.2)	......	......	......	......	4,000						

[1] For allowable stress values for other pipe materials, reference may be made to the tables given in the Code for Pressure Piping. Approximate S values for intermediate temperatures may be obtained by interpolation, but for exact values, refer to the Code. The several types and grades of pipe tabulated above shall not be used at temperatures in excess of the maximum temperatures for which the S values are shown.

[2] A53 pipe not to be used above 750 F.

[3] Slightly higher values permitted for silicon-killed material at higher temperatures.

[4] Value shown for 406 F instead of 450 F.

[5] Federal specification.

TABLE VI.—ALLOWABLE STRESSES (S VALUES) FOR FERROUS MATERIALS[1] AT VARIOUS TEMPERATURES
(Condensed from ASME Boiler Code, Tables P5 and P7, 1943 Edition)

Material	Boiler Code designation	ASTM specification	Minimum tensile strength, psi	S values, psi, for temperatures not to exceed								
				650 F	700 F	750 F	800 F	850 F	900 F	950 F	1000 F	1100 F
Pipe:												
Seamless carbon steel, Grade *A*[2]	SA17, SA71	A53, A106	48,000	9,600	9,100	8,250	7,250	5,850	4,400	2,600	1,350	
Grade *B*[2]	SA18	A53	60,000	12,000	11,400	9,950	8,300	6,350	4,400	2,600	1,350	
Grade B, silicon-killed	SA71	A106	60,000	12,000	11,400	10,400	9,100	7,400	5,600	3,800	2,000	
Resistance welded, Grade *A*[2]	SA58	A135	48,000	8,150	7,750	7,000	6,150	4,950	3,750	2,200	1,150	
Grade *B*[2]	SA58	A135	60,000	10,200	9,700	8,450	7,050	5,400	3,750	2,200	1,150	
Carbon molybdenum	SA45	A206	55,000	11,000	11,000	11,000	10,750	10,500	10,000	8,000	5,000	
Alloy, Grade *P5a*	SA34	A158	60,000	12,000	12,000	12,000	11,800	11,200	10,000	8,000	5,850	2,200
Grade *P5c*	SA34	A158	60,000	11,000	11,000	11,000	11,000	10,850	10,000	8,000	5,850	2,200
Grade *P8a*	SA34	A158	75,000	15,000	15,000	14,600	14,300	14,000	13,400	12,300	10,000	6,000
Grade *P11*	SA34	A158	60,000	12,000	12,000	12,000	11,800	11,200	10,000	8,000	5,850	2,200
Forgings:												
Carbon, Class I[2]	SA8, SA50	A105, A181	60,000	12,000	11,400	10,400	8,300	6,350	4,400	2,600	1,350	
Class II[2]	SA8, SA50	A105, A181	70,000	14,000	13,300	11,900	8,950	6,450	4,400	2,600	1,350	
Carbon molybdenum	SA35	A182	70,000	14,000	14,000	14,000	13,500	12,000	10,200	8,000	5,000	
Castings:[4]												
Carbon, Grade *WCA*[3]	SA56	A216	60,000	12,000	11,400	10,400	9,100	7,400	5,600	3,800	2,000	
Grade *WCB*[3]	SA56	A216	70,000	14,000	13,300	11,900	10,000	7,800	5,600	3,800	2,000	
Carbon molybdenum, *WC1*[5]	SA57	A217	70,000	14,000	14,000	14,000	13,500	12,000	10,200	8,000	5,000	
Carbon molybdenum, *WC2*[5]	SA57	A217	65,000	13,000	13,000	13,000	12,500	11,500	10,000	8,000	5,000	
Bolting:[6,7]												
Carbon, heat-treated	SA261	A261	......	13,000	11,950	10,000	8,000	5,500				
Alloy, Grade *B7*	....	A193	......	16,000	16,000	16,000	16,000	13,000	10,000	6,800	3,600	
Grade *B13*	....	A193	......	16,000	16,000	16,000	16,000	13,800	11,000	8,250	5,850	2,200
Grade *B14*	....	A193	......	16,000	16,000	16,000	16,000	15,000	13,300	11,400	8,800	2,200

[1] For allowable stress values for electric-fusion-welded steel pipe, ASTM A155, lap-welded and butt-welded steel pipe, ASTM A106 and A53, see values given in Table V for temperatures 650 F and above. For other materials, see the Boiler Code. Where piping comes within the jurisdiction of the Boiler Code, values of allowable stress given in the latest revision of the Code should be used.

[2] Higher values permitted if 0.10% minimum silicon is expressly specified.

[3] These stresses permitted only if 0.10% minimum silicon is expressly specified.

[4] A casting quality factor must be assigned to these stresses, see p. 48.

[5] Apply to normalized material only.

[6] Satisfactory for average service. For long-time service, use lower values.

[7] For temperatures of −20 to 400 F, stresses equal to lower of following will be permitted: 16% of tensile strength, 20% of yield point stress.

where t_{min} = minimum pipe wall thickness, in.
p = maximum internal service pressure, psi gauge.
D = outside diameter of pipe, in.
S = allowable stress in material due to internal pressure, at the operating temperature, psi. For values of S, see Table V for power and district-heating piping, Table VI for Boiler Code S values, page 1212 for gas and air piping, page 1155 for oil piping, and page 1237 for refrigeration piping.
C = allowance for threading, mechanical strength, and/or corrosion, in.

The values of C used in formulas (16*e*) and (16*f*) shall not be less than those given in Table VII.

TABLE VII.—ALLOWANCE FOR THREADING, MECHANICAL STRENGTH, AND/OR CORROSION, PERMITTED IN DIFFERENT CODE SECTIONS

[Values of C, inches, for formulas (16*e*) and (16f)]

	Boiler Code, plus Power and District Heating Sections of ASA B31-1942	Gas and Air Section, ASA B31-1942	Oil Section, ASA B31-1942	Refrigeration Section, ASA B31-1942
Threaded steel, wrought-iron, or nonferrous pipe:				
3/8 in. and smaller	0.05	*	0.05	0.05
1/2 in. and larger	Depth of thread[2]		Depth of thread[2]	Depth of thread[2]
Grooved-steel, wrought-iron, or nonferrous pipe	Depth of groove	Depth of groove	Depth of groove	Depth of groove
Plain-end[1] steel or wrought-iron pipe or tube for 1 in. size and smaller	0.05	*	0.05	0.05
Plain-end[1] steel or wrought-iron pipe or tube for sizes above 1 in.	0.065	*	0.05	0.065
Plain-end nonferrous pipe or tube	0.00	0.00	0.00	0.00
Cast-iron pipe, pit-cast	0.18	0.18	0.18	0.18
Cast-iron pipe, centrifugally cast or cast horizontally in green sand molds	0.14	0.14	0.14	0.14

* See Chap. XV on Gas Piping, pp. 1209–1211.
[1] Plain-end pipe includes pipe joined by flared compression couplings, by lap joints, and by welding, *i.e.*, by any method that does not reduce the wall thickness of the pipe at the joint.
[2] For depth of American Standard Taper Pipe Threads, see p. 484.

In the case of *thick-walled pipes*, an alternate formula may be used under the following conditions: in the power and district-heating sections of the Code for Pressure Piping, and in the Boiler

Code, if the minimum thickness of the pipe is equal to or greater than 10 per cent of the *inside* diameter and if the nominal size is 4 in. or greater; and in the oil section of the Code for Pressure Piping, for all conditions. This formula is as follows:

$$t_{min} = \frac{D}{2}\left(1 - \sqrt{\frac{S - p}{S + p}}\right) + C \qquad (16f)$$

where the symbols have the same definitions as before. Formula (16*f*) is a modification of the well-known Lamé formula and results in lesser wall thicknesses than are obtained by the modified Barlow formula (16*e*).

For *gas and air* piping systems, the Code has somewhat different rules which are abstracted on pages 1208 to 1215.

The values of S given in Tables V and VI may be modified somewhat to provide for occasional swings in pressure and temperature which are bound to occur. The *power*, *oil*, and *district-heating* sections of the Code for Pressure Piping permit higher pressures and temperatures than the design conditions for short periods as follows:

Either pressure or temperature, or both, may exceed the nominal design values if the computed stress in the pipe wall calculated by formula (16*e*) or (16*f*) does not exceed the S value allowable for the expected temperature by more than the following allowance for the period of duration indicated:

1. Up to 15 per cent increase above the S value during 10 per cent of the operating period.
2. Up to 20 per cent increase above the S value during 1 per cent of the operating period.

Consideration is being given to substituting the following formula for determining pipe wall thickness in both the Boiler Code and the Code for Pressure Piping:

$$t_{min} = \frac{PD}{2S + 0.8P} + C \qquad (16g)$$

The symbols are defined the same as for formulas (16*e*) and (16*f*). It is proposed that this formula displace both (16*e*) and (16*f*) for all conditions. This would simplify computation of pipe wall thickness since the single formula gives results closely approximating those obtained by taking advantage of both (16*e*) and (16*f*). The proposed formula is a modification of that used in computing minimum thickness of shell plates for power boilers, Par. P-180 of the 1943 Boiler Code.

Several specific limitations in the application of these formulas are specified in the individual sections of the Code. In the *power* and *district-heating* sections, for instance, the following requirements apply:

1. In the case of cast-iron pipe, water-hammer allowances shown in Table VIII must be added to p in computing the minimum wall thickness.

TABLE VIII.—WATER-HAMMER ALLOWANCE FOR CAST-IRON PIPE

Pipe diameter, inches (sizes, inclusive)	Water-hammer allowance, psi
4 to 10	120
12 to 14	110
16 to 18	100
20	90
24	85
30	80
36	75
42 to 60	70

2. Steel or wrought-iron pipe lighter than Schedule 40 shall not be threaded. The value of p in formulas (16e) and (16f) shall not be taken less than 100 psi gauge for any conditions or material.

3. Where steel pipe is threaded and used for steam pressures of 250 psi and over, or for piping under pressure in excess of 100 psi with water temperature of 220 F and over, it shall be seamless of a quality at least equal to ASTM A106 or A53 pipe, and at least Schedule 80 weight in order to furnish added mechanical strength.

4. Threaded wrought-iron pipe or threaded brass or copper pipe shall have a wall thickness at least equal to that specified in (1) and (3) above for steel pipe of a corresponding nominal size.

5. For plain-end nonferrous pipe or tubing, the minimum wall thickness shall be as follows: For nominal sizes up to ¾ in., the nominal thickness shall not be less than specified for Type K of ASTM Specification B88; for nominal sizes ¾ in. and larger, the nominal wall thickness shall not be less than 0.049 in. Additional wall thickness shall be provided as required to care for corrosion, erosion, or mechanical strength.

For similar restrictions in other code sections, see abstracts in corresponding chapters of this handbook.

Design of Special Fittings.—Although intended primarily for computing pipe wall thickness required for resisting bursting pres-

sure, formulas (16*e*) and (16*f*) can be extended to the design of cast or forged fittings and manifolds. Since an exact stress analysis for irregular shapes is impractical,[1] the usual rule for such design is to increase the computed wall thickness in the affected section of cast fittings by 50 per cent to compensate for the weakening effect of branch outlets or irregular shapes. Rules for the reinforcement of welded branch connections will be found in the codes.

In designing *special steel castings* in accordance with the rules of the ASME Boiler Construction Code, the following quality factors shall be applied to the values of allowable stress for castings as given in Table VI:

1. A factor of 70 per cent for castings inspected only in accordance with the minimum requirements of the specification for the material.

2. A factor of 80 per cent for minimum inspection as in (1), plus[2] a thorough surface inspection and sectioning in all critical sections, or complete radiographing of one pilot casting representing every lot of 100 castings, without revealing injurious defects.

3. A factor of 90 per cent for minimum inspection plus[2] the following additional requirements:

(*a*) Thorough surface inspection, plus at least three pilot castings of the first lot of five shall be sectioned at all critical sections, or completely radiographed, without revealing injurious defects.

(*b*) One additional casting from every lot of five shall be sectioned or radiographed without showing defects.

(*c*) All other castings shall have critical sections examined using magnetic powder, or grinding and etching.

(*d*) In the case of a single casting, or any casting of a lot that has been completely radiographed without revealing defects, the 90 per cent factor may be applied.

4. Where all vital areas are exposed for inspection of the full wall thickness by machining, the inspection afforded may be taken in lieu of destructive or radiographic examination required in (3) and a casting factor of 90 per cent may be applied.

[1] See "Some Aspects of the Design of Large High-pressure Steel Valves," by F. D. Cotterman and R. E. Falls, *J. Am. Soc. Naval Engrs.*, Vol. 52, No. 3, p. 23, August, 1940.

[2] For details concerning definition of pilot castings, critical sections, radiographing, injurious defects, and identification, see Case 986, *Mech. Eng.*, December, 1942.

In applying the foregoing casting quality factors to allowable stresses for cast-steel welding-end valves and fittings, the cylindrical welding ends may be proportioned with a quality factor of 100 per cent provided these areas are finish-machined both inside and outside and carefully inspected. In no case, however, shall the thickness of the ends be less, nor more than 15 per cent greater, than that of the adjoining pipe. The machined ends are required to conform to a maximum slope line given in Fig. $P39\frac{1}{2}$ of the Boiler Code. These casting quality factors shall not be applied to the regular line of castings made according to the American Standard for Steel Pipe Flanges and Flanged Fittings, ASA B16e. Both regular and special forgings are generally assumed to have a 100 per cent quality factor.

Hollow Sphere.—The force F due to internal or external pressure which tends to force apart or push together the two halves of a sphere is F [illegible]. The stress in the wall of a thin-walled sphere is $S = \frac{pd}{4t}$, while S [illegible] $2d^3)$ is Lane's formula for any sphere.

FLOW OF FLUIDS

According to the law of conservation of energy, the sum of the potential and kinetic energies is constant at any point in a

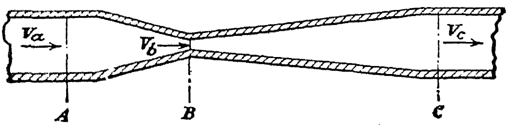


Fig. 5.—Nozzle illustrating continuity of energy.

mass of fluid under motion, provided there is no loss due to friction and no work is done by or on the fluid. This continuity of total energy can be illustrated with reference to the nozzle of Fig. 5. Letting the subscripts a, b, and c used in connection with the symbols denote conditions at sections A, B, and C in the figure, the following equation can be written for w weight units of the fluid passing through each section (the other symbols are as defined on page 6):

$$w(P.E.)_a + w(K.E.)_a = w(P.E.)_b + w(K.E.)_b = w(P.E.)_c + w(K.E.)_c, \text{ etc.} \quad (17a)$$

in which $P.E.$ denotes potential energy and $K.E.$ kinetic energy for one weight unit. In the English system, $P.E.$, $K.E.$, etc., are

expressed in foot-pounds. For one weight unit of the fluid, the equation becomes

$$(P.E.)_a + (K.E.)_a = (P.E.)_b + (K.E.)_b = (P.E.)_c + (K.E.)_c, \text{ etc.} \quad (17b)$$

but the kinetic energy for one weight unit is $K.E. = v^2/2g$, hence

$$(P.E.)_a + \frac{v_a^2}{2g} = (P.E.)_b + \frac{v_b^2}{2g} = (P.E.)_c + \frac{v_c}{2g}, \text{ etc.} \quad (17c)$$

If frictional losses are taken into account, the equation becomes

$$(P.E.)_a + \frac{v_a^2}{2g} = (P.E.)_b + \frac{v_b^2}{2g} + (P.E.)_{a-b} = (P.E.)_c + \frac{v_c^2}{2g} + (P.E.)_{a-c}, \text{ etc.} \quad (17d)$$

in which $(P.E.)_{a-b}$ represents friction loss from point A to point B, etc.

From this point on in the development of the theory it is necessary to distinguish between (1) liquids and (2) gases and vapors, *i.e.*, between fluids which are practically incompressible and those which are compressible. In the case of incompressible fluids, the potential term "$P.E.$" represents energy due to pressure or head of the fluid, while with compressible fluids a change of pressure is attended with a corresponding change in specific volume, which in turn affects the velocity and through it the kinetic-energy term. Such a change in specific volume represents adiabatic work or transfer of energy in compressing or expanding the gas or vapor. Hence, with compressible fluids it is convenient to evaluate $P.E.$ in terms of heat units used in connection with J, the mechanical equivalent of heat. In the English system J is 778, the number of foot-pounds of work equivalent to 1 Btu. To state this distinction in a different way, the potential-energy term represents static head or pressure in the case of liquids, and thermal head expressed in heat units times their mechanical equivalent in the case of gases and vapors. The detailed development for each case is given in the following sections on Flow of Liquids and Flow of Gases and Vapors.

FLOW OF LIQUIDS

Bernoulli's Theorem.—Bernoulli demonstrated in 1738 that the law of conservation of energy is applicable to the flow of liquids. His theorem may be stated as follows: *At every cross*

section of continuous and steady stream of fluid, the total energy is constant, provided no external work is done by or on the fluid.

This theorem may be represented by a formula as follows:

Total head = velocity head + pressure head + head due to elevation above datum, or

$$H = \frac{v^2}{2g} + \frac{p}{y} + h = \text{constant} \tag{18a}$$

The symbols used in this discussion of flow of liquids are defined as follows:

A = area, sq ft.
d' = inside diameter, ft.
g = acceleration of gravity = 32.2 ft per sec per sec.
H = total head, ft of liquid.
h = static head, ft of liquid.
h_λ = friction head loss, ft of liquid.
p = pressure, lb per sq ft.
F = volume, cu ft per sec.
v = velocity, ft per sec.
y = density, lb per cu ft.

Subscripts a, b, and c are used to denote sections where corresponding conditions exist.

The physical significance of Bernoulli's theorem can be better understood by reference to Fig. 6. In this case, a liquid is flowing by gravity from an elevated reservoir through a conduit to a lower elevation taken as datum. Let the subscripts a and b denote conditions at sections A and B, respectively.

Since $v_a^2/2g$ and $v_b^2/2g$ are the heights from which the same mass of liquid would have to fall in order to acquire the respective speeds of v_a and v_b, they are called "velocity heads." Likewise, (p_a/y) and (p_b/y) represent the static heads of the liquid equivalent to the pressures p_a and p_b and are therefore called "pressure heads." Finally, h_a and h_b are the respective elevations above the datum or base level and are called "gravity heads." Hence, if the sum of each velocity, pressure, and gravity head is called "total head" and denoted by H_a and H_b, respectively, the following equation exists:

$$H_b - H_a = 0.$$

Therefore, $\frac{v_a^2}{2g} + \frac{p_a}{y} + h_a = \frac{v_b^2}{2g} + \frac{p_b}{y} + h_b$ constant. This theoretical condition is never actually experienced, since friction

overcome in flowing through the conduit constitutes external work. In actual practice, this condition is taken into account by deducting the frictional loss between the cross sections under consideration. If h_λ is the head lost in friction between the points the energy relation may be equated as follows:

$$\frac{v_a^2}{2g} + \frac{p_a}{y} + h_a = \frac{v_b^2}{2g} + \frac{p_b}{y} + h_b + h_\lambda. \qquad (18b)$$

By suitably adapting the symbols to the conditions, a similar

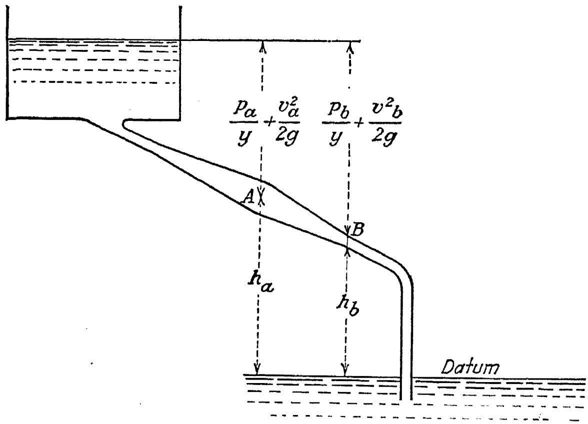


FIG. 6.—Illustration of Bernoulli's theorem.

equation can be written for the flow of any liquid through any conduit. Bernoulli's theorem, properly modified to include the effect of frictional resistances, is the basis of all practical formulas for the flow of liquids.

Equation of the Continuity of Flow.—The following equation may be written for the continuity of flow of a liquid through a conduit, by letting A represent the cross-sectional area of the conduit at the sections A, B, and C (see Figs. 5 and 6), as denoted by the subscripts a, b, and c. F is expressed in volume units of the same system in which the other terms are evaluated.

$$F = A_a v_a = A_b v_b = A_c v_c. \qquad (19)$$

This equation is modified in the case of flow through certain types

of orifices, owing to the contraction of the jet in the throat, as explained on page 55.

Entrance and Exit Losses for Liquids.—When liquid at rest in a reservoir enters a pipe, it must assume the velocity existing in the pipe. This velocity is obtained at the expense of the static head, and the decrease in static head is $h = v^2/2g$ which exists in the pipe as velocity head. The velocity head of the fluid flowing through the pipe is lost at exit, unless a suitable expanding nozzle is provided there to reconvert the velocity head back to static head. This reconversion is very effectively accomplished in the draft tube of water turbines, etc., where in many cases from 80 to 95 per cent of the velocity energy is reclaimed. Where a pipe discharges into the atmosphere or a reservoir without the equivalent of a draft tube, the entire velocity head may be considered as lost. Examples of this kind are the water boxes on condensers, feed water heaters, etc. where the velocity head in the supply pipe and that in the tubes of each pass are successively lost in the water boxes of the equipment.

If k is used to denote the fractional part of a velocity head lost at entrance and in friction or turbulence, on the downstream side, this loss may be shown through the relation $h_\lambda = kv^2/2g$. Numerical values of k can be determined with respect to entrance data such as that shown in Fig. 7 where a simple relationship exists for those cases having identical coefficients of velocity and discharge, *viz.*, where no coefficient of contraction is directly involved. Taking the *reentrant tube* of Fig. 7*c* as an example, it is evident that h must have overcome the entrance loss in addition to producing velocity v. This gives the following basis for determining k from the equation shown for Fig. 7*c* where $v = 0.72\sqrt{2gh}$, from which

$$\text{Total head, } h = 1.93\frac{v^2}{2g}$$

$$\text{Velocity head, } h_v \quad 1.00\frac{v^2}{2g}$$

$$h_\lambda = 0.93\frac{v^2}{2g}.$$

From this it is evident that for a reentrant tube $k = 0.93$, and the general expression for k is $k = \dfrac{1}{C_v^2} - 1$. Owing to the limitations of this method, it cannot be used directly for cases such as the sharp-edged orifice, Fig. 7*a*, or the Borda mouthpiece, Fig. 7*f*,

where there is contraction of the jet and C does not equal C_v. However, the contraction coefficient is never quite so small as 0.5 since k can never be greater than unity. On the strength of the foregoing, the following values have been assigned to k for the several entrances noted in Table IX*A* and the corresponding values of n computed. Velocity heads corresponding to various velocities will be found in Table IX*B*.

TABLE IX*A*.—ENTRANCE LOSSES FOR LIQUIDS

Name	Letter designation in Fig. 7	k value	n = equivalent resistance in number of diameters of straight pipe[1]
Sharp-edged orifice......		1.00	33
Streamline contour......		0.04	1.3
Reentrant tube.........		0.93	31
Venturi adjutage........		0.04	1.3
Square entrance.........		0.49	16
Borda mouthpiece.......		1.00	33
Conical diverging tube...		0.18	6
Conical converging tube.		0.18	6

[1] Computed for a friction factor $f = 0.0075$ same as Table XIV on p. 100.

TABLE IX*B*.—VELOCITY HEADS CORRESPONDING TO VARIOUS VELOCITIES

Velocity, feet per second, v	Velocity head, feet, $h = v^2/2g$	Velocity, feet per second, v	Velocity head, feet, $h = v^2/2g$
0.5	0.004	5.5	0.466
1.0	0.016	6.0	0.560
1.5	0.034	6.5	0.650
2.0	0.062	7.0	0.762
2.5	0.097	7.5	0.876
3.0	0.140	8.0	0.995
3.5	0.190	8.5	1.120
4.0	0.248	9.0	1.26
4.5	0.314	9.5	1.40
5.0	0.389	10.0	1.55

Nozzles and Orifices.—The determination of the rate of flow of a liquid through an orifice or nozzle is based on Torricelli's theorem that *the velocity of the liquid through the orifice or nozzle is the same as that which would be attained by a body falling freely in vacuo through a distance representing the effective head at the center of the orifice.* This effective head h represents the entire static head measured above that existing at the center of the orifice.

Hence, the theoretical velocity should be

$$v = \sqrt{2gh} \tag{20}$$

and the theoretical discharge in cubic feet per second is

$$F = Av = A\sqrt{2gh}. \tag{21}$$

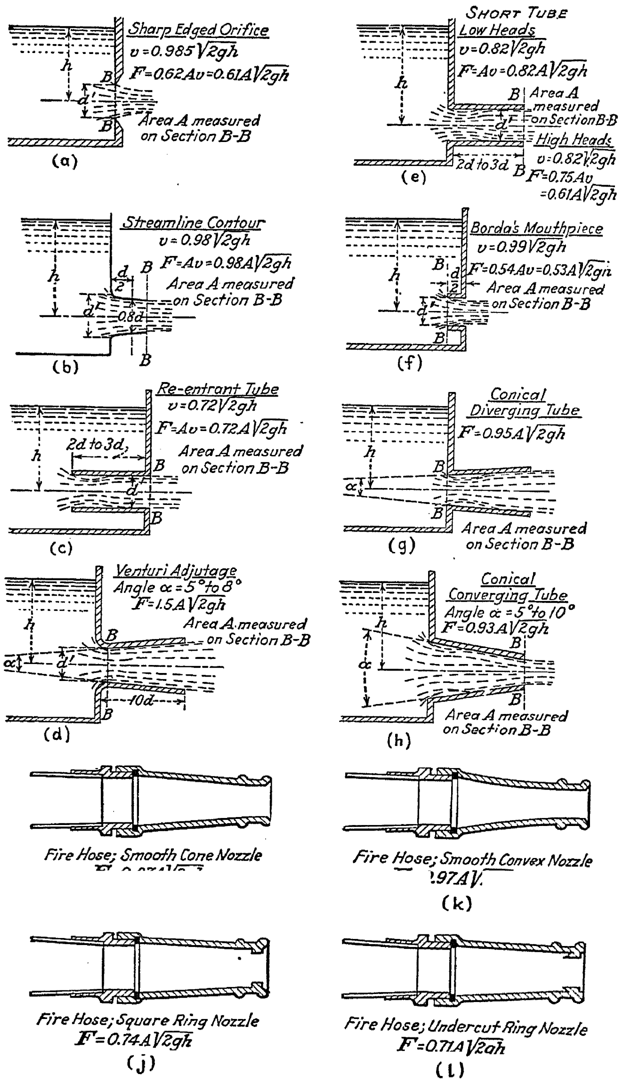
Since the orifice need not necessarily be circular, the term A is used in the general equation to denote area of the orifice or throat rather than its equivalent of $\pi d^2/4$ for circular openings. All the terms, of necessity, must be expressed in the same units of time and space.

The actual mean velocity and discharge through the orifice are somewhat less than given by the above equation, for reasons that will be stated.

The ASME Special Research Committee on Fluid Meters has proposed the following distinction between orifices and nozzles. A *nozzle* has a converging approach of sufficient length and curvature to suppress substantially all contraction. An *orifice* has so little, if any, approach curvature that contraction is fully developed or only partially suppressed. The orifice, if circular, may be further limited to an opening having a width as measured parallel to its axis and including any approach curvature that is less than one-fifth its diameter.

Coefficients of Velocity, Contraction, and Discharge.—The jet of any fluid, whether liquid, gas, or vapor, passing through an orifice usually contracts to an area smaller than the opening itself, depending on the nature of the orifice. This section of minimum area is called the "vena contracta." With liquids it is the critical section where the actual velocity approaches the theoretical given in equations (20) and (21). Figure 7 illustrates the more usual types of nozzles and orifices and shows in which types the vena contracta is most pronounced. The coefficient of contraction C_c is the ratio of the area of the jet at the vena contracta to the area of the orifice. The numerical values of C_c range from about 0.54 to unity, depending on the nature of the orifice and the fluid.

The coefficient of velocity C_v is the ratio of the actual mean velocity of the jet at the vena contracta to the theoretical velocity due to the whole head on the orifice. Its value varies from about 0.95 to 0.995. The vena contracta is the critical section at which the actual velocity most nearly approaches the theoretical corresponding to $\sqrt{2gh}$.



. 7.—Types of orifices and nozzles. Symbols are defined on page 51.

The coefficient of discharge C is the ratio of the actual discharge to the theoretical discharge for the whole area of the orifice, *i.e.*,

$$C = \frac{F}{A\sqrt{2gh}} \quad \text{or} \quad F = CA\sqrt{2gh} \tag{22}$$

C is also equal to the product of the coefficient of velocity C_v and the coefficient of contraction C_c, *i.e.*,

$$C = C_v C_c. \tag{23}$$

Approximate values of C_c, C_v, and C for other common types of orifices and nozzles are shown in connection with the conventional illustrations of Fig. 7. It will be noted that, for convenience, these orifices are shown in the side of a reservoir with the head on the orifice appearing as h, the distance from the center of the orifice to the surface of the liquid. Substantially the same values for C apply, however, if the orifices are placed between pipe flanges or on the end of a pipe discharging into the atmosphere, or for submerged discharge, provided the velocity of approach is taken into account. The head on the orifice is, in each case, the differential head or pressure between the inlet and outlet sides measured on the center line of the orifice.

Effect of Velocity of Approach.—The actual quantity discharged is still further modified by the velocity of approach if this is appreciable with respect to the velocity through the orifice or nozzle. In dealing with the effect of velocity of approach, it is necessary to distinguish between nonexpansive fluids (liquids) and expansive fluids (air, gases, and vapors). In the case of expansive fluids the specific volume changes with change in pressure through the nozzle, and the correction factor for velocity of approach is treated somewhat differently as explained on page 81.

Referring again to the types of orifices and nozzles illustrated in Fig. 7 and assuming them to be on the end of a pipe or between a pair of flanges as illustrated in Figs. 9a and b, the effect of velocity of approach for liquids may be determined as follows: For convenience in comparing this formula with those later derived for the flow of gases under similar conditions, the subscript a will be used to denote conditions in the pipe approaching the orifice, and subscripts t and b to denote conditions in the throat at Sec. B and at vena contracta, respectively.

From the fundamental flow relations of equation (19)

$$F = A_a v_a = A_t v_t = A_b v_b.$$

In the case of liquids, the velocity v_b attained in the vena contracta approximates that corresponding to the whole effective head H on the orifice, *viz.*, $H = \frac{h + v_a^2}{2g}$, where h is the difference in static head across the orifice and $v_a^2/2g$ is the head due to the velocity of the liquid approaching the orifice. Hence, if the efficiency of the orifice in converting the effective head into velocity is expressed by C_v,

$$v_b = C_v \sqrt{2gH} = C_v \sqrt{2gh + v_a^2}$$

and $$A_b = C_c A_t.$$

Substituting in the fundamental flow equation,

$$F = A_a v_a \qquad C_c A_t C_v$$

Squaring and substituting for v_a its equivalent $v_a = F/A_a$,

$$F^2 = C_c^2 C_v^2 A_t^2 \left(2gh + \frac{F^2}{A_a^2}\right).$$

Solving for F,

$$F = C_c C_v A_t \sqrt{2gh} \times \frac{1}{\sqrt{1 - (C_c C_v A_t/A_a)^2}}. \qquad (24a)$$

It will be noted that $C_c C_v = C$ and that equation (24*a*) differs from equation (22) only by the factor

$$K = \frac{1}{\sqrt{1 - (CA_t/A_a)^2}},$$

which is designated the *correction factor for velocity of approach* K.

Hence, $$F = CA_t K \sqrt{2gh}. \qquad (24b)$$

Numerical values for K corresponding to different ratios of CA_t/A_a are given in Table X.

TABLE X.—CORRECTION FACTOR K FOR VELOCITY OF APPROACH LIQUIDS

$\frac{CA_t}{A_a}$	K	$K = \sqrt{1 - (CA_t/A_a)^2}$
0.04	1.0008	$C = C_v C_c$ = product of velocity coefficients and the coefficient of contraction (see Fig. 7).
0.16	1.0130	
0.36	1.0719	
0.64	1.3014	A_t/A_a = ratio of area of orifice to area of pipe.

Sharp-edged Orifice.—In its simplest and most familiar form the *sharp-edged orifice*, otherwise known as the *orifice in a thin plate*, is

merely a circular hole in a thin flat diaphragm that is clamped between the flanges of a pipe joint so that its plane is perpendicular to the axis of the pipe and the hole is concentric with the pipe. A thicker plate is sometimes used, in which case it is chamfered around the hole on the outlet side so as to leave only a thin edge, but the inlet face of the plate remains flat with a sharp 90-deg corner at the edge of the hole.

In the case of average conditions for water, the velocity at the vena contracta is 0.98 to 0.99 of $\sqrt{v_0^2 + 2gh}$, and the coefficient of contraction around 0.62. Hence the quantity of water discharged through this orifice is

$$F = C_v C_c A_t K \sqrt{2gh} = C A_t K \sqrt{2gh}, \tag{25a}$$

or substituting approximate numerical values for C_v, C_c, and C,

$$F = 0.985 \times 0.62 A_t K \sqrt{2gh} = 0.61 A_t K \sqrt{2gh}. \tag{25b}$$

Where the diameter of a *thin*-plate orifice is less than two-tenths of the pipe diameter, the effect of velocity of approach is negligible. But, as the orifice diameter approaches eight-tenths of the pipe diameter, the quantity discharged is increased by as much as 10 per cent, and the results become discordant where the ratio exceeds 0.8.

Head Lost over Orifice.—Since the acceleration and deceleration of the fluid stream in passing through an orifice are accompanied by considerable turbulence and consequent dissipation of energy, it is to be expected that the over-all pressure loss across an orifice will be much greater than where the stream is guided as in a Venturi. The portion of the velocity head not recovered has been found by experiment to agree very nearly with the relation $1 - (A_t/A_a)$, where A_t and A_a represent areas of throat and approaching pipe, respectively. Values determined from this relation are given in Table XI in connection with loss of head over an orifice used for measurement of flow (see page 62).

Venturi Meters.—The Venturi meter is a direct application of Bernoulli's theorem to the measurement of liquids flowing in pipes under pressure. It is frequently applied to the measurement of water flow in cases where the intensity of pressure or the volume of flow renders impracticable the use of the common disk type of water meter or similar devices. Venturi meters are also applied to the measurement of flow of gases as explained in a later paragraph. A description of the Venturi meter and its measurement

of liquid flow can be given by reference to the conventional diagram of Fig. 8*a* and the following symbols which are used in conjunction with those previously defined. The upstream section at A is called the "inlet," the section at B the "throat," and that at C the "outlet." The section at C does not necessarily have to be of the same diameter as that at A, but, in order to show graphically the friction head loss h_λ, it was convenient to assume it such in this instance. Static pressures in the pipe at points A, B, and C are designated as p_a, p_b, and p_c, respectively; the internal diameter d' is measured in feet.

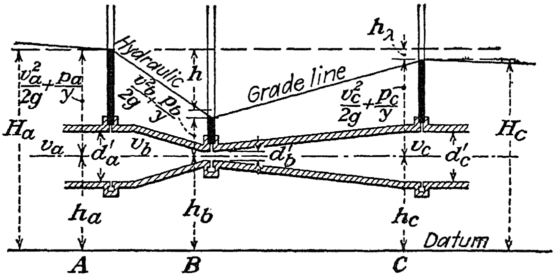


FIG. 8*a*.—Conventional diagram of Venturi meter.

From equation (19), $F = A_a v_a = A_b v_b = A_c v_c$

and $$A_a = \frac{\pi d_a'^2}{4}, \quad A_b = \frac{\pi d_b'^2}{4}, \quad A_c = \frac{\pi d_c'^2}{4}.$$

Then, $$F = \frac{\pi d_a'^2}{4} \times v_a = \frac{\pi d_b'^2}{4} \times v_b = \frac{\pi d_c'^2}{4} v_c$$

and $$v_a = \frac{d_b'^2}{d_a'^2} v_b \quad \text{or} \quad v_a^2 = \frac{d_b'^4}{d_a'^4} v_b^2.$$

From Bernoulli's theorem,

$$\frac{v_a^2}{2g} + \frac{p_a}{y} + h_a = \frac{v_b^2}{2g} + \frac{p_b}{y} + h_b + h'_\lambda = \frac{v_c^2}{2g} + \frac{p_c}{y} + h_c + h_\lambda.$$

If the axis of the meter is set level, the head due to elevation is constant and the terms h_a, h_b, and h_c can be dropped.

Therefore, $$\frac{v_a^2}{2g} + \frac{p_a}{y} = \frac{v_b^2}{2g} + \frac{p_b}{y} + h'_\lambda = \frac{v_c^2}{2g} + \frac{p_c}{y} + h_\lambda.$$

Hence, $$\frac{v_b^2 - v_a^2}{2g} = \frac{p_a - p_b}{y} - h'_\lambda.$$

Dropping h'_λ, which is insignificant, and substituting $\frac{p}{y}$ for $\frac{p_a - p_b}{y}$,

$$\frac{v_b^2 - v_a^2}{2g} = \frac{p}{y} = h.$$

Substituting the value of $v_a^2 = \frac{d_b'^4}{d_a'^4} v_b^2$,

$$v_b = \frac{d_a'^2}{\sqrt{d_a'^4 - d_b'^4}} \sqrt{2gh}.$$

$$F = \frac{\pi d_b^2}{4} \times v_b = \frac{\pi d_a'^2 d_b'^2}{4\sqrt{d_a'^4 - d_b'^4}} \sqrt{2gh}.$$

It is customary to make allowance for friction by applying a coefficient of discharge C. The value of C varies with velocity at the throat, ratio of inlet and outlet diameters, and departures from exact dimensions. Under ordinary conditions the value of C lies between 0.97 and unity. Therefore, inserting C, the value of F becomes

$$F = C\frac{\pi d_a'^2 d_b'^2}{4\sqrt{d_a'^4 - d_b'^4}} \sqrt{2gh}. \tag{26}$$

h in this case represents the difference in the static pressure head at points A and B. It frequently is convenient to measure this differential head with a mercury U tube or manometer and convert the readings into equivalent head of the liquid flowing in the pipe.

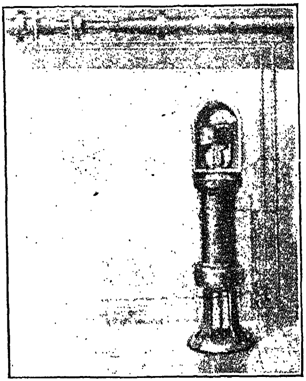


FIG. 8b.—Indicating, totalizing, and recording Venturi meter.

The use of a Venturi meter in measuring quantity of fluid flow involves very little friction loss if the slopes of the converging and diverging nozzles are properly selected. The diverging nozzle on the outlet side serves to convert the increased velocity head back again into static pressure with a very small pressure drop over the entire meter. In this respect it is superior to the flow meters described in the next paragraph. Its considerable length is a drawback in some instances, however, and the flow meter is frequently employed in preference for this reason. Venturi meters are ordinarily equipped with suitable indicating,

totalizing, and recording or integrating devices to register the amount of flow, as illustrated in Fig. 8*b*.

Flow Meters.—The flow meter is another application of Bernoulli's theorem to the measurement of fluid flow. The quantity of flow in the case of liquids is determined directly from the relations of the diameters of inlet side and throat, as in the case of the Venturi meter, and the same general equations apply.[1] The coefficient of discharge varies from about 0.60 to very nearly unity, depending on the nature of the orifice, as previously explained under "Nozzles and Orifices." For accurate work C should be determined by calibration or taken from data furnished by the manufacturer. Typical flow meters are illustrated in Figs. 9*a* and 9*b*. The diverging and converging nozzles of the Venturi meter are replaced by either an orifice plate or short converging nozzle, as shown. No special device is employed to regain the additional velocity head acquired in passing through the restricted opening and, consequently, much of this head is permanently lost.

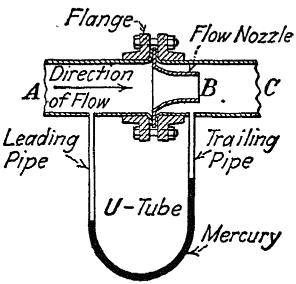


FIG. 9*a*.—Illustration of the principle of operation of flow meters employing the flow nozzle to obtain a differential pressure.

The loss of head over a thin-plate or sharp-edged orifice, such as shown in Fig. 9*b*, is found from the following relation. All heads are expressed in feet of liquid and velocities in feet per second. For definition of symbols, see page 51.

TABLE XI.—FACTOR m FOR DETERMINING OVER-ALL FRICTION LOSS ACROSS ORIFICE

d_t/d_a	A_t/A_a	m	
0.2	0.04	0.96	$m = 1 - \frac{A_t}{A_a}$
0.4	0.16	0.84	
0.6	0.36	0.64	
0.8	0.64	0.36	

[1] See "Flow Measurement by Means of Standardized Nozzles and Orifice Plates," Part 5, Chap. 4 of "Flow Measurement, 1940," ASME Power Test Codes, PTC19.5.4.

$$h_\lambda = h_a - h_c = m(h_a - h_b) = m\left(\frac{v_b^2 - v_a^2}{2g}\right). \tag{27}$$

Values of the coefficient m are given in Table XI.

The simplest device for indicating rate of flow is illustrated in a conventional way in Fig. 9*a*. A more complicated electrical device for recording rate of flow and integrating total quantities is shown in Fig. 9*b*.

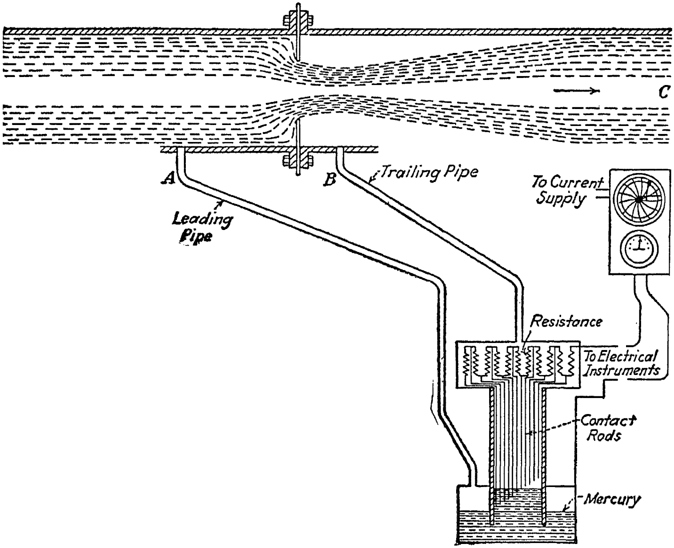


FIG. 9*b*.—Illustrating the principle of operation of flow meters employing the orifice plate to obtain a differential pressure.

Pitot Tubes.—The Pitot tube is a device for determining the velocity head of any fluid flowing through any conduit. In the case of the Pitot tube, the same general laws apply to the flow of both liquids and gases and vapors. Its use is more common in the measurement of water and air flow. From the velocity head, the corresponding velocity is readily determined from the relation $v = \sqrt{2gh}$. Figure 10*a* illustrates in a conventional way the principles used in determining the velocity head with a Pitot tube. Manometer A is connected so as to indicate the static or internal fluid head or pressure in the conduit; manometer B is connected to indicate the total head which consists of static pressure plus

velocity head; manometer C is connected to indicate directly the velocity head, which is the differential head between total head and static head. Figure 10*b* illustrates a compound Pitot tube

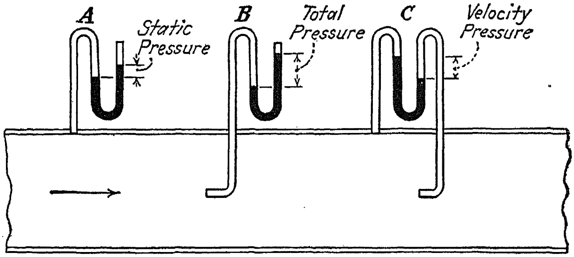


FIG. 10*a*.—Principle of the Pitot tube.

equivalent to the arrangement used with manometer C. The inclined manometer shown in Fig. 10*b* is used in measuring the flow of air or gases where the heads are too small to read accurately

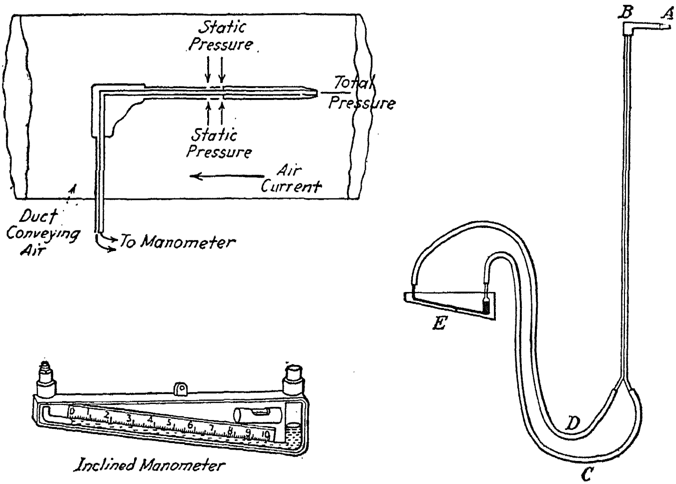


FIG. 10*b*.—Pitot tube.

with an ordinary U tube. Colored kerosene, water, mercury, or other liquid can be used in such a manometer, depending on its calibration and the heads to be measured.

In Fig. 10*b*, the tube is inserted into the conduit in such a way that its part *A-B* is parallel to the flow of fluid, with the end *A* toward the flow. The part *A-B* consists of an inner tube which transmits the total pressure to the tube *D*, and an outer jacket through which the static pressure is transmitted to the tube *C*. This outer jacket contains several small holes through which the static pressure acts. The two pressures are transmitted to the ends of the differential inclined manometer *E*, which is a U tube arranged with one leg at an angle so that the linear deflection of the liquid per unit of head is increased.

The velocity of flow is not constant at all points in the cross section of the conduit. Near the walls, flow is retarded by friction,

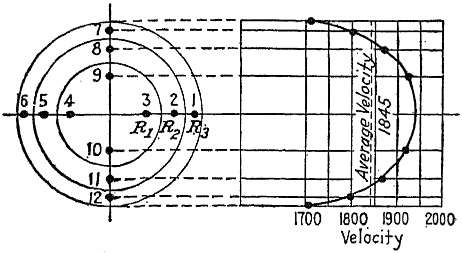


FIG. 10*c*.—Division of round pipe into annular zones for Pitot tube traverse.

and it reaches a maximum at the center. It is, therefore, necessary to measure the velocity head at several points in the cross section and calculate the velocity at each by the formula $v = \sqrt{2gh_2}$ in order to obtain an average figure. The numerical average of these velocities may then be used as the average velocity for the entire cross section. The quantity of fluid flowing can be computed readily from the average velocity and the cross-sectional area of the conduit. Since the velocity head is usually read in linear units of head h_1 of some liquid *A*, such as water or mercury, it is necessary to convert these readings into linear units of head of the fluid *B* in the conduit. This is accomplished by multiplying h_1 by a conversion factor z which is determined as shown on page 30. The relation between h_1 and h_2 then is $h_2 = zh_1$ and an equation may be written

$$v = \sqrt{2gzh_1}. \tag{28}$$

For a round conduit, the cross-sectional area should be divided into a number of annular zones of *equal area* and a traverse of the conduit made in both a horizontal and a vertical direction as

shown in Fig. 10c. For each foot of pipe diameter, the cross section should be divided into at least three of these zones. Table XII gives the distance from the center of the pipe at which each reading should be taken in percentage of pipe diameter. It is important that the velocities be computed separately and averaged, for the velocity varies as the square root of the pressure, and accurate results cannot be obtained by averaging the pressure readings. The method outlined above is essentially the same as recommended by the ASME Special Research Committee on Fluid Meters.[1]

TABLE XII.—PIPE TRAVERSE FOR PITOT TUBE READINGS (Distance from Center of Pipe to Point of Reading in Percentage of Pipe Diameter)

Number of equal areas in traverse	Number of readings	1st R_1	2d R_2	3d R_3	4th R_4	5th R_5	6th R_6	7th R_7	8th R_8
3	12	20.4	35.3	45.5					
4	16	17.7	30.5	39.4	46.6				
5	20	15.5	27.2	35.3	41.7	47.4			
6	24	14.5	25.0	32.3	38.2	43.3	47.9		
7	28	13.4	23.1	29.9	35.3	40.1	44.3	48.2	
8	32	12.5	21.6	28.0	33.2	37.6	41.5	45.1	48.4

Similar methods are used by hydraulic engineers although even with large pipes it is usually considered sufficiently accurate to make the traverse in the center circle and three or four rings only.

The following sample problem will aid in making clear the use of this formula and method. Problem: Air at atmospheric pressure and 70 F is flowing through a 10-in. inside diameter conduit. The velocity head at point 2 was observed as 0.25 in. of water at 70 F. The corresponding velocity in feet per second is determined as follows:

$$z = \frac{\text{density of water in lb per cu ft}}{(\text{density of air in lb per cu ft}) \times (\text{number of in. in 1 ft})}$$

$$z = \frac{62.4}{0.07493 \times 12} = 69.5.$$

$$v = \sqrt{2gzh_1} = \sqrt{2 \quad 32.2 \times 69.5 \times 0.25} = \sqrt{1120}$$
$$= 33.46 \text{ ft per sec,}$$

[1] "Fluid Meters, Their Theory and Application," Part 1, 4th ed., 1937, published by the American Society of Mechanical Engineers, 29 West 39th St., New York, 18, N. Y.

or, letting $V = 60v$ = velocity in feet per minute,

$$V = 33.46 \times 60 = 2{,}008 \text{ ft per min.}$$

For the purpose of this illustration assume that the average velocity for all the points 1, 2, 3, etc., has been calculated and found to be 2,500 ft per min. The quantity Q_a of air flowing per minute then is

$$Q_a = 2{,}500 \frac{\pi 10^2}{4 \times 144} = 1{,}363 \text{ cfm.}$$

For convenience in repeated calculations, equation (28) can be written in simplified form to suit the particular conditions. Suppose, for instance, that the velocity of any fluid is desired in feet per minute or per second when the velocity head readings are given in inches of water. By a solution similar to that given above,

$$v = 18.275 \sqrt{\frac{h_1}{y}} \quad \text{velocity in ft per sec}$$

$$V = 1{,}096.5 \sqrt{\frac{h_1}{y}} = \text{velocity in ft per min}$$

where y = density of the fluid in pounds per cubic foot under the conditions at which h_1 was measured.

Where extreme accuracy is not required in measuring the flow of fluids with a Pitot tube, an average velocity may be determined by applying a coefficient to the velocity measured at the center of the conduit. Various authorities give a coefficient of from 0.81 to 0.82 for air or gas flow in circular conduits, by which the *velocity head* readings taken at the center of the pipe should be multiplied to obtain the corrected average head. Consequently, the velocity based on the observed pressure readings may be multiplied by the coefficient 0.91 to obtain the corrected average velocity for air or gas, since the square root of 0.815 is 0.91. Similarly, in the case of water flow, the mean velocity may be determined, approximately, by multiplying the center velocity by 0.83 or 0.84. This ratio of average velocity to maximum velocity should be accurate within about 3 per cent plus or minus in the case of water.

The principles and construction of Pitot tubes are discussed at some length in "Fluid Meters, Their Theory and Application," Part I, *Report* of ASME Special Research Committee on Fluid Meters, 4th ed., 1937,[1] from which the following is taken:

[1] This report also contains much valuable information regarding Venturi meters, flow meters, and other types of meters, with an extensive bibliography.

The Pitot tube is suitable for use only where the conditions of flow, as indicated by the readings, are reasonably steady. Steady readings are indicative of steady flow. The question of symmetrical distribution of flow in the pipe is not as important as steady flow. Well designed bends of long radius in the pipe will affect the distribution of velocity across the diameter of the pipe for a considerable distance beyond the bend, but this is not necessarily objectionable. It is desirable to have a run of straight pipe, at least 15 pipe diameters long, following short bends, valves, and poorly designed intakes, and immediately preceding the Pitot tube to minimize the disturbances to flow which may result from such features. The greater the length of straight pipe ahead of the Pitot tube the greater is the probability that parallel stream flow will exist, which will increase the accuracy of the velocity measurement. The uniformity of the readings obtained at the various points on repeated trials is the most reliable criterion by which to judge the character of the indications.

Measurement by Volume or Weight.—The methods described above for measurement of flow in most cases involve the use of coefficients directly or indirectly derived by comparison with volumetric or weight measurements. In the case of measuring water, for instance, the first and last word in checking the accuracy of the measuring device is to take its discharge for a given time interval into a tank mounted on scales, or into a tank of known volume. Such volumetric or weight measurements are considered the most authentic methods of testing boilers, steam turbines, water pumps, and other power-plant equipment. Where weigh tanks are used in power-plant testing, it is customary to install them in pairs, so that the measurement can be made continuous by emptying one tank while the other is being filled.

FLOW OF GASES AND VAPORS

Nozzles and Orifices.—In order to avoid needless repetition in these paragraphs, the general term "nozzle" will be employed to include both nozzle and orifices, except where otherwise definitely stated, and similarly "gas" will be considered as including both gases and vapors.

In the flow of gases through nozzles, it is necessary to deal with the thermal head available to produce velocity rather than pressure head as used for liquids (see page 50). Under normal conditions, adiabatic flow obtains through nozzles, *i.e.*, no heat is added or rejected except in the form of external work; the external work in this case going to impart velocity energy to the gas itself. The effect of friction is to decrease the amount of available thermal head actually converted into kinetic energy, the portion representing friction remaining as thermal head in the gas.

Although friction heat remains as a part of the total heat of the gas yet friction represents a degradation of energy, since this total heat now exists at a lower pressure and temperature. Friction may be due either to resistance to flow offered by the sides of the nozzle, or to turbulence in the gas itself.

The efficiency of a nozzle represents the percentage of available head which is actually converted into velocity energy. Loss in efficiency may be due to friction or to poor nozzle contour. The portion of the available thermal head not converted into kinetic energy remains as latent or sensible heat in the gas, as explained above. The external work done by any particular weight unit of the gas in expansion through the nozzle is expended as thrust against the weight unit next ahead, this thrust tending to increase the velocity of the preceding unit.

The following symbols will be used throughout in connection with the flow of gases. In order to avoid possible confusion, the terms will be expressed in common English engineering units rather than general designations for heat, energy, velocity, and the like. The subscripts a, b, c, etc., will be used to designate conditions at the sections to which they refer. The subscript 1 refers to conditions just preceding the nozzle, and the subscript n to those just following.

$K.E.$ = kinetic energy, ft-lb.
$P.E.$ = potential energy, ft-lb.
w = weight of gas, lb per sec.
g = acceleration due to gravity, 32.2 ft per sec.^{2}
v_1 = velocity approaching nozzle, ft per sec.
v = velocity at any section, ft per sec.
v_n = velocity leaving nozzle, ft per sec
A = area of any section, sq ft.
a = area of any section, sq in.
V = specific volume of gas, cu ft per lb.
H = actual heat content above datum, Btu.
H_{ad} = adiabatic heat drop for frictionless flow, Btu per lb.
H' = heat content corresponding to frictionless adiabatic flow, Btu per lb.
J = mechanical equivalent of heat, 778 ft-lb per Btu.
e = nozzle efficiency expressed as a decimal fraction.
P = absolute pressure at any section, lb per sq ft.
p = absolute pressure at any section, psi.
k = adiabatic exponent, *viz.*, (specific heat at constant pressure) $\div$ (specific heat at constant volume).

Continuity of Mass.—A gas, being elastic, is assumed to fill completely the pipe or nozzle through which it is flowing. From this assumption follows the equation for continuity of mass:

$$= \frac{Av}{V} = \frac{A_a v_a}{V_a} = \frac{A_b v_b}{V_b}, \text{ etc.} \quad \text{(see Fig. 11).} \tag{29}$$

While A, v, and V may have different values at each section along the channel, w does not change.

Continuity of Energy.—From equation (17c) for continuity of energy

$$(P.E.) + \frac{v^2}{2g} = (P.E.)_a + \frac{v_a^2}{2g} = (P.E.)_b + \frac{v_b^2}{2g}, \text{ etc.}$$

but in the case of gases, it is convenient to express potential energy or thermal head as $P.E. = JH$, hence,

$$JH + \frac{v^2}{2g} = JH_a + \frac{v_a^2}{2g} = JH_b + \frac{v_b^2}{2g}, \text{ etc.} \tag{30}$$

Velocity Produced by a Given Change in Heat Content.—The change in velocity corresponding to a given change in heat content is determined for adiabatic flow from the relation $K.E. = v^2/2g$.

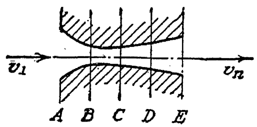


FIG. 11.—Nozzle for expansive fluids.

For the general case of flow between any two sections A and B, the conversion of thermal head into kinetic energy for 1 lb of gas, is as follows:

$$(K.E.)_b - (K.E.)_a = J(H_a - H_b) = \frac{v_b^2 - v_a^2}{2g}.$$

For the entire nozzle

$$(K.E.)_n - (K.E.)_1 = J(H_1 - H_n) = \frac{v_n^2 - v_1^2}{2g}.$$

If the velocity of approach is negligible, as for a small nozzle fed by a large pipe or reservoir, v_1 and $(K.E.)_1$ may be considered as equal to zero, and the equation becomes for any section

$$(K.E.) = J(H_1 - H) = \frac{v^2}{2g}.$$

Hence, $v = \sqrt{778 \times 2 \times 32.2(H_1 - H)} = \sqrt{50{,}071(H_1 - H)}$
$= 223.7\sqrt{H_1 - H}$. (31a)

In equation (31a), $(H_1 - H)$ represents the actual heat drop taking into account nozzle efficiency. The actual heat drop is

the product of nozzle efficiency and the adiabatic heat drop (H_{ad}) for frictionless flow, *i.e.*, $H_1 - H = e(H_1 - H') = eH_{ad}$.

Hence
$$v = 223.7\sqrt{eH_{ad}}. \tag{31b}$$

Since the effect of friction is to retain more heat as thermal head than would be the case with frictionless flow, it is necessary to provide larger areas at each successive cross section of the nozzle for a given weight of flow than would be required for the ideal condition of frictionless flow. Analysis shows that this is true for two reasons: first, the velocity produced decreases as the square root of the efficiency; and, second, the friction heat retained as thermal head tends to increase the volume of the gas over what it would be with frictionless flow. In the case of steam, the friction heat goes to increasing the quality, or to resuperheating the steam. This return of the heat of friction to the expanding fluid is spoken of as reheat. Its magnitude is defined in terms of the reheat factor explained below. In the case of multistage expansion as in a turbine, such resuperheating tends to give a cumulative efficiency for the entire series of expansions which is higher than the efficiencies of the individual stages. This is because the friction loss in a preceding stage remains as thermal head which is still available for conversion to velocity energy in the following stage.

Reheat Factor.—The reheat factor R for a series of nozzles is the ratio of the cumulative efficiency of the series to the average of the individual efficiencies of the series. It is also equal to the sum of the individual adiabatic heat drops for the series, divided by the over-all adiabatic heat drop. The reheat factor is always greater than unity, but approaches unity as the individual nozzle efficiencies approach 100 per cent, or in other words as the actual expansion line approaches the vertical line representing perfect adiabatic expansion on a total heat-entropy diagram. These relations are expressed in symbols as follows:

e_s = average of individual efficiencies of the nozzles in series.
e_c = cumulative efficiency for the entire series based on the over-all adiabatic heat drop.
H_{ad} = adiabatic heat drop in any individual nozzle, Btu per lb.
ΣH_{ad} = summation of adiabatic heat drops in all nozzles taken individually, Btu per lb.
H'_{ad} = over-all adiabatic heat drop, Btu per lb.
R = reheat factor, which is defined as

Entrance and Exit Losses for Expansive Fluids.—In the case of expansive fluids flowing from a reservoir into a pipe and out again, the entrance and exit losses are represented by the energy corresponding to the velocity head. In this case velocity is created at the expense of the internal energy of the fluid. From equation (31*b*) the adiabatic heat drop required to produce a velocity v in the pipe is $H_{ad} = v^2/50{,}071e$. The corresponding pressure drop can be read from a total heat-entropy diagram. In case an expanding nozzle is provided at the outlet of the pipe, a good share of this velocity energy can be reconverted into pressure. In case no expanding nozzle is provided at the outlet, the combined entrance and exit losses are equal to the energy required to produce the velocity in the pipe as explained above.

Contour of Nozzles for Elastic Fluids.—The chief function of a nozzle is to generate velocity rather than merely to furnish a passage from one chamber to another. It is desirable, therefore, to proportion the contour of the nozzle so as to obtain the most efficient conversion from thermal head to velocity. In general, the shape of nozzles for expansive fluids, such as gases, should be of the convergent divergent form illustrated in Fig. 11, provided the back pressure does not exceed the critical pressure at the throat. The reason for this contour is the changing relation of velocity and specific volume during expansion. The proper contour can be determined from the relations of equation (29) from which it is apparent that Av/V is constant at any section The ratio of v/V determines the proper area A at any cross section. If the values of v/V are plotted for successive steps in expansion, it will be found that the ratio first increases to a maximum corresponding to conditions at the throat of the nozzle, and then decreases from this section on. Since the value of A varies inversely as v/V, it follows that the nozzle contour should converge to a throat, and then diverge to the outlet. These relations are readily worked out for steam with reference to a steam table or Mollier diagram, or by the laws of gases (see page 142) in the case of air or other gases.

The rates of convergence and divergence of a nozzle depend on its length as well as on the theoretical area required at any section. As usually designed, nozzles are considerably modified in contour from the general shape shown in Fig. 11. The con-

vergent section may be reduced until it is little more than a well-rounded entrance without appreciably affecting the efficiency of the nozzle. The divergent section is usually a truncated cone or pyramid with its mouth tangent at the throat to the rounded entrance portion. The angle between the sides of the divergent portion should not exceed 12 deg for high efficiency. The cross sections of nozzles are usually circular, square, or rectangular as best suits the particular application. In the case of nozzles where the back pressure exceeds the critical pressure at the throat, the divergent section is not required. The relations of inlet pressure, throat pressure, and back pressure are explained in the paragraphs which immediately follow.

Practice shows that the cross section of a nozzle, whether circular, elliptical, square, or rectangular (providing the two latter have rounded corners), has very little effect on the efficiency, provided the inner surfaces are smooth and the ratio of the area at the throat to that of the mouth is correctly proportioned. The *velocity* efficiency of a properly proportioned nozzle with straight-line divergence varies from 95 to 97 per cent, corresponding to an *energy* efficiency of 92 to 94 per cent, so that usually it is not considered worth while to attempt to follow the more difficult but theoretically exact divergence curve.

For a more comprehensive description of nozzle design than space will permit here, reference may be made to "Thermodynamics," by J. E. Emswiler, "Principles of Thermodynamics," by G. A. Goodenough, "Steam Turbines," by Moyer, "Steam Turbines," by Goudie, etc.

Critical Pressure.—The section of smallest area in a nozzle is called the "throat." In the case of an orifice, the vena contracta is equivalent to the throat of a nozzle proper. For each individual gas, the throat pressure bears a certain definite and constant relation to the initial pressure, provided the back (or discharge) pressure does not exceed the throat pressure. The throat pressure which satisfies these conditions is termed the *critical* pressure, and is expressed as a certain decimal fraction of the initial pressure. For saturated steam the critical pressure is $0.578P_1$, for superheated steam 0.55 to 0.57 depending on the amount of superheat, and for air $0.528P_1$. The critical pressure P_t is determined from the formula

$$P_t = P_1 \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}. \tag{32}$$

For air $k = 1.40$. For steam initially dry and saturated $k = 1.135$. The value of k for steam depends on its initial quality. According to Zeuner the value of k for saturated steam is reduced by 0.001 for each part of moisture. The value k for superheated steam is 1.30. Table XXIII on page 166 gives values of k for several common gases and vapors. The corresponding critical pressures range from 0.487 to $0.565P_1$. (See also pages 79 and 80.)

The velocity of flow at the critical pressure is referred to as the "acoustic velocity" since it is equal to the velocity at which sound would travel in the fluid at that pressure and the corresponding specific volume (see pages 254 and 264).

For back pressures less than the critical pressure, a change in back pressure does not change in any way the throat pressure or the pressures at any other sections of the nozzle, provided the initial pressure remains constant. If the back pressure exceeds the critical pressure, the divergent portion of the nozzle is of no further use and may be dispensed with. Where the back pressure exceeds the critical pressure, the throat pressure is the same as the back pressure.

The critical pressure phenomenon may occur also in the case of fluid flow through pipes. For a discussion of this problem see "Flow in Pipes Limited by Acoustic Velocity" on page 254.

WEIGHT OF DISCHARGE THROUGH A NOZZLE

General Equation.—The weight of gas or vapor discharged through a nozzle under conditions of adiabatic flow may be calculated from the conditions existing at the throat. The fundamental equations involving these conditions are

$$v_t = 223.7\sqrt{H_1 - H_t} \text{ [from equation (31}a\text{)]},$$

$$w = \frac{A_t v_t}{V_t} = \frac{223.7 A_t}{V_t}\sqrt{H_1 - H_t}. \tag{33}$$

The above formulas and those immediately following apply to conditions of frictionless flow with no allowance for contraction of the jet. The effect of friction and contraction of the jet is taken into account by employing a coefficient of discharge of 0.97 to 0.99. Such a coefficient is included in the constants of empirical formulas. In the case of a vapor, such as saturated steam which tends to become wet during expansion, certain other allowances must be made as explained under "Weight of Saturated Steam Discharged through a Nozzle" on page 78.

It is now necessary to distinguish between two cases: (1) where the back pressure is less than or equal to the critical pressure, and (2) where the back pressure exceeds the critical pressure.

Case 1, Back Pressure Equal to or Less than Critical Pressure.—The following formula has been derived from the relations given above in equations (32) and (33) in connection with the laws of adiabatic expansion:

$$w = A_t\left(\frac{2}{k+1}\right)^{\frac{1}{k-1}}\sqrt{2g\,\frac{k}{k+1}\,\frac{P_1}{V_1}} \tag{34}$$

This equation takes into account the fact that in Case 1 the pressure at the throat bears a definite relation to the initial pressure [see equation (32)]. Numerical values of k for air and steam with varying quality and superheat are given in the preceding paragraph on "Critical Pressure." The value of k for any other gas is determined by the ratio of its specific heat at constant pressure to that at constant volume. Equation (34) can be greatly simplified in the case of perfect gases, as explained in a later paragraph on "Weight of Air Discharged through a Nozzle," by substituting numerical values for k and g, and substituting for V_1 its value from the perfect gas equation given on page 144.

Case 2, Back Pressure Greater than Critical Pressure.—In a similar manner to that described for Case 1, it can be shown that where the back pressure exceeds the critical pressure

$$w = \frac{A_t}{V_t}\sqrt{2gP_1V_1\,\frac{k}{k-1}\left[1-\left(\frac{P_n}{P_1}\right)^{\frac{k-1}{k}}\right]}. \tag{35a}$$

This equation takes into account the fact that in Case 2 the pressure at the throat is equal to the back pressure. Under the conditions of Case 2 it is sometimes more convenient to undertake a solution of the general equations (31a and 33) which readily can be accomplished by substituting for V_t and H_t in equation (33) their equivalents (in this Case 2 only) of V_n and H_n, respectively. Equation (35a) then becomes

$$w = \frac{223.7A_t}{V_n}\sqrt{H_1 - H_n}. \tag{35b}$$

If it is desired to express the throat area in square inches rather than square feet, the constant can be adjusted as follows by dividing by 144:

$$w = \frac{1.553a_t}{V}\sqrt{H_1 - H_n}. \tag{35c}$$

Weight of Air Discharged through a Nozzle.—In the case of the adiabatic flow of perfect gases where the temperature is sufficiently above the boiling point so that consideration of latent heat does not enter, the thermal head is correctly measured by the specific heat and the absolute temperature. Under these conditions a simple relation can be worked out for Case 1 where the back pressure does not exceed the critical pressure. Transposing terms in the perfect gas equation on page 144 it is evident that

$V_1 = \dfrac{RT_1}{P_1}$ for any perfect gas, and that for air $V_1 = \dfrac{53.37T_1}{P_1}$. Substituting this expression for V_1 in equation (34) and giving k its numerical value 1.4 and simplifying

$$w = 0.53\,\frac{A_tP_1}{\sqrt{T_1}} \tag{36a}$$

Since it is usually more convenient to express throat area in square inches and initial pressure in pounds per square inch absolute, the formula may be rewritten in these units without disturbing the constant, since this change of units involves 144 in both numerator and denominator. The equation then becomes

$$w = 0.53\,\frac{a_tp_1}{\sqrt{T_1}}. \tag{36b}$$

This formula is attributed to Fliegner.

In Case 2 where the *back pressure exceeds the critical pressure* the numerical value of k for any particular gas can be substituted in equation (35a). This equation also can be simplified to some extent in its application to any particular gas by substituting numerical values for k and g and substituting for V_1 and V_t in the perfect gas equation on page 144. For the case of air where $k = 1.40$ and $V = 53.37T/P$, and where $P_t = P_n$ when the back pressure exceeds the critical pressure,

$$w = 2.05A_tP_n\sqrt{\frac{1}{T_1}\left(\frac{P_1}{P_n}\right)^{0.286}\left[\left(\frac{P_1}{P_n}\right)^{0.286} - 1\right]}. \tag{37}$$

This equation also can be written without change of constant if the area is expressed in square inches and the absolute pressures in pounds per square inch.

The equivalent equation for air given below, which is attributed to Fliegner, is much easier to solve. The throat area in this case

is in square inches and the pressures in pounds per square inch absolute.

$$w = 1.06a_t \sqrt{\frac{p_n(p_1 - p_n)}{T_1}}. \tag{38}$$

Equation (38) is the equivalent of equation (37) with a discharge coefficient of 0.97 to 0.99 included.

Weight of Superheated Steam Discharged through a Nozzle.— The weight of *superheated* steam discharged through a nozzle may be calculated from equations (34) and (35a) for perfect gases by assigning a value of 1.30 to k. For Case 1, where the *back pressure does not exceed the critical pressure,* substitution in equation (34) gives

$$w = 3.786A_t \sqrt{\frac{P_1}{V_1}}. \tag{39a}$$

Or, reducing the above area to square inches and the pressure to pounds per square inch:

$$w = 0.3155a_t \sqrt{\frac{p_1}{V_1}}, \tag{39b}$$

in which formula the specific volume V_1 can be read from the corresponding p_1 in the steam tables. These formulas apply where the initial superheat is sufficient to have the steam remain somewhat superheated in the throat (see also Grashof's and Napier's and similar formulas with correction factors for superheat described under "Weight of Saturated Steam Discharged through a Nozzle.")

In Case 2 where the *back pressure exceeds the critical pressure,* a similar substitution of 1.3 for k may be made in equation (35a), as follows if V_n is substituted for V_t:

$$w = 16.74 \frac{A_t}{V_n} \sqrt{P_1 V_1 \left[1 - \left(\frac{P_n}{P_1}\right)^{0.231}\right]}. \tag{40a}$$

In case it is desired to express the area at the throat in square inches and the pressures in pounds per square inch, the constant changes as follows:

$$w = 1.395 \frac{a_t}{V_n} \sqrt{p_1 V_1 \left[1 - \left(\frac{p_n}{p_1}\right)^{0.231}\right]}. \tag{40b}$$

(See also correction factors for Case 2 used in connection with Grashof's and Napier's and similar formulas described under "Flow of Saturated Steam through a Nozzle.")

Weight of Saturated Steam Discharged through a Nozzle.—Since the weight of saturated steam discharged through a nozzle depends on the initial quality of the steam it is expedient to resort to empirical equations of which a number are available. For Case 1, *where the back pressure does not exceed the critical pressure*, the following formula which is given in Moyer's "Steam Turbines" is generally used in turbine design. x_1 represents the initial quality of the steam expressed as a decimal fraction (for instance, for steam of 98 per cent quality, $x_1 = 0.98$)

$$w = \frac{a_t p_1^{0.97}}{60\sqrt{x_1}}. \tag{41}$$

This is practically equivalent to the formula generally known as "Grashof's."

The most commonly used formula for calculating the quantity of steam discharged through a nozzle is attributed to Napier. Its simple form makes it convenient to apply under any condition of Case 1. For steam initially dry and saturated, Napier's formula is

$$w = \frac{a_t p_1}{70}. \tag{42}$$

For other conditions, either superheated or wet, it is necessary to apply a correction factor. For superheated steam, the following correction factor is commonly used in which D_1 is the number of Fahrenheit degrees superheat under initial conditions:

$$\text{Correction factor} = \frac{1}{1 + 0.00065D_1}.$$

For steam initially wet, Emswiler in "Thermodynamics" gives

$$\text{Correction factor} = \frac{1}{1 - 0.012m_1},$$

where m_1 is the number of per cent moisture under initial conditions. The correction factor $1/\sqrt{x_1}$ used in equation (41) can also be applied with slightly different results. Napier's equation for the flow of wet steam then becomes

$$w = \frac{a_t p_1}{70\sqrt{x_1}}.$$

Grashof derived the following formula for dry saturated steam which is very nearly equivalent to equation (41):

$$w = 0.01654 a_t p_1^{0.9696}. \tag{43}$$

The same correction factors given above can be applied if the initial condition is wet or superheated.

According to Goudie in "Steam Turbines," saturated steam becomes supersaturated during its expansion from initial to throat conditions, *i.e.*, no condensation from adiabatic expansion takes place, and hence the specific volume is less than if condensation does take place.

NOTE.—If condensation takes place, the latent heat liberated is converted into sensible heat and tends to increase the volume over conditions of supersaturation.

Under these conditions Goudie gives the weight discharged as

$$w = 0.0173 a_t p_1^{0.969}. \tag{44}$$

According to Stoney and Elce in *Engineering*, Dec. 2, 1921, if the theory of supersaturation is correct, the expansion of steam through a nozzle is so rapid that condensation cannot take place down to a pressure below the critical pressure. If this is the case, saturated steam acts down to the throat as a perfect gas with an adiabatic index k of 1.3. According to this theory, the discharge of saturated steam through a nozzle can be calculated for Case 1 by equation (34) using a value of 1.3 for k, which is equivalent to equations (39*a*) and (39*b*) for superheated steam. For Case 2 the same value of k can be substituted in equation (35*a*) which is equivalent to equations (40*a*) and (40*b*) for superheated steam.

According to Emswiler,

> Careful experimental investigation shows that the weight of steam delivered by a nozzle discharging against a back pressure is not quite a straight-line function of the initial pressure, thus confirming the rational equation. The curve for Napier's rule, a straight-line function, crosses the experimental curve at two points; the maximum error by Napier's rule, within the range of ordinary pressure, is slightly over 2 per cent. The maximum error by Grashof's equation is about 1 per cent.

If it is desired to use a rational rather than an empirical formula, equation (34) can be employed and a value of 1.135 assigned to k, provided the steam is initially dry and saturated. According to Zeuner, the value of k for saturated steam is reduced by 0.001 for each per cent of moisture.

Likewise for Case 2, *where the back pressure exceeds the critical pressure*, various empirical formulas are generally used because of the greater facility of their solution. For steam initially dry and saturated, Napier gave the following equation:

$$w = \frac{a_t p_n}{42}\sqrt{\frac{3(p_1 - p_n)}{2p_n}}. \tag{45}$$

In case the steam is initially wet or superheated, correction factors must be applied as described above under Case 1.

The following correction factor for application to Napier's and Grashof's and similar formulas given under Case 1 can be conveniently used to make them applicable to the conditions of Case 2. If K is the correction factor and $r = 1 - (p_n/p_1)$,

$$K = 2.182\sqrt{r(1 - 1.19r)}.$$

The following table of values of K for different ratios of p_n/p_1 furnishes a convenient means of determining this correction factor without calculation:

$\frac{p_n}{p_1}$	0.98	0.96	0.94	0.92	0.90	0.88	0.86	0.84	0.82	0.80
K	0.321	0.428	0.512	0.585	0.646	0.698	0.744	0.784	0.818	0.850
$\frac{p_n}{p_1}$	0.78	0.76	0.74	0.72	0.70	0.68	0.66	0.64	0.62	0.60
K	0.877	0.901	0.922	0.940	0.956	0.970	0.981	0.988	0.995	0.998

For steam initially dry and saturated, Grashof's formula may then be written, for Case 2:

$$w = K0.01654a_t p_1^{0.9696}.$$

This correction factor can be applied to other formulas of Case 1 in a similar way to make them applicable to the conditions of Case 2. If the steam is initially wet or superheated, the correction factors for such conditions also must be applied.

If it is desired to use a rational rather than an empirical formula, equation (35*a*) can be employed and a value of 1.135 assigned to k for steam initially dry and saturated. According to Zeuner the value of k for saturated steam is reduced by 0.001 for each per cent of moisture.

Flow of Fuel Gas through Orifices.—According to the "American Gas Handbook"[1] the following formula can be used for computing the flow of natural or manufactured gas through orifices such as those existing in intermediate and low-pressure burners, and in similar applications where the back pressure exceeds the critical pressure. The basic formula is $Q = 1{,}658.5a_tK\sqrt{h/s}$, where the symbols are as defined on pages 77 and 82 and h is the pressure drop over the orifice in inches of water column. The discharge coefficient K depends on the design of the particular orifice, with a range of 0.6 to 0.9. A value of 0.78 is a good practical

[1] Published by American Gas Journal, Inc., 53 Park Place, New York, N. Y.

average. If this figure is substituted in the above formula, it reduces to $Q = 1{,}300a_t\sqrt{h/s}$. This can be still further simplified to suit local conditions by inserting the values for specific gravity and the usual pressure that would be found at the orifice. Thus with a gas of 0.667 gravity and a normal pressure of 6 in. of water column the equation becomes $Q = 3{,}900a_t$.

Effect of Velocity of Approach on Weight of Gas Discharged through a Nozzle.—In the preceding derivations for weight of gas discharged through a nozzle, the velocity of approach has been neglected. This is permissible where the area at the throat of the nozzle is small with respect to the area of the pipe supplying it, or where the nozzle is discharging from a reservoir. If the approach to the nozzle is through a pipe whose cross-sectional area is less than 20 times the area of the throat, it becomes necessary to apply a correction factor for velocity of approach. Such a correction factor applying to liquids was derived on page 57. In the case of steam and gases, where the specific volume changes in passing through the nozzle, the above-mentioned formula does not apply with literal exactness. The error, however, is so slight for steam and gases flowing through well-rounded orifices that the correction factor for liquids may be used by dropping the coefficient of discharge C, which is otherwise compensated for in the flow formulas. The correction factor for velocity of approach then becomes

$$K' = \sqrt{\frac{1}{1 - \dfrac{A_t^2}{A_1^2}}}.$$

This correction factor K' may be applied directly to equations (34) to (45) inclusive, in the same way as described for correction factor K above. This correction factor also may be applied to sharp-edged orifices, although the results are not so uniform or satisfactory.[1]

FRICTION LOSS IN PIPES

Investigation and analysis of friction loss for fluid flow in pipes have been confined in the main to the characteristics of individual fluids. As a result, most of the formulas in general use are of an empirical form adjusted approximately to fit the characteristics of some one particular fluid under certain limiting conditions.

[1] See "Principles of Engineering Thermodynamics," by P. J. Kiefer and M. C. Stuart, John Wiley & Sons, Inc., New York, 1930, p. 240.

Examples of such empirical formulas are those attributed to Babcock, Carpenter, Fritzsche, and Unwin, or others for steam; Hazen and Williams, Saph and Schoder, Chézy, Darcy, and Fanning, or others for water; Unwin and Harris, or others for air; Spitzglass or Weymouth for gas, etc. These formulas have a certain general similarity and usually can be reduced to forms like those immediately following where the flow factor f involves an experimentally determined coefficient of friction. In some of the formulas mentioned above for steam, air, or gas, the friction factor f involves a term containing the pipe diameter, such as $[1 + (3.6/d)]$. This is of interest in connection with a rational method of determining f described in a later paragraph. In certain other formulas for friction in pipes, the investigators have tried to obtain agreement with test results by using fractional exponents for v and d, with fair success in a limited range of application. Examples of formulas involving fractional exponents are that of Fritzsche for steam and that of Saph and Schoder for water.

List of Symbols.—The following symbols are used throughout the sections on Friction Loss in Pipes and Flow of Gas and Air in Pipes.

A = inside area of pipe, sq ft.
a = inside area of pipe or orifice, sq in.
d = actual inside diameter of pipe, in.
d' = actual inside diameter of pipe, ft.
F = flow, cu ft per sec, or sec-ft.
f = friction factor, pure number, which has the same significance in all formulas and equations throughout this section. For numerical values see Fig. 15a, page 108.
G = flow, gpm.
g = acceleration due to gravity = 32.2 ft per sec^2.
h = head in feet of fluid flowing through pipe, or pressure drop in inches of water column as stated in context.
h_λ = friction head in feet of fluid flowing through pipe.
k = coefficient of resistance for bends, fittings, and valves.
L = equivalent length of pipe, ft.
l = equivalent length of pipe, miles.
M = weight of flow, lb per min.
N = Reynolds number, dimensionless.
N' = dimensionless flow function, see Fig. 15b and Table XV.
n = equivalent resistance in number of pipe diameters.

p_a = pressure, psi abs, for standard conditions approximating atmospheric pressure, usually assumed in this handbook as 14.7 psi abs.

p_0 = pressure, psi abs, for any standard conditions denoted by subscript 0.

p_1 = initial pressure in pipe. psi abs.

p_2 = final pressure in pipe, psi abs.

p_c = average flow pressure in pipe, psi abs, = $(p_1 + p_2)/2$.

p_λ = pressure drop, psi, in L linear ft of pipe.

Q_0 = flow of air or gas, cu ft per *minute* measured at any standard conditions denoted by subscript 0.

Q_a = flow of air or gas, cu ft per *minute* measured at standard conditions approximating atmospheric pressure at 60 F.

Q_c = flow of air or gas, cu ft per *minute* measured at average flow conditions corresponding to p_c and T_c.

Q_{60} = flow of air or gas, cu ft per *hour* measured at any standard conditions denoted by subscript 0.

Q = flow of air or gas, cu ft per *hour* measured at standard conditions approximating atmospheric pressure at 60 F.

R = mean hydraulic radius, ft or in. as stated.

S = specific gravity of liquid referred to water = 1 (see page 124).

s = specific gravity of gas referred to air = 1 (see page 167).

T = absolute temperature, degrees F.

T_a = absolute temperature, degrees, corresponding to standard conditions of 14.7 psi abs at 60 F = 520 F.

T_0 = absolute temperature, degrees F, for any standard conditions denoted by subscript 0.

T_c = absolute temperature, degrees F, for flow conditions in the line.

μ = absolute viscosity, lb mass per ft sec ($\mu = 0.000672z$), or lb force sec per sq ft ($\mu = 0.0000209z$), as stated.

V = specific volume, cu ft per lb, = $1/y$. Same subscripts are used as with p, Q, T, and y (sometimes used as volume instead of specific volume).

v = velocity, ft per sec.

W = weight of flow, lb per hr.

y = density of fluid, lb per cu ft (except where otherwise stated), under the conditions of pressure and temperature at which volume is measured, = $1/V$. (NOTE.—In the case of steam, y is identical with the values of $1/V$ in the

steam tables.) See distinguishing subscripts listed below for air.

y_0 = density of air, lb per cu ft, for any standard conditions denoted by subscript 0.

y_a = density of air, lb per cu ft, for standard conditions of 14.7 psi abs at 60 F.

y_c = density of air, lb per cu ft, under average pressure and temperature conditions in the line.

z = absolute viscosity in centipoises relative to water at 68 F (see also definition for μ above).

General Formulas Involving Velocity.—Several of the commonly used formulas for friction loss in pipes can be derived, as shown on the following pages, from a basic assumption that friction loss varies directly as the length of pipe and the square of the average velocity of flow and inversely as the mean hydraulic radius and the density. The term $2g$ is frequently written as a denominator under v^2, probably because of the usual conception that v^2 should appear in this form derived from the fundamental hydraulic relation of $v = \sqrt{2gh}$. The use of $2g$ in the denominator under v has no especial significance, however, since it is a constant which can be combined conveniently with certain other constants in connection with a coefficient of friction or friction factor determined by experiment. Numerous careful investigators have demonstrated that test results can be fitted to the above-described relation by resorting to either fractional exponents for v and d or else employing a friction factor which varies as some function of v and d in connection with other variables. The former case is exemplified by the work of Saph and Schoder on water, as shown on page 270, while the latter method was used by Wilson, McAdams, and Seltzer in their general solution described on pages 107 to 137. The basic relation mentioned at the beginning of this paragraph is stated in symbols as follows:

$$h_\lambda = f\frac{L}{R} \times \frac{v^2}{2g}. \tag{46}$$

Pressure drop in pounds per square inch can readily be substituted for friction loss in feet of head as follows:

The weight of a column of fluid having a cross-sectional area of 1 sq ft and a height of h_λ ft is yh_λ lb. Hence, the pressure p_λ in pounds per square inch produced by this column of fluid is $p_\lambda = yh_\lambda/144$. Hence, $h_\lambda = 144p_\lambda/y$. Substituting in the above

equation,

$$p_\lambda = \frac{f}{144} \times \frac{yL}{R} \times \frac{v^2}{2g}, \tag{47}$$

which holds true for a conduit of any cross-sectional shape. But

$$R = \frac{\text{cross-sectional area}}{\text{wetted perimeter}} = \frac{\pi d'^2/4}{\pi d'} = \frac{d'}{4}$$

for a round pipe where the diameter d' is expressed in feet. If d is expressed in inches,

$$R = \frac{\frac{\pi}{4} \times \left(\frac{d}{12}\right)^2}{\pi \frac{d}{12}} = \frac{d}{48}.$$

Substituting in equation (47),

$$p_\lambda = \frac{f}{144} \times \frac{48yL}{d} \times \frac{v^2}{64.4} = \frac{fyLv^2}{193.2d}. \tag{48a}$$

Transposing and solving for v,

$$v = 13.9 \sqrt{\frac{p_\lambda d}{fyL}}. \tag{48b}$$

Where tables are available that give *specific volume* V in cubic feet per pound instead of density y in pounds per cubic foot, this formula can be adapted readily from the relation $y = 1/V$.

$$p_\lambda = \frac{fLv^2}{193.2dV}, \tag{49a}$$

$$v = 13.9 \sqrt{\frac{p_\lambda dV}{fL}}. \tag{49b}$$

Should it be desired to use *specific gravity* S relative to water instead of density y, the following substitution can be made in the foregoing formulas. The density of water at 60 F is approximately 62.4 lb per cu ft (see Table XL on page 267), hence $y = 62.4$ and

$$p_\lambda = \frac{fS62.4Lv^2}{193.2d} = \frac{0.323fSLv^2}{d}, \tag{50a}$$

$$v = 13.9 \sqrt{\frac{p_\lambda d}{f62.4SL}} = 1.76 \sqrt{\frac{p_\lambda d}{fSL}}. \tag{50b}$$

General Formulas Involving Volume of Flow.—In applying friction-loss formulas, it frequently is convenient to solve in terms of quantity of flow rather than velocity. The general formulas

are readily converted to terms of *volume* instead of velocity in the following way. Let F represent cubic feet per second, or second feet.

$$F = Av = \frac{\pi}{4}\left(\frac{d}{12}\right)^2 v, \text{ from which}$$

$$v = \frac{4 \times 144F}{\pi d^2} = \frac{183.3F}{d^2} \quad \text{and} \quad v^2 = \frac{33{,}616F^2}{d^4}.$$

Hence, by substitution in equation (48),

$$p_\lambda = \frac{33{,}616fyLF^2}{193.2d^5} = \frac{174fyLF^2}{d^5}, \tag{51a}$$

$$F = 0.0757\sqrt{\frac{p_\lambda d^5}{fyL}}. \tag{51b}$$

Where it is desired to solve in terms of *specific volume* V instead of density y, the relation $y = 1/V$ can be substituted.

$$p_\lambda = \frac{174fLF^2}{Vd^5}, \tag{52a}$$

$$F = 0.0757\sqrt{\frac{p_\lambda Vd^5}{fL}}. \tag{52b}$$

In the case of *liquids* where it is desired to use *specific gravity* relative to water instead of density, 62.4 S can be substituted for y as before.

$$p_\lambda = \frac{174f62.4SLF^2}{d^5} = \frac{10{,}860fSLF^2}{d^5}, \tag{53a}$$

$$F = 0.0757\sqrt{\frac{p_\lambda d^5}{f62.4SL}} = 0.00958\sqrt{\frac{p_\lambda d^5}{fSL}}. \tag{53b}$$

Similarly, in case it is desired to solve in terms of *U. S. gallons per minute* G, where there are 7.48 gal in 1 cu ft,

$$G = 60 \times 7.48Av = 60 \times 7.48 \times \frac{\pi}{4} \times \left(\frac{d}{12}\right)^2 v = 2.448d^2v$$

$$\text{and} \quad v^2 = \frac{G^2}{5.992d^4}.$$

Substituting in equation (48),

$$p_\lambda = \frac{fyL}{193.2d} \times \frac{G^2}{5.992d^4} = \frac{fyLG^2}{1{,}159d^5}. \tag{54a}$$

Transposing and solving for G,

$$G = 34.1\sqrt{\frac{p_\lambda d^5}{fyL}}. \tag{54b}$$

In case it is desired to use *specific gravity* relative to water instead of density, 62.4 S can be substituted for y as before.

$$p_\lambda = \frac{f62.4SLG^2}{1{,}159d^5} = \frac{fSLG^2}{18.6d^5}, \tag{55a}$$

$$G = 34.1\sqrt{\frac{p_\lambda d^5}{f62.4SL}} = 4.31\sqrt{\frac{p_\lambda d^5}{fSL}}. \tag{55b}$$

General Formulas Involving Weight of Flow.—For cases where it is convenient to solve in terms of weight of flow in pounds per hour W and density in pounds per cubic foot y, the following substitutions can be made:

$$W = 3{,}600Ayv = 3{,}600 \times \frac{\pi}{4}\left(\frac{d}{12}\right)^2 yv = 19.64d^2yv.$$

Solving for v, $v = \dfrac{W}{19.64d^2y}$ and $v^2 = \dfrac{W^2}{385.5d^4y^2}$.

Substituting in equation (48),

$$p_\lambda = \frac{fyLW^2}{193.2 \times 385.5d^5y^2} = \frac{fLW^2}{74{,}500yd^5}, \tag{56a}$$

$$W = 273\sqrt{\frac{p_\lambda yd^5}{fL}}. \tag{56b}$$

In the case of air, gas, or steam, should it be desired to solve in terms of specific volume V in cubic feet per pound, $y = 1/V$ can be substituted in equation (56) as follows:

$$p_\lambda = \frac{fLVW^2}{74{,}500d^5}, \tag{57a}$$

$$W = 273\sqrt{\frac{p_\lambda d^5}{fLV}}. \tag{57b}$$

See also pages 165 to 198 for further developments of the formula for use with gas and air, and pages 246 to 254 for use with steam.

Relative Carrying Capacity of Pipes.—Frequently it is desirable to know what number of pipes of one size will be equal in carrying capacity to a larger pipe of the same length. (In order to determine the relative carrying capacity of pipes of different lengths, it is necessary to employ the methods described in succeeding paragraphs under "Complex Pipe Lines.") This relation can be determined readily from the pressure-drop formulas for volume or weight of flow in the preceding paragraphs as follows: Using $p_\lambda = fLW^2/74{,}500yd^5$ and letting N represent the number of pipes

of inside diameter d_2 which have a combined carrying capacity equal to one pipe of inside diameter d_1:

$$p_\lambda = f\frac{LW^2}{74{,}500yd_1^5} = f\frac{L\left(\frac{W}{N}\right)^2}{74{,}500yd_2^5}.$$

Since the pressure drop p_λ is the same in both cases and the weight carried per pipe of diameter d_2 is W/N, if the weight carried in the pipe of diameter d_1 is W, we may cancel the numerical constant and identical terms f, L, y and solve for N:

$$N^2 = \frac{d_1^5}{d_2^5} \text{ and } N = \left(\frac{d_1}{d_2}\right)^{5/2}.$$

This relation obtained from the general formula may be stated as follows: For *turbulent flow the carrying capacities of pipes vary as the square root of the fifth power of their inside diameters.* A similar relation can be worked out for the *Unwin-Carpenter-Babcock type of equation* given on page 247.

In this case $$N = \frac{d_1^3\sqrt{d_2 + 3.6}}{d_2^3\sqrt{d_1 + 3.6}}.$$

The two equations above give almost identical results for small values of N. For large values of N (around 200), the results obtained by the Unwin relation may exceed those of the general relation by 50 per cent. Values of N obtained from the general relation are most frequently used in hydraulic work, while those from the Unwin-Carpenter-Babcock relation are commonly used for steam, air, and gas. Table XIII, which is based on the Unwin-Carpenter-Babcock relation, gives the relative carrying capacities of standard-weight wrought pipes and pipes of stated inside diameters under conditions of turbulent flow. Table XIII is sufficiently accurate for low values of N in most cases with any fluid, whether water, steam, air, or gas. Where large values of N are involved or considerable exactness is required, the method indicated above will serve as a guide for making similar solutions from some empirical formula which more closely fits the particular flow conditions. For the case of *viscous or straight-line flow* described on page 105, *the relative carrying capacities vary as the fourth power of the inside diameter.* The relative carrying capacities of water pipes with turbulent flow are shown in Table XLVIII on page 286.

TABLE XIII.—RELATIVE CARRYING CAPACITY OF STANDARD-WEIGHT (SCHEDULE 40)[1] WROUGHT PIPE AND ACTUAL INSIDE DIAMETERS (For air, steam, and gas)

Diameter	½	¾	1	1½	2	2½	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	Diameter
½	......	2.27	4.88	15.8	31.7	52.9	96.9	205	377	620	918	1,292	1,767	2,488	3,014	3,786	4,904	5,927	7,321	8,535	9,717	½
¾	2.60	......	2.05	6.97	14.0	23.3	42.5	90.4	166	273	405	569	779	1,096	1,328	1,668	2,161	2,615	3,226	3,761	4,282	¾
1	7.55	2.90	......	3.45	6.82	11.4	20.9	44.1	81.1	133	198	278	380	536	649	815	1,070	1,263	1,576	1,837	2,092	1
1½	24.2	9.30	3.20	......	2.00	3.34	6.13	13.0	23.8	39.2	58.1	81.7	112	157	190	239	310	375	463	539	614	1½
2	54.8	21.0	7.25	2.26	......	1.67	3.06	6.47	11.9	19.6	29.0	40.8	55.8	78.5	95.1	119	155	187	231	269	307	2
2½	102	39.4	13.6	4.23	1.87	......	1.83	3.87	7.12	11.7	17.4	24.4	33.4	47.0	56.9	71.5	92.6	112	138	161	184	2½
3	170	65.4	22.6	7.03	3.11	1.66	......	2.12	3.89	6.39	9.48	13.3	20.9	23.7	31.2	39.1	50.6	61.1	75.5	88.0	100	3
4	376	144	49.8	15.5	6.87	3.67	2.21	....	1.84	3.02	4.48	6.30	8.61	12.1	14.7	18.5	23.9	28.9	35.7	41.6	47.4	4
5	686	263	90.9	28.3	12.5	6.70	4.03	1.83	....	1.65	2.44	3.43	4.69	6.60	8.00	10.0	13.0	15.7	19.4	22.6	25.8	5
6	1,116	429	148	46.0	20.4	10.9	6.56	2.97	1.63	....	1.48	2.09	2.85	4.02	4.86	6.11	7.91	9.56	11.8	13.8	15.6	6
7	1,707	656	226	70.5	31.2	16.6	10.0	4.54	2.49	1.51	....	1.41	1.93	2.71	3.28	4.12	5.34	6.45	7.97	9.31	10.6	7
8	2,435	936	322	101	44.5	23.8	14.3	6.48	3.54	2.18	1.43	.....	1.35	1.93	2.33	2.92	3.79	4.57	5.67	6.60	7.52	8
9	3,335	1,281	440	137	60.8	32.5	19.5	8.85	4.85	2.98	1.95	1.37	.....	1.41	1.71	2.14	2.77	3.35	4.14	4.83	5.50	9
10	4,393	1,688	582	181	80.4	42.9	25.8	11.7	6.40	3.93	2.57	1.80	1.32	.....	1.21	1.52	1.97	2.38	2.94	3.43	3.91	10
11	5,642	2,168	747	233	103	55.1	33.1	15.0	8.22	5.05	3.31	2.32	1.70	1.28	.....	1.26	1.63	1.88	2.43	2.83	3.22	11
12	7,087	2,723	938	293	129	69.2	41.6	18.8	10.3	6.34	4.15	2.91	2.13	1.61	1.26	.....	1.30	1.57	1.93	2.26	2.58	12
13	8,657	3,326	1,146	358	158	84.5	50.7	23.0	12.6	7.75	5.07	3.56	2.60	1.98	1.53	1.22	.....	1.21	1.49	1.74	1.98	13
14	10,600	4,070	1,403	438	193	103	62.2	28.2	15.4	9.48	6.21	4.35	3.18	2.41	1.88	1.50	1.22	.....	1.24	1.44	1.64	14
15	12,824	4,927	1,698	530	234	125	75.3	34.1	18.7	11.5	7.52	5.27	3.85	2.92	2.27	1.81	1.48	1.21	.....	1.17	1.35	15
16	14,978	5,758	1,984	619	274	146	88.0	39.9	21.8	13.4	8.78	6.15	4.51	3.41	2.66	2.12	1.73	1.42	1.18	.....	1.14	16
17	17,537	6,738	2,322	724	320	171	103	46.6	25.6	15.7	10.3	7.20	5.27	3.99	3.11	2.47	2.03	1.66	1.37	1.17	.....	
18	20,327	7,810	2,691	840	371	198	119	54.1	29.6	18.2	11.9	8.35	6.11	4.63	3.60	2.87	2.35	1.92	1.59	1.36	1.16	
20	26,676	10,249	3,532	1,102	487	260	157	70.9	38.9	23.9	15.6	10.9	8.02	6.07	4.73	3.76	3.08	2.52	2.08	1.78	1.52	
24	42,624	16,376	5,644	1,761	778	416	250	113	62.1	38.2	25.0	17.5	12.8	9.70	7.55	6.01	4.92	4.02	3.32	2.84	2.43	
30	75,453	28,990	9,990	3,117	1,378	736	443	201	110	67.6	44.2	31.0	22.7	17.2	13.4	10.7	8.72	7.14	5.88	5.03	4.30	
36	120,100	46,143	15,902	4,961	2,193	1,172	705	319	175	108	70.4	49.3	36.1	27.3	21.3	16.9	13.9	11.3	9.37	8.01	6.85	
42	177,724	68,282	23,531	7,341	3,245	1,734	1,044	473	259	159	104	73.0	53.4	40.5	31.5	25.1	20.5	16.8	13.9	11.9	10.1	
48	249,351	95,818	33,020	10,301	4,554	2,434	1,465	663	363	223	146	102	75.0	56.8	44.2	35.2	28.8	23.5	19.4	16.6	14.2	
Diameter	½	¾	1	1½	2	2½	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	

The upper portion above the diagonal lines of blanks pertains to nominal pipe sizes while the lower portion is for pipe of the actual internal diameter given.

Equivalent number of pipes = $(d_1^3/d_2^3)\sqrt{(d_2 + 3.6)/(d_1 + 3.6)}$. *Ex.* 1.—Carrying capacity of ¾-in. standard-weight pipe is equivalent to 2.27 half-inch standard-weight pipes. *Ex.* 2.—Carrying capacity of pipe with inside diameter of 13 in. is equivalent to 3.56 pipes with inside diameter of 8 in.

[1] Standard-weight pipe 10 in. and smaller is identical with Schedule 40 pipe.

The data in this table are reproduced by permission from "Steam Power Plant Engineering," by G. F. Gebhardt, 4th ed., John Wiley & Sons, Inc., New York.

Complex Pipe Lines.—The formulas given above for flow in simple pipe lines can be extended to apply to divided circuits, series circuits consisting of different size pipes, combinations of divided and series circuits, and distribution networks. The following solutions for complex pipe lines are based on the general flow formula $p\lambda = fLW^2/74{,}500yd^5$, etc. In order to simplify the calculations, it often is convenient to neglect the fact that the friction factor f may vary with the pipe diameter, and assume that f is the same for all pipe diameters. This is not strictly correct, but it avoids involving the diameter in a complex term, such as is used in the Unwin-Carpenter-Babcock type of formula, and greatly facilitates the solution. The general method here indicated can be employed readily with any of the fractional exponent type of empirical formulas commonly applied to various fluids. The assumption that carrying capacity varies as the fifth power of the diameter used in the present solution is not strictly accurate in some instances. In the case of water at ordinary atmospheric temperatures, the fifth power relation gives quite accurate results, as shown by Saph and Schoder's[1] tests which indicated a fractional exponent for d of 4.97. In the case of natural gas, Weymouth[2] arrives at a value of $5\frac{1}{3}$ as the exponent of d. The majority of empirical equations for all fluids, however, use exponents for d very closely approximating 5. Fritzsche in his formula for the flow of steam gives 4.97.

Cross[3] has analyzed the flow problem in distribution networks for water-supply systems and has indicated how to extend his methods to computing the flow of air, gas, or steam in networks and to the design of electrical circuits. Simplifications and short cuts for this method have been evolved by others.[4] See also (*a*)

[1] "An Experimental Study of the Flow of Water in Pipes," *Trans. ASCE Paper* 964, Vol. 51, p. 253, 1903.

[2] "Problems in Natural Gas Engineering," *Trans. ASME*, Vol. 34, p. 185, 1912; see pp. 187,197, "Piping Handbook." For another solution of complex pipe lines based on the Weymouth formula see pp. 45–52 of "Flow of Natural Gas through High-pressure Transmission Lines," by T. W. Johnson and W. B. Berwald, *Monograph* 6, Bureau of Mines. Reprints may be obtained from the American Gas Association, 420 Lexington Ave., New York, N.Y.

[3] "Analysis of Flow in Networks of Conduits or Conductors," by Prof. Hardy Cross, *Univ. Illinois, Bull.* 286, November, 1936, Engineering Experiment Station, Urbana, Ill.

[4] (*a*) "The Hardy Cross Method, Its Practical Application in Determining Flow and Heads in Pipe Systems," by D. R. Taylor, *Water Works and Sewerage*, Vol. 90, No. 3, March and April, 1943. Contains numerous aids to solution and

Distribution Network or Street Mains, page 1191 in Chap. XV on "Gas Piping," and (*b*) "Solution of Special Problems in Pipe Flow (Water Distribution Systems) by Graphical Analysis," by Grant K. Palsgrove.[1]

Complex-series Pipe Line.—A complex-series pipe line consists of two or more runs of different size pipe in series. The problem in this case is to determine the length of some one size of pipe having a frictional resistance equivalent to the series. The first step is to select the size pipe in which the equivalent length is to be expressed, and the next to convert each run in turn into the corresponding length of the size pipe selected. This is readily done from the general friction-loss formula, since the pressure drop, quantity of discharge, and density are the same by definition for both actual and equivalent pipes. The subscripts 1, 2, 3, etc., are used to designate the different sections of pipe in series, with subscript 1 referring to the size pipe selected to which the others are to be converted. Small letters l and d are used to denote actual lengths and diameters, while capital letters L and D are used for equivalent lengths and diameters. By definition $l_1 = L_1$, and $d_1 = D_1 = D_2 = D_3$, etc. For each successive section considered, the following relations hold:

From the general flow formula $p_\lambda = \dfrac{f_2 l_2 W^2}{74{,}500 y d_2^5}$,

and $$p_\lambda = \frac{f_1 L_2 W^2}{74{,}500 y D_2^5}.$$

Dividing one equation by the other, $1 = \dfrac{f_2}{f_1}\dfrac{l_2 D_2^5}{L_2 d_2^5}$,

or $$L_2 = l_2 \frac{f_2}{f_1}\left(\frac{D_2}{d_2}\right)^5,$$

but $$D_2 = d_1;$$

hence, $$L_2 = l_2 \frac{f_2}{f_1}\left(\frac{d_1}{d_2}\right)^5;$$

and, similarly, $$L_3 = l_3 \frac{f_3}{f_1}\left(\frac{d_1}{d_3}\right)^5, \text{ etc.}$$

---

a bibliography of references to the Hardy Cross method.

(*b*) *Eng. News-Record*, Oct. 1, 1936, and Mar. 3, 1938, and *Civil Civil Eng.*, May, 1938.

(*c*) "Computation of Flows in Distribution Systems," by Weston Gavett, *Jour. AWWA*, Vol. 35, No. 3, pp. 267–287, March, 1943.

[1] *Rensselaer Polytechnic*, *Bull.* 37, August, 1932.

The total equivalent length L of the series of connected pipes is $L = L_1 + L_2 + L_3 \cdots + L_n$ where L_1, L_2, L_3, etc., are the equivalent lengths reduced to diameter d_1 for the respective sections that make up the series.

Example.—A series pipe line shown in Fig. 12*a* consists of three runs of 6-in., 8-in., and 10-in. standard-weight (Schedule 40) wrought pipe af 300-, 400-, and 500-ft lengths, respectively. What is the equivalent length of 6-in. pipe assuming the same friction factor for each, *i.e.*, $f_1 = f_2 = f_3$?

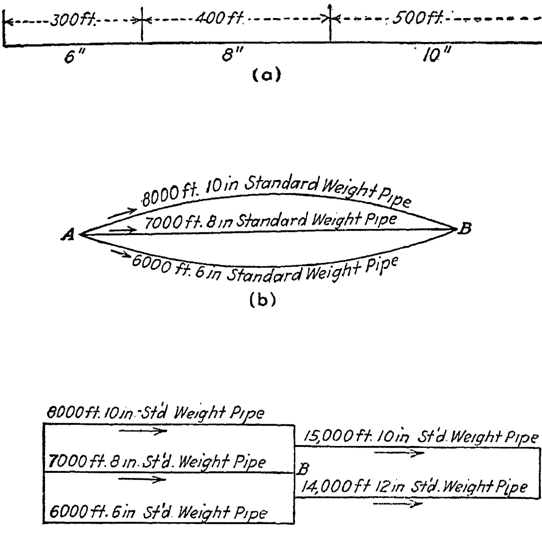


FIG. 12.—*a*, complex-series pipe line; *b*, divided-circuit pipe system; *c*, complex-series loop pipe system.

From Table X, page 366, the fifth powers of the inside diameters of 6-, 8-, and 10-in. standard-weight pipe are 8,206, 32,380, and 101,000, respectively.

The equivalent length in 6-in. pipe of section (1) is, of course, 300 ft.

The equivalent length of section (2) is $L_2 = 400 \left(\frac{8,206}{32,380}\right) = 101$ ft.

The equivalent length of section (3) is $L_3 = 500 \left(\frac{8,206}{101,000}\right) = 41$ ft.

Total equivalent length of 6-in. pipe = 442 ft.

The answer is that 442 ft of 6-in. standard-weight pipe have a frictional resistance equal to that of the complex-series pipe line shown in Fig. 12*a*.

Divided Circuit or Loop.—In the divided circuit shown in Fig. 12*b*, the solution is made in two steps: (1) each branch is reduced to an equivalent diameter D_n corresponding to a common length which is taken to agree with the length of that branch whose diameter and friction factor are made the basis for expressing equivalent lengths; (2) substituting the equivalent diameters and their corresponding common length in the flow formula, the flow through each branch can be computed and the total taken for all branches.

Step (2) above can be simplified from the following considerations. The condition existing in a loop system is that the initial and final pressures are the same for each pipe, and the only difference between the computations for flow is the $\sqrt{D_n^5}$. A summation of all the quantities $\sqrt{D_n^5}$ can be made, therefore, and the summation used in the flow formula. This makes it possible to obtain the answer with a single solution of the flow formula.

A solution for step (1) is derived as follows from the general flow formula. For any one branch of the loop system, W, p_λ and y are the same for both actual and equivalent conditions. The subscripts 1, 2, 3, etc., are used to designate the different branch pipes of the loop with subscript 1 referring to the size pipe selected to which the others are to be converted. Small letters l and d are used to denote actual lengths and diameters, while capital letters L and D are used for equivalent lengths and diameters. By definition $d_1 = D_1$ and $l_1 = L_1 = L_2 = L_3$, etc. Writing the flow equations for one of the branches in terms of both actual and equivalent dimensions, and dividing one equation by the other:

$$p_\lambda = \frac{f_2 l_2 W_2^2}{74{,}500 y d_2^5} \quad \text{and} \quad p_\lambda = \frac{f_1 L_2 W_2^2}{74{,}500 y D_2^5}$$

the quotient is

$$1 = \frac{f_2 l_2 D_2^5}{f_1 L_2 d_2^5}$$

or

$$D_2 = d_2 \left(\frac{f_1 L_2}{f_2 l_2}\right)^{1/5}.$$

But

$$L_2 = l_1,$$

hence,

$$D_2 = d_2 \left(\frac{f_1 l_1}{f_2 l_2}\right)^{1/5},$$

and similarly

$$D_3 = d_3 \left(\frac{f_1 l_1}{f_3 l_3}\right)^{1/5}, \text{ etc.}$$

This completes the derivation required for step (1).

The two methods of accomplishing step (2) mentioned above are executed as follows: The pressure drop, density, and friction

factor for each branch of the loop system are identical. Assuming a common length l_1 for all branches, their corresponding equivalent diameters have just been computed. It is now possible to write the following series of equations (based on the general flow equation) for flow in the different branches:

$$W_1 = \sqrt{\frac{74{,}500 p_\lambda y d_1^5}{f_1 l_1}};\ W_2 = \sqrt{\frac{74{,}500 p_\lambda y D_2^5}{f_1 l_1}};$$

$$W_3 = \sqrt{\frac{74{,}500 p_\lambda y D_3^5}{f_1 l_1}}, \text{ etc.}$$

The total flow W through the loop is the summation of the flow through the separate branches, or $W = W_1 + W_2 + W_3 \cdots + W_n$. An examination of the above equations shows, however, that the right-hand members are identical with the exception of the terms $d_1^{5/2}$, $D_2^{5/2}$, $D_3^{5/2}$, etc. The summation may then be written

$$W = \left(\frac{74{,}500 p_\lambda y}{f_1 l_1}\right)^{1/2} (d_1^{5/2} + D_2^{5/2} + D_3^{5/2} \cdots + D_n^{5/2}),$$

from which it is apparent that the equivalent diameter for the entire loop reduced to an equivalent length l_1 is

$$d = (d_1^{5/2} + D_2^{5/2} + D_3^{5/2} \cdots + D_n^{5/2})^{2/5}.$$

Example.—One million cubic feet of natural gas per hour, measured at standard pressure and temperature (see page 158), are to be transmitted through a loop system consisting of three pipe lines of about equal smoothness (see Fig. 12*b*). The pressure of the gas at the delivery end (B) of the system is to be 50 psi abs. The specific gravity of the gas is 0.65 and the temperature in the pipe is 60 F. Determine the required initial pressure of the gas at A. Although the effect of friction factor is of only minor importance in this case, it is carried through for the purpose of illustration.

The actual inside diameters of 6-, 8-, and 10-in. standard-weight steel pipe are 6.065, 7.981, and 10.02 in., respectively (see Table I, page 357). The friction factors used below were computed from Fig. 15*b* and Table XV (see pages 111 to 118) by assuming an approximate inlet pressure of 70 psi abs. Using 8,000 ft as the equivalent length for each pipe,

$$D_1 = d_1 = 10.02.$$

$$D_2 = 7.981 \left(\frac{\text{[illegible]}}{0.00395} \times \frac{\text{[illegible]}}{7{,}000}\right)^{\text{[illegible]}} = 8.19.$$

$$D_3 = 6.065 \left(\frac{0.00393}{0.00410} \times \frac{8{,}000}{6{,}000}\right)^{1/5} = 6.37.$$

$$\begin{aligned} D_1^{5/2} &= 10.02^{5/2} = 317.9 \\ D_2^{5/2} &= 8.19^{5/2} = 191.9 \\ D_3^{5/2} &= 6.37^{5/2} = \underline{102.4} \\ \Sigma D^{5/2} &= 612.2 \\ D &= 13.02 \text{ in.} \end{aligned}$$

Or, in other words, the resistance of the entire loop is equivalent to that of a single pipe having an inside diameter of 13.02 in. and a length of 8,000 ft.

The balance of the answer can be obtained readily through substitution in equation (65c) where $Q = 182\sqrt{\frac{(p_1^2 - p_2^2)d^5}{fsl}}$ and solving as follows: The friction factor f is determined from Figs. 15a and 17a through the relation $Qs/dz = (1{,}000{,}000 \times 0.65)/(13.02 \times 0.0105) = 4{,}750{,}000$, from which $f = 0.0040$. Substituting and solving for p_1

$$1{,}000{,}000 = 182\sqrt{\frac{(p_1^2 - 50^2)13.02^5}{0.0040 \times 0.65 \times 8{,}000}}.$$

$$p_1 = 64.62.$$

Answer.—The required initial pressure at A is 64.62 or in round numbers 65 psi abs.

Series-loop System.—A piping system made up of two sections or more of loop systems in series, as shown in Fig. 12c, is known as a "series-loop system." A large variety of such systems exist. The general method of solution is as follows: (1) first determine a single equivalent line for each loop as explained above under Divided Circuits; (2) next determine a single equivalent line for the series of single equivalent lines, following the method explained under Complex-series Pipe Lines. The explanation under the above headings, supplemented by the illustrations given there, is sufficient for handling problems of this sort. An example illustrating an actual solution for a series-loop system is given in connection with natural gas on page 197. In the example, the carrying capacity is assumed to vary as the $\frac{8}{3}$ power of the diameter instead of the $\frac{5}{2}$ power. As was previously mentioned, the use of the $\frac{8}{3}$ power of d in connection with natural gas was proposed by Thomas R. Weymouth to take into account the variation in friction factor with diameter.

Equivalent Resistance of Bends, Fittings, and Valves.—According to the published results of investigators in this field[1] the

[1] The following references have been selected as covering various phases of the subject:

(a) "Friction of Water in Pipes and Fittings," by F. E. Giesecke, *Univ. Texas, Bull.* 1759, Oct. 20, 1917.

(b) "Resistance of Fittings to Flow through Pipe Lines," by Dean Foster, *Trans. ASME*, Vol. 42, pp. 647–669, 1920. Also *Mech. Eng.*, November, 1920.

(c) "Experiments on Loss of Head in Valves and Pipes of One-half to Twelve Inches Diameter," by C. I. Corp and R. O. Ruble, 1922, *Univ. Wisconsin, Bull.*, Engineering Series, Vol. IX, No. 1.

(d) "The Friction of Water in Elbows," by F. E. Giesecke, C. P. Reming, and J. W. Knudson, Jr., *Univ. Texas, Bull.*, 2712, Mar. 22, 1927.

(e) "Frictional Resistance and Flexibility of Seamless-tube Fittings Used in

pressure drop through bends, fittings, and valves varies according to the same functions of velocity of flow and pipe diameter that are used in the corresponding pressure-drop formula for straight pipe. Owing to this convenient similarity, it is possible to express the resistance of fittings and valves as equivalent to so many feet of straight pipe and to compute the pressure drop for the whole line as one unit.

With turbulent flow (see Fig. 15*a*) all investigators agree that the loss through bends, valves, and fittings, like that through straight pipe, varies with a power of the velocity, approximating the square, and inversely as a function of the inside pipe diameter, approximating the first power. Adjusting the exponents of v and d was resorted to by various investigators to compensate for varia-

---

Pipe Welding," by Sabin Crocker and Arthur McCutchan, *Trans. ASME*, 1931, FSP-53-17.

(*f*) "The Loss of Head in Cast Iron Tees," by F. E. Giesecke, W. H. Badgett, and J. R. D. Eddy, *Bull.* 41, Texas Engineering Experiment Station, May 1, 1932.

(*g*) "Loss in 90-degree Pipe Bends of Constant Circular Cross Section," by Albert Hoffman, *Trans. Munich Hydraulic Institute, Bull.* 3, translated and published by ASME, 1935.

(*h*) "Loss of Energy in Miter Bends," by Hans Kirchbach, *ibid.*

(*i*) "Loss in Oblique-angled Pipe Branches," by Franz Petermann, *ibid.*

(*j*) "Energy Loss in Smooth- and Rough-surfaced (Miter) Bends and Curves in Pipe Lines," by Werner Schubart, *ibid.*

(*k*) "Pressure Losses for Fluid Flow in 90° Pipe Bends," by K. Hilding Beij, Bureau of Standards, *Research Paper*, RP 1110, *Jour. Research Nat. Bur. Standards*, Vol. 21, July, 1938.

(*l*) "A Comparative Study of Friction Heads in Screwed and Welded Elbows," by F. E. Giesecke and J. S. Hopper, Journal Section ASHVE, *Heating, Piping and Air Conditioning*, January, 1942, pp. 56–60.

(*m*) "Flow of Fluids through Valves, Fittings, and Pipe," Crane Company, *Tech. Paper* 409, May, 1942.

(*n*) Report of tests in catalogues of Taylor Forge and Pipe Works.

(*o*) "Experiments at Detroit, Michigan, on the Effect of Curvature upon the Flow of Water in Pipes," by Gardiner S. Williams, Clarence W. Hubbell, and George H. Fenkell, *Trans. ASCE*, Vol. 47, pp. 1–369, 1902.

(*p*) "Flow of Fluids in Curved Passages," by J. Eustice, *Engineering* (London), Nov. 13, 1925, pp. 604–605. See also *Eng. and Contr.*, Vol. LXV, Apr. 14, 1926 (Water Works Issue).

(*q*) "Pressure Loss in Elbows and Duct Branches," by Andrew Vazsonyi, *Trans. ASME*, Vol. 66, No. 3, pp. 177–183, April, 1944.

(*r*) "Pressure Losses in Marine Fuel Oil Systems," by Cyrus Beck, *Jour. Am. Soc. Naval Engrs.*, Vol. 56, pp. 62–83, February, 1944; pp. 235–271, May; pp. 366–395, August. This series of three articles relates Reynolds numbers to *laminar-flow* pressure losses in straight pipe, bends, and fittings as determined in tests made by the Bureau of Ships, U.S. Navy.

tions in the friction factor f with viscosity, density, velocity, and pipe diameter which variations were not well understood at one time. In the light of subsequent knowledge of the relation of the friction factor to Reynolds numbers, as used in the rational formula (see page 107), it seems better to keep v^2/d in the formula and make these adjustments through Reynolds numbers and f, thus maintaining a universal relation applicable to all fluids and conditions. Although empirical formulas with adjusted exponents do well enough in some cases for the limited field in which they are applicable, they cannot be used as a basis for methods that are intended for extension to other fluids and conditions.

The advantages of having a definite and universally applicable coefficient of resistance for each type of bend, fitting, or valve have prompted the adoption of the following simplified, but reasonably accurate, method, which facilitates computing any resistance as being equivalent to so many feet of straight pipe. The discrepancies involved in this simplified method probably are of a lesser order than variations due to indeterminates such as degree of roughness of the pipe interior or exact geometrical similitude of shape.

In deriving the expression for such a coefficient of resistance k, it is convenient to start from the concept of hydraulics in which losses are expressed as so many velocity heads or some fraction of a velocity head, as the case may be, and write (see page 82 for definition of symbols):

$$h_\lambda = k \times \frac{v^2}{2g}$$

where k is the whole or fractional number of velocity heads lost in going through a valve or fitting. Also, from equation (46) (see page 84) for friction loss in straight pipe

$$h_\lambda = \frac{fL}{R} \times \frac{v^2}{2g} = \frac{4fL}{d'} \times \frac{v^2}{2g},$$

since for a round pipe the hydraulic radius $R = d'/4$ ft. Equating the two expressions for h_λ to one another gives

$$k \times \frac{v^2}{2g} = \frac{4fL}{d'} \times \frac{v^2}{2g}, \quad \text{from which} \quad k = 4f \times \frac{L}{d'}.$$

The expression L/d', where L and d' both are expressed in feet, is well known as the *equivalent resistance in pipe diameters* and for

convenience will be designated as n to denote the *number* of pipe diameters in length which is equivalent to some particular bend, fitting, or valve. Hence,

$$k = 4fn \quad \text{and} \quad n = \frac{k}{4f}.$$

The relation between f, k, and n for standard 90-deg elbows will be apparent on reference to Fig. 15*a*, page 108, where the n values for curve E read from the scale at the left correspond to f values read from the scale at the right for some intermediate condition of roughness between curves B and C. For instance, for a Reynolds number of 100,000, f equals, say, 0.0075 for an intermediate condition of roughness. The coefficient of resistance k for a standard 90-deg screwed elbow is taken as 0.9 from references (*a*) and (*m*) of the footnote on page 95. From this n is computed to be $n = k/4f = 0.9/4 \times 0.0075 = 30$. This checks well enough with the value of $n = 28$ read from curve E of Fig. 15*a*. Had the value of $f = 0.0056$ for a Reynolds number of 100,000 been read directly from curve B ("clean" steel or cast iron) of Fig 15*a*, the corresponding value of k could be computed $k = 4 \times 0.0056 \times 28 = 0.63$, which agrees better with Foster's value of 0.67 for the coefficient of resistance for standard elbows (see footnote (*b*), page 95).

Referring again to Fig. 15*a* it is evident that the n values for curve E, as well as the f values for curve C or intermediate smoothness values, do not vary a great deal throughout the turbulent flow region. Obviously, from the foregoing equations the k values, which are interrelated with f and n, must remain constant to at least the same degree. That this is true is evidenced by references (*g*), (*h*), and (*j*) listed in the footnote on page 96, which show k to be very nearly constant for Reynolds numbers above 50,000. Hence it can be concluded that, for most practical purposes, the k value for any given type of fitting can be taken as constant throughout the turbulent flow region irrespective of pipe size, flow velocity, or nature of the fluid.

According to Crane Company's *Tech. Paper* 409 [see reference (*m*) on page 96] the coefficient k is practically constant for any one type of valve, and for small globe valves it is almost independent of the nominal size. In general, the higher the resistance of the valve or fitting, the more nearly independent of size is the resistance coefficient. Two objects are said to be geometrically

similar if all the linear dimensions of one are in a constant ratio to the corresponding dimensions of the other and if all angles of the one are equal to the corresponding angles of the other. Strictly, the surfaces must be similar also, *viz.*, they must have the same relative roughness.

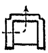
In the light of the foregoing discussion and on the basis of published test results (see references listed in footnote on page 95), the authors have exercised their best judgment in assigning k and n values to various types of bends, fittings, and valves and have computed the results set down in Table XIV. Although these assignments are deemed to give approximations satisfactory enough for most purposes, they are by no means final and it is expected that the values will be revised from time to time as more information becomes available.

The n values used in Table XIV have been computed with an assumed friction factor f of 0.0075, which corresponds (see Fig. 15a) to a Reynolds number of 100,000, and an intermediate condition of roughness between curves B and C. Where greater accuracy is desired in computing equivalent lengths for any particular job, a specific n value can be computed with reference to f for the corresponding Reynolds number and condition of pipe interior as read from Fig. 15a, using the relation $n = k/4f$.

The *equivalent lengths in feet* shown in Table XIV have been computed on a basis that the inside diameter corresponds to that of Schedule 40 (standard-weight) steel pipe which, again, is close enough for most purposes involving other schedules of pipe. Where a more specific solution for equivalent length is desired, this may be made by multiplying the actual inside diameter of the pipe in inches by $n/12$, or the actual inside diameter in feet by n, as read from the table heading. Likewise the equivalent length values can be used with reasonable accuracy for copper or brass fittings and bends. In the case of valves, however, the equivalent length in feet of copper or brass pipe should be taken as 40 to 50 per cent longer than for steel pipe, owing to the lesser resistance per foot of copper or brass pipe. See also values assigned by Giesecke as given on page 968 and 969.

Should the *friction loss* across the bend, fitting, or valve be desired instead of the equivalent length, this may be computed from the formula $h_\lambda = kv^2/2g$ which gives the head lost in feet of fluid conveyed through the pipe. If pressure drop is desired in pounds per square inch, $h_\lambda = 144p_\lambda/y$ can be substituted in the

TABLE XIV.—EQUIVALENT RESISTANCE OF BENDS, FITTINGS,

		Screwed fittings[2]				90° welding elbows and smooth bends[3]					
		45° ell	90° ell	180° close return bends	Tee	$R/d = 1$	$R/d = 1\frac{1}{2}$	$R/d = 2$	$R/d = 4$	$R/d = 6$	$R/d = 8$
k factor =		0.42	0.90	2.00	1.80	0.48	0.36	0.27	0.21	0.27	0.36
L/d' ratio[5] n =		14	30	67	60	16	12	9	7	9	12
Nominal pipe size, in.	Inside diam. d, in., Sched. 40[7]										
		L = equivalent length in feet of Schedule 40 (standard weight) straight pipe[7]									
½	0.622	0.73	1.55	3.47	3.10	0.83	0.62	0.47	0.36	0.47	0.62
¾	0.824	0.96	2.06	4.60	4.12	1.10	0.82	0.62	0.48	0.62	0.82
1	1.049	1.22	2.62	5.82	5.24	1.40	1.05	0.79	0.61	0.79	1.05
1¼	1.380	1.61	3.45	7.66	6.90	1.84	1.38	1.03	0.81	1.03	1.38
1½	1.610	1.88	4.02	8.95	8.04	2.14	1.61	1.21	0.94	1.21	1.61
2	2.067	2.41	5.17	11.5	10.3	2.76	2.07	1.55	1.21	1.55	2.07
2½	2.469	2.88	6.16	13.7	12.3	3.29	2.47	1.85	1.44	1.85	2.47
3	3.068	3.58	7.67	17.1	15.3	4.09	3.07	2.30	1.79	2.30	3.07
4	4.026	4.70	10.1	22.4	20.2	5.37	4.03	3.02	2.35	3.02	4.03
5	5.047	5.88	12.6	28.0	25.2	6.72	5.05	3.78	2.94	3.78	5.05
6	6.065	7.07	15.2	33.8	30.4	8.09	6.07	4.55	3.54	4.55	6.07
8	7.981	9.31	20.0	44.6	40.0	10.6	7.98	5.98	4.65	5.98	7.98
10	10.02	11.7	25.0	55.7	50.0	13.3	10.0	7.51	5.85	7.51	10.0
12	11.94	13.9	29.8	66.3	59.6	15.9	11.9	8.95	6.96	8.95	11.9
14	13.13	15.3	32.8	73.0	65.6	17.5	13.1	9.85	7.65	9.85	13.1
16	15.00	17.5	37.5	83.5	75.0	20.0	15.0	11.2	8.75	11.2	15.0
18	16.88	19.7	42.1	93.8	84.2	22.5	16.9	12.7	9.85	12.7	16.9
20	18.81	22.0	47.0	105	94.0	25.1	18.8	14.1	11.0	14.1	18.8
24	22.63	26.4	56.6	126	113	30.2	22.6	17.0	13.2	17.0	22.6

[1] Compiled from various sources, (see footnote, p. 95) and adjusted according to the best equivalents. Values for welding fittings are for conditions where bore is not obstructed by weld values given by Giesecke on pp. 968, 969.

[2] Flanged fittings have three-fourths the resistance of screwed elbows and tees (see reference

[3] Tabular figures give the extra resistance due to curvature alone, to which should be added

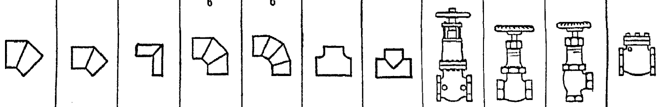
[4] Small size socket-welding fittings are equivalent to miter elbows and miter tees.

[5] Equivalent resistance in number of diameters of straight pipe computed for a value of f =

[6] For condition of minimum resistance where the center-line length of each miter is between

[7] For pipe having other inside diameters, the equivalent resistance may be computed from

AND VALVES, LENGTH OF STRAIGHT PIPE IN FEET[1]

Miter elbows[4] (No. of miters)					Welding tees		Valves (screwed, flanged, or welded)			
1-45°	1-60°	1-90°	2-90°	3-90°	Forged	Miter[4]	Gate	Globe	Angle	Swing check
0.45	0.90	1.80	0.60	0.45	1.35	1.80	0.21	10	5.0	2.5
15	30	60	20	15	45	60	7	333	167	83
			[6]	[6]						
L = equivalent length in feet of Schedule 40 (standard weight) straight pipe[7]										
0.78	1.55	3.10	1.04	0.78	2.33	3.10	0.36	17.3	8.65	4.32
1.03	2.06	4.12	1.37	1.03	3.09	4.12	0.48	22.9	11.4	5.72
1.31	2.62	5.24	1.75	1.31	3.93	5.24	0.61	29.1	14.6	7.27
1.72	3.45	6.90	2.30	1.72	5.17	6.90	0.81	38.3	19.1	9.58
2.01	4.02	8.04	2.68	2.01	6.04	8.04	0.94	44.7	22.4	11.2
2.58	5.17	10.3	3.45	2.58	7.75	10.3	1.21	57.4	28.7	14.4
3.08	6.16	12.3	4.11	3.08	9.25	12.3	1.44	68.5	34.3	17.1
3.84	7.67	15.3	5.11	3.84	11.5	15.3	1.79	85.2	42.6	21.3
5.04	10.1	20.2	6.71	5.04	15.1	20.2	2.35	112	56.0	28.0
6.30	12.6	25.2	8.40	6.30	18.9	25.2	2.94	140	70.0	35.0
7.58	15.2	30.4	10.1	7.58	22.8	30.4	3.54	168	84.1	42.1
9.97	20.0	40.0	13.3	9.97	29.9	40.0	4.65	222	111	55.5
12.5	25.0	50.0	16.7	12.5	37.6	50.0	5.85	278	139	69.5
14.9	29.8	59.6	19.9	14.9	44.8	59.6	6.96	332	166	83.0
16.4	32.8	65.6	21.9	16.4	49.2	65.6	7.65	364	182	91.0
18.8	37.5	75.0	25.0	18.8	56.2	75.0	8.75	417	208	104
21.1	42.1	84.2	28.1	21.1	63.2	84.2	9.85	469	234	117
23.5	47.0	94.0	31.4	23.5	70.6	94.0	11.0	522	261	131
28.3	56.6	113	37.8	28.3	85.0	113	13.2	629	314	157

judgment of the authors. See text for application to other pipe schedules and for copper or brass spatter or backing rings; if appreciably obstructed, use values for "Screwed Fittings." See also

(*m*) of footnote on p. 96)].
the full length of travel. For corrugated and creased bends, see item 5, p. 133.

0.0075 from the relation $n = k/4f$, see text.
d and $2\frac{1}{2}d$, see footnote *j*, p. 96.
the above n values.

former equation to get

$$p_\lambda = \frac{ky}{144} \times \frac{v^2}{2g}.$$

Effect of Curvature on Resistance of Bends.—A misconception has been more or less prevalent to the effect that all short-radius bends and fittings necessarily cause greater pressure drop than do long-radius bends. Published results show that the least over-all resistance is produced by bends having a radius of 2 to 4 pipe diameters,[1] and that elbows or bends having a radius of 1½ to $2d$ do not compare unfavorably with those having a radius of 4 to $6d$ or greater. The agreement of nearly all observers that a curvature between 2 and $4d$ gives a minimum over-all pressure drop is explained in the references by the supposition that too short turns on the one extreme give excessive pressure drops, whereas on the other extreme of easy curvature the disturbance persists over a greater length of travel.[2]

Losses Due to Sudden Enlargement or Contraction.—As with the equivalent resistance of bends, fittings, and valves, it is convenient to express the losses due to sudden enlargement or contraction as a fractional part of a velocity head or as the equivalent of so many diameters of straight pipe. The relations involved are similar to those for entrance and exit losses (see pages 53 and 72). In the case of a sudden enlargement the loss cannot exceed the head required to produce the given change in velocity, while with a sudden contraction the loss usually is somewhat less owing to a lesser amount of turbulence. This is in keeping with the well-known fact that the energy losses accompanying a decrease in velocity generally are greater than those associated with an increase. According to Merriman,[3] however, when the ratio of d_2/d_1 is higher than 0.77 the loss due to sudden contraction is greater than that due to sudden expansion.

Referring to Fig. 13, the loss produced by a given change in velocity $(v_1 - v_2)$ caused by a sudden *enlargement* from diameter d_1 to diameter d_2 may be expressed as

$$h_\lambda = \frac{(v_1 - v_2)^2}{2g}$$

[1] See references (*e*), (*k*), (*m*), (*o*), and (*p*) listed in footnote on p. 95.

[2] See references (*o*) and (*p*) listed in footnote on p. 95, discussed also in the closure to *e*.

[3] "Treatise on Hydraulics," by Mansfield Merriman, John Wiley & Sons, Inc., New York, 1925.

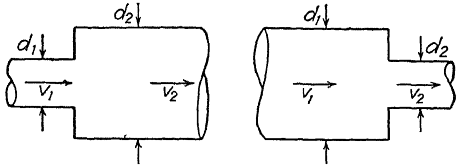
which also may be written

$$h_\lambda = \left(1 - \frac{d_1^2}{d_2^2}\right)^2 \frac{v_1^2}{2g} = k\frac{v_1^2}{2g},$$

from which the losses due to a *sudden enlargement* are

d_1/d_2	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
k factor	0.98	0.92	0.83	0.71	0.56	0.41	0.28	0.13	0.04
L/d ratio n	33	31	28	24	19	14	9.3	4.3	1.3

Considerable difference of opinion exists as to the magnitude of the losses occasioned by *sudden contractions*. Nearly every hydraulics text gives a different set of values. Brightmore[1] con-



(a)-Sudden Enlargement (b)-Sudden contraction

FIG. 13.—Loss due to sudden enlargement or contraction.

cluded that the loss due to a sudden contraction is 0.7 of the loss from a sudden enlargement as given in the preceding paragraph. This is conceded by some to give satisfactory results where velocities are low, say below what would correspond to a loss of 1 ft of head in hydraulics. For higher velocities a more complicated expression seems to be required for k. The following expression has been used by Merriman and others for the loss of head in *sudden contractions:*

$$h_\lambda = \left(\frac{1}{C_c} - 1\right)^2 \frac{v_2^2}{2g} = k\frac{v_2^2}{2g}$$

where $$C_c = 0.582 + \frac{0.0418}{1.1 - d_2/d_1}.$$

The following values for k and n for *sudden contractions* seem a satisfactory compromise:

d_2/d_1	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
k factor	0.46	0.45	0.42	0.40	0.36	0.28	0.19	0.10	0.04
L/d ratio n	15	15	14	13	12	9.3	6.3	3.3	1.3

[1] A. W. Brightmore, *Proc. Inst. Civil Engrs.*, Vol. 169, p. 323.

Force Exerted by Pressure Drop and by Change in Direction of Flow.—Certain open-ended pipes such, for instance, as the safety-valve vent piping illustrated in Fig. 12 on page 891 tend to pull out of slip joints or other loose connections owing solely to the push exerted by pressure drop through the line. Under these circumstances some form of anchor or other restraint is needed to prevent disengagement at the joint. A method of computing the force involved is indicated in connection with Fig. 12.

Where there is a change in direction of flow, as in a pipe bend or fitting, a corresponding change in momentum is involved which must be balanced by a reaction in the piping and supports. Except for rare cases of low-pressure lines operating at high velocities, the stresses and reactions are relatively small compared with those set up by fluid pressure and usually can be ignored with safety. Those having occasion to solve such problems can refer to the section on "Anchors and Braces", pages 1053 to 1056, which also gives the pressure thrust due to change of direction. Longitudinal stress due to pressure is not included there. For this, see page 32.

General Comments on Flow Formulas.—As pointed out in a preceding paragraph the retention of $2g$ as a separate entity in flow formulas has no significance aside from certain applications where it is used in converting elevation head to velocity or for expressing losses in terms of velocity head. Hence, in most friction-loss equations $2g$ can be combined conveniently into the numerical constant that embraces any required adjustment between units of time and dimension. On the other hand the friction factor f as properly conceived (see Rational Solution, pages 107 to 137) is not a constant but varies as a dimensionless function of other properties of the pipe and fluid conveyed. This being the case, it is advantageous to keep f as a pure friction factor uncombined with formula constants which differ only between formulas. With this end in view and starting with this edition of the "Piping Handbook," the formulas throughout this chapter have been rearranged so that f always denotes the same thing irrespective of the formula in which it appears, and any adjustment required between formulas is incorporated in separate numerical constants.

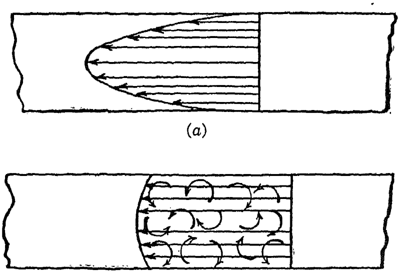
The effect of roughness of the pipe interior and the distinction between viscous and turbulent flow also enter into the value of f. In addition to the rational method for determining f described on

pages 107 to 137 which can be used in connection with any of the foregoing formulas, various empirical modifications are given later in this chapter in separate sections devoted to these fluids. The foregoing general derivations are grouped together for convenience in reference, to avoid repetition, and to demonstrate the relation between the rational formula and the various empirical formulas commonly applied to specific fluids under limited conditions. Attention also is called to the fact that, whereas the units of time and dimension differ in some cases between similar formulas, this merely affects the numerical value of the formula constant, while f continues unchanged and the general forms of the equations remain comparable.

Where empirical equations have been applied to specific fluids, the variations of friction factor with velocity, pipe diameter, and roughness were correlated with test results for certain limited conditions. Since velocity depends on pipe diameter, the effect of both on friction factor can be approximated in a single expression in terms of d as was done in the Unwin and Spitzglass formulas for air, gas, and steam (see pages 175 to 186) or the Darcy formula for water (see page 269). Better results can be obtained through making f a function of both diameter and velocity through employing fractional exponents for v and d as was done in the Fritzsche formulas for air, gas, and steam (see pages 177 to 180), or in the hydraulic formulas of William-Hazen, Saph-Schoder, etc. The variations of density and viscosity with temperature usually are ignored in empirical formulas, although they should not be except possibly for flow at atmospheric temperature. Recent investigations of the flow of oils in which viscosity plays a particularly important part have led to the development of the rational solution (see pages 107 to 137) which is applicable to all fluids and conditions and is fast supplanting the old empirical formulas except in very limited applications.

Viscous Flow.—Another phase of the problem which has been more thoroughly investigated in connection with the flow of oils is the distinction between viscous or straight-line flow at low velocities and turbulent flow at higher velocities. These two conditions of flow are illustrated in Fig. 14. The equations given above apply to turbulent flow, which is the usual condition for gases and nonviscous liquids, such as water. If a liquid tends to adhere to the pipe walls, then any forced motion of the liquid through the pipe is resisted by this adhesive grip and by internal

resistance due to cohesion between adjacent layers of the liquid. Flow under these conditions is a sort of telescopic sliding of one layer along another, with the central core moving the fastest and the outermost layers moving hardly at all. Viscous flow persists until a certain definite critical velocity is exceeded, when flow becomes turbulent. For viscous flow the loss of head along the pipe varies directly as the length, the velocity, and the viscosity,



(a) (b)

FIG. 14.—Distinction between (a) viscous and (b) turbulent flow.

and inversely as the cross-sectional area of the pipe. The roughness of the pipe has little or no effect on true viscous flow. Poiseuille's formula which is generally used for viscous flow is stated as follows (see pages 82 to 84 for definition of symbols):

$$p_\lambda = \frac{0.000668zLv}{d^2} \quad \text{or} \quad p_\lambda = \frac{0.1225zLF}{d^4},$$

$$\text{or} \quad p_\lambda = \frac{0.00003402zLW}{yd^4} \quad \text{or} \quad p_\lambda = \frac{0.00003402zLVW}{d^4},$$

$$\text{or} \quad p_\lambda = \frac{0.000273zLG}{d^4} \quad \text{or} \quad p_\lambda = \frac{0.00003402zLQ}{d^4}.$$

The above formulas for viscous flow hold good up to a certain critical region or point which is conveniently determined from fundamental relations explained in succeeding paragraphs and shown graphically in Fig. 15. These relations show that viscous flow exists providing $dvS/z < 0.11$; $GS/dz < 0.27$; dvy/z and $dv/Vz < 6.9$; $Gy/dz < 17$; $W/dz < 133$; $Qs/dz < 1{,}700$.

NOTE.—Read < is less than, and > is greater than.

Rational Solution.—Recent investigations[1] have shown that the friction factor f in the formulas on pages 84 to 87 holds true for both viscous and turbulent flow and is actually a function of pipe diameter, average velocity of flow, density, and absolute viscosity of the fluid. To state this relation in symbols, f is a function of $d'vy/\mu$ (d' here represents inside diameter in feet). The relation of these terms is dimensionless, since the feet, seconds, pounds, etc., cancel out, leaving only a pure number as indicated below

$$\frac{d'vy}{\mu} = \frac{(\text{ft})\left(\frac{\text{ft}}{\text{sec}}\right)\left(\frac{\text{lb}}{\text{ft}^3}\right)}{\frac{(\text{lb})}{(\text{sec})\,(\text{ft})}} = \text{pure number.}$$

[1] (*a*) "Model Experiments and Forms of Empirical Equations," by Earle Buckingham, *Trans. ASME*, Vol. 37, 1915, p. 263. Professor Buckingham's work is based on the principle of "dimensional homogeneity" which states that all the terms of any correct and complete physical equation must have the same dimensions.

(*b*) Contribution 40 from Research Laboratory of Applied Chemistry, M.I.T., by R. E. Wilson, W. H. McAdams, and M. Seltzer, published in the *Jour. Ind. Eng. Chem.*, February, 1922, p. 105. See also "Principles of Chemical Engineering," by W. H. Walker, W. K. Lewis, W. H. McAdams, and E. R. Gilliland, McGraw-Hill Book Company, Inc., New York, 1937.

(*c*) "The Flow of Fluids," by W. H. McAdams. Paper presented before The American Society of Refrigeration Engineers, December, 1924, and published in *Refrigerating Engineering*, Vol. 11, p. 279, February, 1925.

(*d*) "The Flow of Air and Steam in Pipes," by W. H. McAdams and T. K. Sherwood, *Mechanical Engineering*, October, 1926, pp. 1025–1029.

(*e*) "Professor Durand's Statement," by Prof. W. F. Durand, *Jour. Electricity*, San Francisco, May 1, 1920, reprinted in "Handbook of Petroleum Industry," David T. Day, Editor in Chief, and in "National Pipe Standards," of The National Tube Company.

(*f*) "The Flow of Liquids through Short Tubes," by Winslow H. Herschel, *Trans. ASCE*, Vol. 84, p. 527, 1921.

(*g*) "Flow in Pipes," by Michael Aisenstein, *Trans. ASME*, 1928.

(*h*) "Heat Power Engineering," Part III, by Barnard, Ellenwood, and Hirshfeld, John Wiley & Sons, Inc., New York, 1933, pp. 781–821.

(*i*) "Flow of Fluids in Closed Conduits," by R. G. S. Pigott, *Mechanical Engineering*, August, 1933, p. 497.

(*j*) "A Study of the Data on the Flow of Fluids in Pipes," by Emory Kemler, *Trans. ASME*, 1933, HYD-55-2.

(*k*) "Flow of Solids in Piping," by H. E. Babbitt and D. H. Caldwell, *Heating, Piping and Air Conditioning*, July and August, 1942, pp. 423–427, 491–494. A practical application of the rational method to the flow of sludges, sewage, muds, and suspensions in process work. See also "The Flow of Solid-Liquid Mixtures," by same authors published in Proceedings of Second Hydraulics Conference, June 1–4, 1942, *Univ. Iowa Studies in Engineering, Bull.* 27, No. 400, which contains a bibliography of 15 references.

multiplied by 100 to obtain the viscosity in centipoises, *i.e.*, 1 poise is equal to 100 centipoises. It is also the usual practice to express inside pipe diameter in inches rather than feet. As explained above, it is convenient to evaluate the friction factor f as a function of dvy/z rather than $d'vy/\mu$, and to write formulas for pressure drop in pounds per square inch, combining all constants so far as practicable. Although this simplified procedure does not permit the direct use of *true* Reynolds numbers, it utilizes what might be termed a Reynolds *criterion* which accomplishes the same result with less conversions. The relation of true Reynolds numbers to these various criteria is shown in the various abscissas scales of Fig. 15*a*. Where true Reynolds numbers are known, it is possible through this combination of scales to read the corresponding f values directly from either Fig. 15*a* or 15*b*.

For ease in handling solutions involving the variations of the rational formula, it is convenient to evaluate f in the following different ways:

$$f\alpha \frac{dvS}{z};\ f\alpha \frac{GS}{dz};\ f\alpha \frac{dvy}{z} = \frac{dv}{Vz};\ f\alpha \frac{Gy}{dz};\ f\alpha \frac{W}{dz};\ f\alpha \frac{Qs}{dz};\ \text{and}\ f\alpha \frac{d'vy}{\mu}.$$

The symbol α means "is a function of," and here denotes that f is a function of the other quantities at the right of the symbol. The relation $f\alpha \frac{d'vy}{\mu}$ represents the friction factor f as a function of the true or basic Reynolds number. These relations are worked out graphically in Fig. 15*a*. It should be noted that, with one exception, these expressions require separate abscissas scales. For any given pipe size and flow conditions, however, the value of f determined by one of the above expressions is identical with that determined by each of the other expressions. In using the rational method of solution, the proper procedure is to use the expression for f and the form of the friction formula best suiting the particular problem.

The numerical constants preceding f in the friction-loss formulas are kept separate and uncombined with f in order to maintain f as a dimensionless quantity which can be compared directly with similar values worked out for use in metric or other formulas.

The evaluation of f as a function of dvy/z, etc. (or dvS/z as they chose to express it), has been carefully determined by Wilson, McAdams, and Seltzer (see footnote on page 107) from the published results of various investigators of friction loss in pipes. The

above-named authors chose to state their values of f as a function of dvS/z where S denotes specific gravity with reference to water. The use of S serves well enough when dealing with liquids, but the present authors have converted the results of Wilson, McAdams, and Seltzer so as also to employ y representing density in pounds per cubic foot and V representing specific volume in addition to the specific gravity S, in order to obtain greater convenience in applying the solution to gases and steam as well as liquids. The values of f given in the graph shown in Fig. 15*a* are presented in terms of y, V, and S so as to be conveniently applicable to any solution.

Values of f given in Fig. 15 do not include the factor 4, which represents the ratio of the diameter to the hydraulic radius of a circular pipe. This factor is included already in the numerical constants which accompany f in the general flow formulas as used in this handbook (see page 85). It should be noted that in similar plots of friction factors against Reynolds numbers used by other authors [see, for instance, footnote (*h*) on page 107], this factor of 4 sometimes is included in the f values instead of in the formula constants, in which event the f values read from such charts are four times those of Fig. 15. Formulas for f used in plotting Fig. 15*a* are given on page 130.

Solution for Pressure Drop by Rational Method.—Pressure drop can be computed in the rational method by substituting the value of f as determined above in the respective equation for pressure drop. Starting with the fourth edition, this handbook has been revised so that the friction factor f has the same significance in all formulas throughout Chap. II. Hence all formulas containing this term can be used for solutions by the rational method. Attention is called to the existence of the following formulas containing f:

1. General flow formulas for all fluids, see equations (46) to (57) on pages 84 to 87.
2. Basic flow formulas for gas and air, see equations (58) to (75) on pages 167 to 173.
3. Basic hydraulic formulas for water at atmospheric temperature, see page 269.

For solutions in the *viscous* flow region, f is not necessarily required (see formulas on page 106), although it can be used if desired with the formulas listed in 1, 2, and 3 above.

The rational solution is a simple straightforward affair where pressure drop is the unknown and the other terms such a quantity

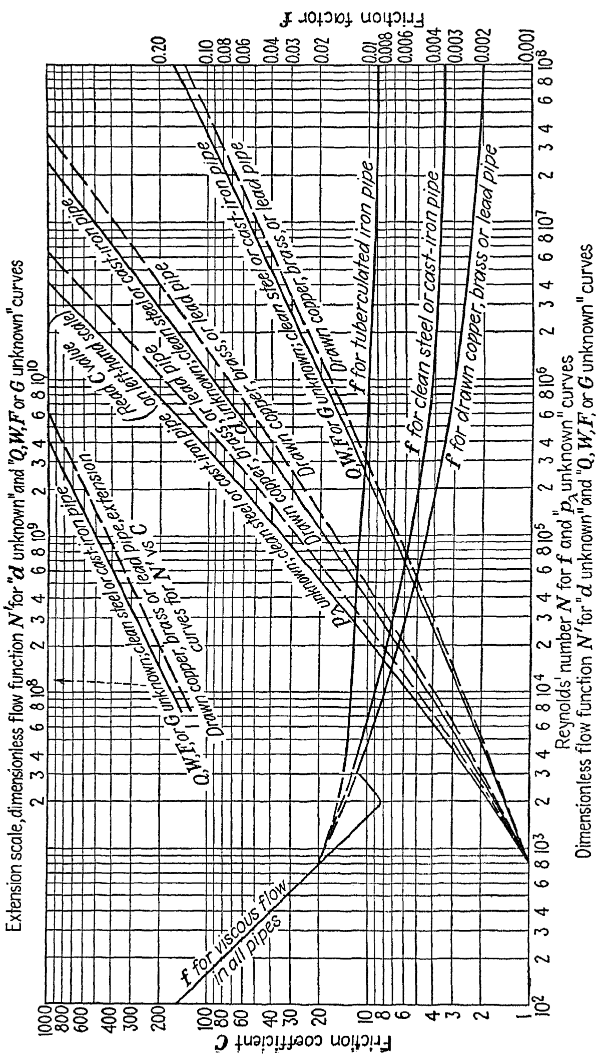


FIG. 15b.—Chart for determining "friction factor f" or "friction coefficient C" for direct solution in computing fluid flow in pipes by rational formula. For explanation, see pages 113–118.

of flow, velocity, and diameter are known and available for use in determining f. Where one or more of these quantities are unknown and have to be determined with respect to some desired pressure drop, there are two possible methods of solution as described in the next section.

Direct Solution for Capacity, Pipe Diameter, and Pressure Drop by Rational Method.—The foregoing discussion of the so-called "rational method" assumes that all factors in the Reynolds number or Reynolds criterion are available, *i.e.*, diameter, velocity, density, and viscosity. Knowledge of the *Reynolds criterion* establishes the appropriate value of the *friction factor f*, thus permitting a direct calculation of the pressure drop. However, if any factor of the Reynolds criterion is unknown, it becomes necessary to resort to a trial-and-error solution as is done in the first solution of Example 6 on page 136, where the diameter is unknown, or to use a variation of the rational method that will permit a direct solution. Where the designer has only an occasional computation to make, the trial-and-error system may suffice, but if frequent calculations are required in which one of the Reynolds criterion factors is unknown it is worth while becoming acquainted with one or more of the schemes that have been developed for direct solution.[1] The following material is based on the paper by B. F. Ruth.

The direct solution by this method is simple, consisting of only three steps as follows:

1. Calculate a "dimensionless flow function" N' using a formula from Table XV that involves the factors that are known.

2. For the dimensionless flow function N' of step (1) read off the "friction coefficient" C from Fig. 15*b*.

3. Solve the appropriate equation from Table XV, using the friction coefficient from step (2). This gives the desired value of the unknown.

[1] Reference may be made to one or more of the following articles:

(*a*) "Direct Solution of Isothermal Flow in Pipes," by B. F. Ruth, *Ind. Eng. Chem.*, August, 1939, p. 985.

(*b*) "Direct Solution of Isothermal Flow in Long Pipes," by C. F. Bonilla, *Ind. Eng. Chem.*, May, 1939, p. 618.

(*c*) "Direct Solution for Pipe Capacity," by Dr. J. R. Zwickl, *Power*, March, 1940, p. (147) 67.

(*d*) "A Survey of Flow Calculation Methods," by S. P. Johnson, Reprinted Papers and Program, Aeronautic and Hydraulic Division, ASME, June, 1934, p. 98.

TABLE XV.—FORMULAS FOR DIRECT SOLUTION FOR CAPACITY, PIPE DIAMETER, AND PRESSURE DROP BY THE RATIONAL METHOD

Section A.—General-purpose formulas for all fluids including liquids, steam, air, gases, and vapors

Unknown term	Known terms	Reynolds number, N, or flow function, N'	Value of C	Formula for value of the unknown	Relationship between f and C
p_λ	d', v, y, μ, L	$N = \frac{d'vy}{\mu}$	ponding curve of Fig. 15b	$p_\lambda = \frac{CL\mu v}{145d'^2}$	$f = \frac{16C\mu}{d'vy}$
	d', v, y, z, L	$N = 1{,}488\frac{d'vy}{z}$		$p_\lambda = \frac{CLzv}{215{,}500d'^2}$	$f = \frac{Cz}{93d'vy}$
	d, v, y, z, L	$N = 124\frac{dvy}{z}$		$p_\lambda = \frac{CLzv}{1{,}497d^2}$	$f = \frac{Cz}{7.75dvy}$
	d, v, S, z, L	$N = 7{,}440\frac{dvS}{z}$			$f = \frac{Cz}{484dvS}$
	$W, y\ d, z, L$	$N = 6.3\frac{W}{dz}$		$p_\lambda = \frac{CLzW}{29{,}400yd^4}$	$f = 2.53\frac{Czd}{W}$
	W, S, d, z, L			$p_\lambda = \frac{CLzW}{1{,}840{,}000Sd^4}$	
	F, y, d, z, L	$N = 22{,}700\frac{Fy}{dz}$		$p_\lambda = \frac{CLzF}{8.16d^4}$	$f = \frac{Czd}{1{,}420Fy}$
	F, S, d, z, L	$N = 1{,}417{,}000\frac{FS}{dz}$			$f = \frac{Czd}{88{,}600FS}$
	G, y, d, z, L	$N = 50.7\frac{Gy}{dz}$		$p_\lambda = \frac{CLzG}{3{,}670d^4}$	$f = \frac{Czd}{3.17Gy}$
	G, S, d, z, L	$N = 3{,}160\frac{GS}{dz}$			$f = \frac{Czd}{198GS}$

d	W, y, z, L, p_λ	$N' = \dfrac{82.3W}{z\sqrt[4]{LzW/p_\lambda y}}$	To be read from corres	$d = 0.0765\sqrt[4]{\dfrac{CLzW}{p_\lambda y}}$	$f = \dfrac{Cz\sqrt[4]{CLzW/p_\lambda y}}{5.15W}$
	W, S, z, L, p_λ	$N' = \dfrac{231W}{z\sqrt[4]{LzW/p_\lambda S}}$		$d = 0.0272\sqrt[4]{\dfrac{CLzW}{p_\lambda S}}$	$f = \dfrac{Cz\sqrt[4]{CLzW/p_\lambda S}}{14.48WS}$
	F, y, z, L, p_λ	$N' = \dfrac{38,300Fy}{z\sqrt[4]{LzF/p_\lambda}}$		$d = 0.593\sqrt[4]{\dfrac{CLzF}{p_\lambda}}$	$f = \dfrac{Cz\sqrt[4]{CLzF/p_\lambda}}{2,390Fy}$
	F, S, z, L, p_λ	$N' = \dfrac{2,380,000FS}{z\sqrt[4]{LzF/p_\lambda}}$			$f = \dfrac{Cz\sqrt[4]{CLzF/p_\lambda}}{149,000FS}$
	G, y, z, L, p_λ	$N' = \dfrac{392Gy}{z\sqrt[4]{LzG/p_\lambda}}$		$d = 0.129\sqrt[4]{\dfrac{CLzG}{p_\lambda}}$	$f = \dfrac{Cz\sqrt[4]{CLzG/p_\lambda}}{24.6Gy}$
	G, S, z, L, p_λ	$N' = \dfrac{24,450GS}{z\sqrt[4]{LzG/p_\lambda}}$			$f = \dfrac{Cz\sqrt[4]{CLzG/p_\lambda}}{1,534GS}$
W	p_λ, y, d, z, L	$N' = 185,400\dfrac{p_\lambda yd^3}{Lz^2}$		$W = \dfrac{29,400p_\lambda d^4y}{CLz}$	$f = \dfrac{C^2Lz^2}{11,600p_\lambda yd^3}$
	p_λ, S, d, z, L	$N' = 11,580,000\dfrac{p_\lambda Sd^3}{Lz^2}$		$W = \dfrac{1,840,000p_\lambda d^4S}{CLz}$	$f = \dfrac{C^2Lz^2}{724,000p_\lambda Sd^3}$
F	p_λ, y, d, z, L	$N' = 185,400\dfrac{p_\lambda yd^3}{Lz^2}$		$F = \dfrac{8.16p_\lambda d^4}{CLz}$	$f = \dfrac{C^2Lz^2}{11,600p_\lambda yd^3}$
	p_λ, S, d, z, L	$N' = 11,580,000\dfrac{p_\lambda Sd^3}{Lz^2}$			$f = \dfrac{C^2Lz^2}{724,000p_\lambda Sd^3}$
G	p_λ, y, d, z, L	$N' = 185,400\dfrac{p_\lambda yd^3}{Lz^2}$		$G = \dfrac{3,670p_\lambda d^4}{CLz}$	$f = \dfrac{C^2Lz^2}{11,600p_\lambda yd^3}$
	p_λ, S, d, z, L	$N' = 11,580,000\dfrac{p_\lambda Sd^3}{Lz^2}$			$f = \dfrac{C^2Lz^2}{724,000p_\lambda Sd^3}$

TABLE XV.—(*Concluded*)

Section B.—Special formulas for gas or air measured at 60 F and 14.7 psi abs

Unknown term	Known terms	Reynolds number, N, or flow function, N'	Value of C	Formula for value of the unknown	Relationship between f and C
Q	$p_1, p_2, s, d, z, L, T_c$	$N' = 249{,}600 \frac{(p_1^2 - p_2^2)sd^3}{Lz^2T_c}$	To be read from corresponding curve of Fig. 15b	$Q = \frac{1{,}000(p_1^2 - p_2^2)d^4}{CLz}$	$f = \frac{C^2Lz^2T_c}{15{,}704(p_1^2 - p_2^2)sd^3}$
W				$W = \frac{39{,}728(p_1^2 - p_2^2)sd^4}{CLzT_c}$	
p_λ	p_1, Q, s, d, z, L, T_c	$N = \frac{250Q_s}{dzT_c}$		$p_\lambda = p_1 - \sqrt{p_1^2 + \frac{CLzQ}{1{,}000d^4}}$	$f = \frac{CzdT_c}{15.8Q_s}$
	p_2, Q, s, d, z, L, T_c			$p_\lambda = -p_2 + \sqrt{p_2^2 + \frac{CLzQ}{1{,}000d^4}}$	
d	$p_1, p_\lambda, Q, s, z, L, T_c$	$N' = \frac{1{,}400Q_s}{zT_c \sqrt[4]{LzQ/(p_1p_\lambda - p_\lambda^2)}}$		$d = 0.178 \sqrt[4]{\frac{CLzQ}{2p_1p_\lambda - p_\lambda^2}}$	$f = \frac{CzT_c \sqrt[4]{CLzQ/(2p_1p_\lambda - p_\lambda^2)}}{87.6Q_s}$
	and, since $p_2 = p_1 - p_\lambda$, $p_1, p_2, Q, s, z, L, T_c$	$N' = \frac{1{,}400Q_s}{zT_c \sqrt[4]{LzQ/(p_1^2 - p_2^2)}}$		$d = 0.178 \sqrt[4]{\frac{CLzQ}{p_1^2 - p_2^2}}$	$f = \frac{CzT_c \sqrt[4]{CLzQ/(p_1^2 - p_2^2)}}{87.6Q_s}$

NOTES.—See text for explanation of the formulas and their application. For definition of symbols see pp. 82 to 84.

Distinction between *specific gravity* terms for liquids and gases:

S denotes the specific gravity of a liquid at its actual flow temperature referred to water having a density of 62.4 lb per cu ft (see p. 124 for explanation and references to data).

s denotes the specific gravity of a gas under standard conditions (approximating 14.7 psi abs and 60 F) relative to air under the same conditions (see p. 158). Adjustment to any other flow conditions is made through the pressure terms in the formulas.

Explanation of Table.—It will be noted that Table XV consists of two sections, the first of which applies to all fluids, be they liquid, vapor, or gas. In using this section care should be exercised to take the density or specific gravity of the fluid corresponding to the *average pressure under actual flow conditions* in the pipe. This is relatively simple where the density remains practically constant, as in the case of liquids, or where the pressure drop in a gas or vapor line is small in comparison with the static pressure in the line.

The second section of the table was prepared to facilitate solving problems in the flow of gas or air where significant changes of density may occur as the result of pressure drop and where common practice refers to the flow in terms of "free gas" or "free air." In these formulas the specific gravity s has the commonly available value corresponding to standard conditions of 14.7 psi abs and 60 F referred to dry air at the same conditions (see page 158). Adjustments of the specific gravity from standard conditions to the actual average pressure and temperature in the line is made through the pressure and temperature terms in the special formulas.

Solution for Capacity.—If, for example, the weight of flow W is unknown when p_λ, y, d, L, and z are given, the dimensionless "flow function" N' as computed from the formula in Section A of Table XV is

$$N' = 185{,}400 \frac{\mu \lambda y \omega}{Lz^2}.$$

For this value of the function, read the respective value of C from Fig. 15*b* referring to the curve "Q, W, F, or G Unknown." With this value of C and the other known factors, the desired weight of flow in pounds per hour can be determined from the corresponding formula in Section A of Table XV as

$$W = \frac{29{,}400 p_\lambda d^4 y}{CLz}.$$

A solution for capacity in terms of F or G for liquids or of Q for free air or gas is made in exactly the same manner, using the appropriate formulas from Section A or B of Table XV.

Solution for Diameter.—If the diameter d of a line is sought when the values of, say, F, S, z, L, and p_λ are known, the dimensionless

"flow function" is computed from the formula

$$N' = \frac{2{,}380{,}000FS}{z\sqrt[4]{\dfrac{LzF}{p_\lambda}}}$$

For this value of the function, read the corresponding value of C from Fig. 15*b* referring to the curve "d Unknown." With this value of C and the other known factors the required diameter can be determined from the corresponding formula of Table XV as

$$d = 0.593\sqrt[4]{\frac{CLzF}{p_\lambda}}.$$

Solution for Pressure Drop.—The formulas of Table XV and the curve "p_λ Unknown" of Fig. 15*b* may be used in calculating the pressure drop as an alternate to the use of f values and the formulas on pages 85 to 87. Both calculations give the same answer for p_λ. Suppose, for example, it is desired to calculate the pressure drop when G, y, d, z, and L are known. From Section A of Table XV the Reynolds number

$$N = 50.7\frac{Gy}{dz}.$$

For this value of the function, read the corresponding value of C from Fig. 15*b* referring to the curve "p_λ Unknown." With this value of C and the other known factors the pressure drop can be determined from the corresponding formula of Table XV as

$$p_\lambda = \frac{CLzG}{3670d^4}.$$

Relationship between f and C.—Since the direct solution using C values gives the same answer to a problem as would be obtained by the trial-and-error solution using f values, it follows that some definite relation exists between f and C. Table XV shows this relationship in terms of the several groups of known factors. It is not necessary to calculate the value of f in using the direct solution. The formulas are presented merely to show that the direct solution is based upon the same values of f that are used consistently throughout this chapter.

Determination of Viscosity.—The values of z for a number of liquids at various temperatures given in Fig. 16 were taken largely

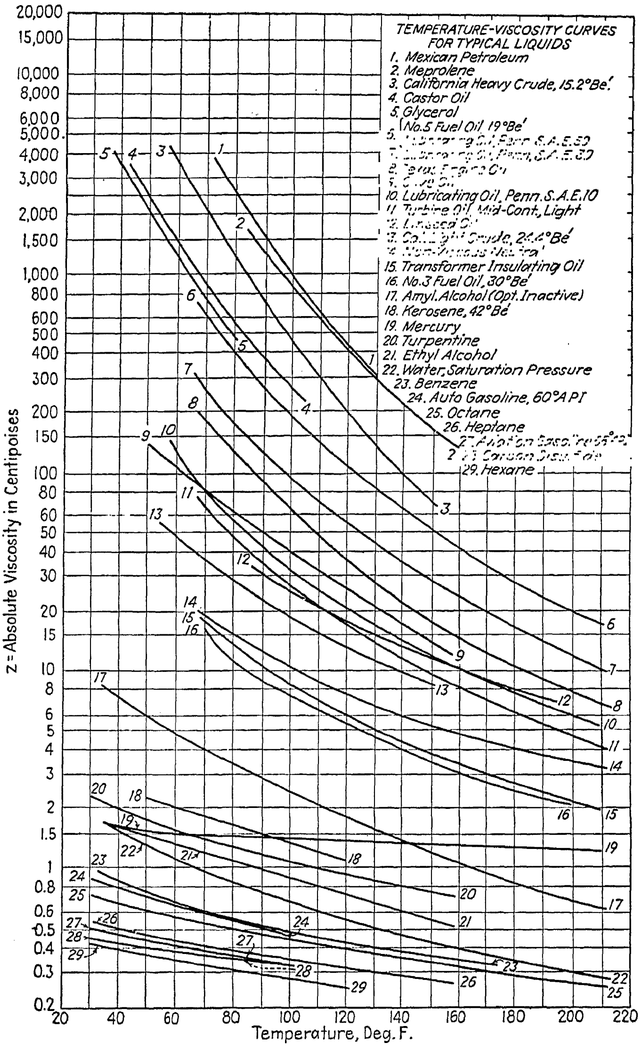
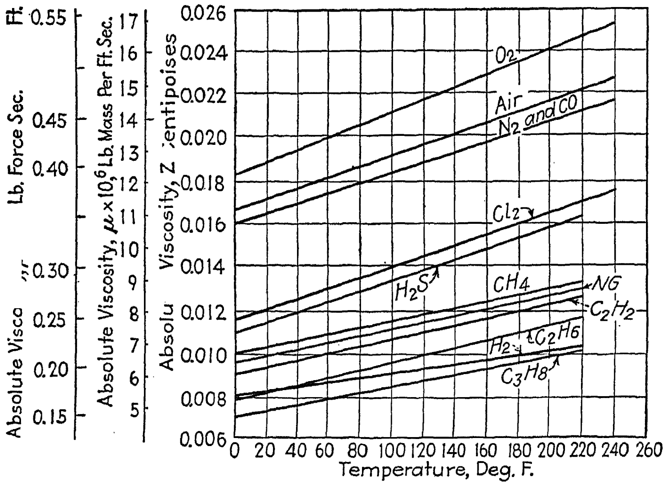


Fig. 16.—Temperature *vs.* viscosity curves for liquids.

from published data.[1] Viscosity data for commercial gases given in Fig. 17*a* were taken from Report of ASME Special Research Committee on Fluid Meters, Part I, 4th ed., 1937, page 97, except



Symbol	Name	Sutherland's constant	Symbol	Name	Sutherland's constant
O_2	Oxygen	138	H_2S	Hydrogen sulfide	331
N_2	Nitrogen	111	CH_4	Methane	198
CO	Carbon monoxide	118	C_2H_2	Acetylene	
Cl_2	Chlorine	325	H_2	Hydrogen	77.5
Air	Air	114	NG	Natural gas	133
C_3H_8	Propane	284	C_2H_6	Ethane	252

FIG. 17*a*.—Temperature *vs.* viscosity of gases.

for natural gas, ethane, and propane, values for which are from U. S. Bureau of Mines *Tech. Paper* 555 on "Viscosity of Natural Gas," by W. B. Berwald and T. W. Johnson.[2] Since the viscosi-

[1] (*a*) "Principles of Chemical Engineering," by W. H. Walker, W. K. Lewis, W. H. McAdams, and E. R. Gilliland, McGraw-Hill Book Company, Inc., New York, 1937.

(*b*) "Chemical Technology of Petroleum," by W. A. Gruse and D. R. Stevens, McGraw-Hill Book Company, Inc., New York, 1942.

[2] For further data on the viscosity of nitrogen see "The Dynamic Viscosity of Nitrogen," by W. L. Sibbitt, G. A. Hawkins, and H. L. Solberg, *Trans. ASME*, Vol. 65, No. 5, pp. 401–405, July, 1943.

ties of gas mixtures, such as blast furnace gas, are difficult to compute the values usually are obtained experimentally. Berwald and Johnson have outlined in *Tech. Paper* 555 a method for approximating the viscosity of natural gases which may be applied to other gases of similar composition. Viscosities of commonly used refrigerant gases are included in Chap. XVI of "Piping Handbook."

Viscosities of gases at temperatures in excess of those shown in Fig. 17*a* may be extrapolated by Sutherland's[1] formula, which is as follows:

$$z = z_0 \left(\frac{0.555T_0 + c}{0.555T + c}\right)\left(\frac{T}{T_0}\right)^{3/2}$$

where z = viscosity, in centipoises, at temperature T.
z_0 = viscosity, in centipoises, at temperature T_0.
T = temperature, in degrees F abs, for which viscosity is desired.
T_0 = temperature, in degrees F abs, for which viscosity is known.
c = Sutherland's constant, values for which are shown for the gases indicated in Fig. 17*a*.

Temperature, in general, influences the viscosity of fluids to a much greater extent than does pressure. Experimental data on the viscosity of gases at high pressures, however, indicate that Maxwell's law, which states that the viscosity of a gas is independent of the pressure, is not entirely correct. Where experimental data are not available, a method of predicting the viscosity of a gas at high pressures, with respect to the viscosity at atmospheric pressure and the critical pressure and temperature of the gas, is contained in an article by E. W. Comings and R. S. Egly entitled "Viscosity of Gases and Vapors at High Pressures."[2]

The effect of pressure on viscosity of liquids varies with the liquid involved. The viscosity of water[3] increases only slightly with increase in pressure, but that of many organic liquids and

[1] See "International Critical Tables," Vol. 5, p. 1, 1st ed., 1929, published for the National Research Council by the McGraw-Hill Book Company, Inc., New York.

[2] *Ind. Eng. Chem.*, Vol. 32, No. 5, pp. 714–718, 1940. See also, "Viscosities of the Methane-Propane System," by Leo B. Bicher, Jr. and Donald L. Katz, *Ind. Eng. Chem.*, Vol. 35, No. 7, pp. 754–761, 1943, with 19 references.

[3] For the viscosity of water at higher pressures and temperatures, see "Properties of Ordinary Water-Substance," by N. E. Dorsey, Reinhold Publishing Corporation, New York, 1940, Table 86, p. 188.

mineral, animal, and vegetable oils increases as much as twenty times at pressures of 15,000 psi. For further data, see the "International Critical Tables," Vol. VII, pages 222 and 223.

Viscosity data for superheated steam at pressures from atmospheric to 2,500 psi and temperatures from 250 to 1200 F are given in Fig. 17*b*. This chart is based on viscosity determinations at Purdue University as reported in a paper on "The Viscosity of Superheated Steam" by G. A. Hawkins, H. L. Solberg, and A. A. Potter, *Trans. ASME*, 1940, Vol. 62, No. 8, pp. 677–688. The

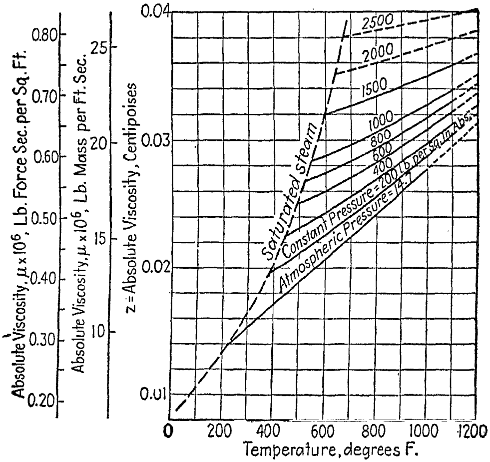


FIG. 17*b*.—Temperature *vs.* viscosity of steam.

curves have been modified slightly and extrapolated from the data in their paper.

For most commercial purposes it is desirable to make an independent determination of the viscosity of the liquid in question. This can be done in a practical way by employing a standard instrument, such as a Saybolt Universal viscometer or Saybolt Furol, Engler, or other commonly used viscometer.[1] The significance of viscosity is explained at more length under "Properties of Oils" on page 204. A convenient chart for converging such viscometer readings into kinematic viscosity is given in Fig. 18.

[1] See ASTM D88 for "Standard Method of Test for Viscosity by Means of the Saybolt Viscosimeter," or ASTM D445 for "Method of Test for Kinematic Viscosity."

The kinematic viscosity of a fluid is equal to its absolute viscosity divided by its density at the temperature at which the viscosity is measured. These relations expressed in metric and English units

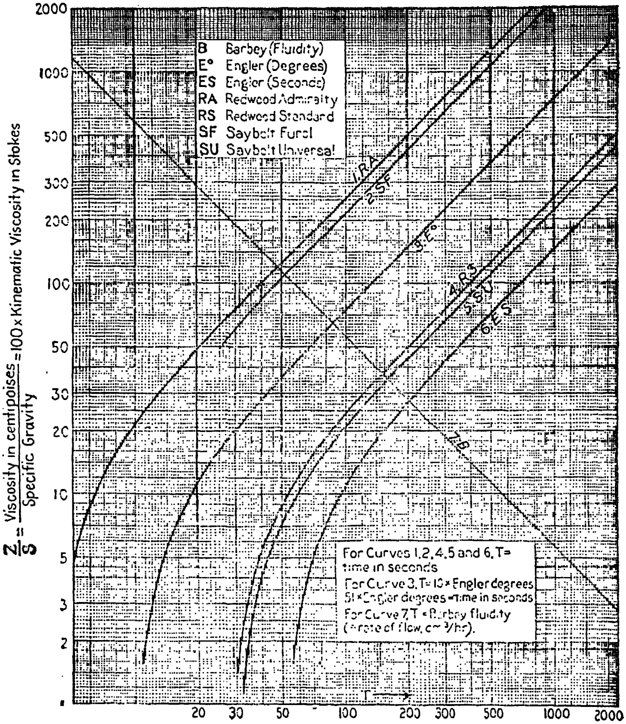


FIG. 18.—Viscometer time of efflux *vs.* z/S. Note: The value of z/S when time of efflux T exceeds 2,000 is 10 times the value of z/S corresponding to $T/10$. (*Reproduced by permission from International Critical Tables.*)

are, respectively, as follows:

$$\text{Kinematic viscosity in sq cm per sec} = \frac{\text{absolute viscosity in poises}}{\text{density in g per cu cm}}$$

$$= \frac{\text{absolute viscosity in centipoises}}{100 \text{ density in g per cu cm}} = \frac{z}{100y}$$

$$= \frac{\text{absolute viscosity in centipoises}}{\text{specific gravity}} = \frac{z}{100S}$$

where poise = 1 g mass per cm sec = cgs unit of absolute viscosity.
centipoise = 0.01 poise.
stoke = 1 sq cm per sec = cgs unit of kinematic viscosity.
centistoke = 0.01 stoke.
centipoises = centistokes × density (at designated temperature).
y = density, g per cu cm.

$$\text{Kinematic viscosity in sq ft per sec} = \frac{\text{absolute viscosity in lb mass per ft sec}}{\text{density in lb per cu ft}}$$

No name for the English unit of kinematic viscosity is generally recognized. Conversion from one set of units of kinematic viscosity to the other may be accomplished by the following relation: a kinematic viscosity of 1 sq ft per sec equals 929 times the kinematic viscosity in sq cm per sec.

The following formula may be used to determine kinematic viscosity from Saybolt Universal viscometer readings:

$$\text{Kinematic viscosity in sq cm per sec} \quad 0.0022T - \frac{1.8}{T}$$

where T = Saybolt time, sec.

The corresponding formula for Saybolt Furol is

$$\text{Kinematic viscosity in sq cm per sec} = 0.022T - \frac{2.03}{T}$$

(See ASTM D446 for Standard Method for Conversion of Kinematic Viscosity to Saybolt Universal Viscosity.)

Specific Gravity, Density, Etc.—The specific gravity sometimes used is the ratio of the weight of 1 cu ft of the liquid at the temperature in question to the weight of 1 cu ft of water at its temperature of maximum density, *i.e.*, 39.1 F (4 C). The weight of 1 cu ft of water under these conditions is 62.425 lb. There is no generally accepted practice as to what temperature of water shall be used in specific-gravity determinations. Water at 60 F is frequently used, and sometimes at 62 F or 68 F. The density of water does not change materially between 39 F and 68 F and calculations are usually made on the approximate basis that water weighs 62.4 lb per cu ft. For the present purpose in friction-loss calculations, this value is as accurate as the other factors warrant.

A schedule of temperature corrections for Baumé hydrometer readings is given in Table XVI. Table XVII gives the specific gravities corresponding to various degrees Baumé readings of the Bureau of Standards and API scales (see also Table XXVIII, page 203).

TABLE XVI.—TEMPERATURE CORRECTIONS TO READINGS OF BAUMÉ HYDROMETERS IN AMERICAN PETROLEUM OILS AT VARIOUS TEMPERATURES[1]
(Standard at 60 F, modulus 140)

Observed temperature, degrees Fahrenheit	Observed degrees Baumé							
	20.0	30.0	40.0	50.0	60.0	70.0	80.0	90.0
	Add to observed degrees Baumé							
30	1.7	2.0	2.4	3.0	3.7	4.3	5.0	5.7
32	1.6	1.9	2.3	2.8	3.4	4.0	4.7	5.3
34	1.5	1.8	2.1	2.6	3.1	3.7	4.3	4.9
36	1.4	1.6	2.0	2.4	2.9	3.4	4.0	4.6
38	1.3	1.5	1.8	2.2	2.6	3.1	3.6	4.2
40	1.2	1.4	1.6	2.0	2.4	2.8	3.2	3.8
42	1.1	1.2	1.5	1.8	2.2	2.5	2.9	3.4
44	0.9	1.1	1.3	1.6	2.0	2.2	2.6	3.0
46	0.8	0.9	1.1	1.4	1.7	1.9	2.3	2.7
48	0.7	0.8	0.9	1.2	1.4	1.6	2.0	2.3
50	0.6	0.7	0.8	1.0	1.2	1.4	1.6	1.9
52	0.5	0.6	0.7	0.8	1.0	1.1	1.3	1.5
54	0.3	0.4	0.5	0.6	0.8	0.9	1.0	1.1
56	0.2	0.3	0.3	0.4	0.5	0.6	0.6	0.7
58	0.1	0.1	0.1	0.2	0.3	0.3	0.3	0.4
	Subtract from observed degrees Baumé							
60	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0 0
62	0.1	0.1	0.1	0.2	0.2	0.3	0.3	0.4
64	0.2	0.3	0.3	0.4	0.4	0.6	0.6	0.7
66	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
68	0.5	0.6	0.6	0.7	0.9	1.1	1.3	1.4
70	0.6	0.7	0.8	0.9	1.1	1.4	1.6	1.7
72	0.7	0.8	0.9	1.1	1.3	1.6	1.9	2.1
74	0.8	0.9	1.1	1.3	1.6	1.8	2.2	2.5
76	0.9	1.1	1.3	1.5	1.8	2.1	2.5	2.8
78	1.0	1.2	1.4	1.7	2.0	2.4	2.8	3.1
80	1.1	1.3	1.5	1.8	2.2	2.6	3.1	3.5
82	1.2	1.4	1.7	2.0	2.5	2.9	3.4	3.9
84	1.3	1.5	1.8	2.2	2.7	3.2	3.7	4.3
86	1.4	1.7	2.0	2.4	2.9	3.4	4.0	4.6
88	1.6	1.8	2.1	2.6	3.1	3.7	4.2	4.9
90	1.7	2.0	2.3	2.7	3.3	3.9	4.5	5.2
92	1.8	2.1	2.4	2.9	3.5	4.2	4.8	5.6
94	1.9	2.2	2.6	3.1	3.8	4.4	5.1	5.9
96	2.0	2.3	2.7	3.3	4.0	4.6	5.4	6.3
98	2.1	2.4	2.9	3.4	4.2	4.9	5.7	6.6
100	2.2	2.6	3.0	3.6	4.4	5.1	6.0	6.9
102	2.3	2.7	3.2	3.8	4.6	5.4	6.3	7.2
104	2.4	2.9	3.3	4.0	4.8	5.7	6.6	7.5
106	2.5	3.0	3.5	4.2	5.0	5.9	6.9	7.9
108	2.7	3.1	3.6	4.3	5.2	6.2	7.2	8.2
110	2.8	3.2	3.7	4.4	5.4	6.4	7.5	8.5
112	2.9	3.3	3.9	4.6	5.6	6.7	7.7	8.8
114	3.0	3.4	4.0	4.7	5.8	6.9	7.9	9.1
116	3.1	3.6	4.1	4.9	6.0	7.1	8.2	9.4
118	3.2	3.7	4.3	5.1	6.2	7.3	8.5	9.8
120	3.3	3.8	4.4	5.3	6.4	7.5	8.8	10.1

This table is calculated from the same data as Table II, *Circ.* 57, Bureau of Standards.

[1] Reprinted by permission from "Handbook of Petroleum Industry," Vol. II, by the late David T. Day, published by John Wiley & Sons, Inc., New York, 1922.

TABLE XVII.—RELATION OF BAUMÉ SCALES TO SPECIFIC GRAVITIES AT 60/60 F

Bureau of Standards and API Scales (see also p. 203)

Hydrometer readings, degrees	Sp. gr. at 60°/60° F.			Hydrometer readings, degrees	Sp. gr. at 60°/60° F.			Hydrometer readings, degrees	Sp. gr. at 60°/60° F.		
	Liquids heavier than water	Liquids lighter than water			Liquids heavier than water	Liquids lighter than water			Liquids heavier than water	Liquids lighter than water	
	Bu. of Stds.		A.P.I.		Bu. of Stds.		A.P.I.		Bu. of Stds.		A.P.I.
0	1.0000	......	......	34	1.3063	0.8537	0.8550	68	1.8831	0.7071	0.7093
1	1.0069	......	......	35	1.3182	0.8485	0.8498	69	1.9079	0.7035	0.7057
2	1.0140	......	......	36	1.3303	0.8434	0.8448	70	1.9333	0.7000	0.7022
3	1.0211	......	......	37	1.3426	0.8383	0.8398	71	1.9595	0.6965	0.6988
4	1.0284	......	......	38	1.3551	0.8333	0.8348	72	1.9863	0.6931	0.6952
5	1.0357	......	......	39	1.3679	0.8284	0.8299	73	2.0139	0.6897	0.6919
6	1.0432	......	......	40	1.3810	0.8235	0.8251	74	2.0423	0.6863	0.6886
7	1.0507	......	......	41	1.3942	0.8187	0.8203	75	2.0714	0.6829	0.6852
8	1.0584	......	......	42	1.4078	0.8140	0.8155	76	2.1015	0.6796	0.6819
9	1.0662	......	......	43	1.4216	0.8093	0.8109	77	2.1324	0.6763	0.6787
10	1.0741	1.0000	1.0000	44	1.4356	0.8046	0.8063	78	2.1642	0.6731	0.6754
11	1.0821	0.9929	0.9930	45	1.4500	0.8000	0.8017	79	2.1970	0.6699	0.6722
12	1.0902	0.9859	0.9861	46	1.4647	0.7955	0.7972	80	......	0.6667	0.6690
13	1.0985	0.9790	0.9792	47	1.4796	0.7910	0.7927	81	......	0.6635	0.6659
14	1.1069	0.9722	0.9725	48	1.4949	0.7865	0.7883	82	......	0.6604	0.6628
15	1.1154	0.9655	0.9659	49	1.5104	0.7821	0.7839	83	......	0.6573	0.6597
16	1.1240	0.9589	0.9593	50	1.5263	0.7778	0.7796	84	......	0.6542	0.6566
17	1.1328	0.9524	0.9529	51	1.5426	0.7735	0.7753	85	......	0.6512	0.6536
18	1.1417	0.9460	0.9465	52	1.5591	0.7692	0.7711	86	......	0.6482	0.6506
19	1.1508	0.9396	0.9402	53	1.5761	0.7650	0.7669	87	......	0.6452	0.6476
20	1.1600	0.9333	0.9340	54	1.5934	0.7609	0.7628	88	......	0.6422	0.6446
21	1.1694	0.9272	0.9279	55	1.6111	0.7568	0.7587	89	......	0.6393	0.6417
22	1.1789	0.9211	0.9218	56	1.6292	0.7527	0.7547	90	......	0.6364	0.6388
23	1.1885	0.9150	0.9159	57	1.6477	0.7487	0.7507	91	......	0.6335	0.6360
24	1.1984	0.9091	0.9100	58	1.6667	0.7447	0.7467	92	......	0.6306	0.6331
25	1.2083	0.9032	0.9042	59	1.6861	0.7407	0.7428	93	......	0.6278	0.6303
26	1.2185	0.8974	0.8984	60	1.7059	0.7368	0.7389	94	......	0.6250	0.6275
27	1.2288	0.8917	0.8927	61	1.7262	0.7330	0.7351	95	......	0.6222	0.6247
28	1.2393	0.8861	0.8871	62	1.7470	0.7292	0.7313	96	......	0.6195	0.6220
29	1.2500	0.8805	0.8816	63	1.7683	0.7254	0.7275	97	......	0.6167	0.6193
30	1.2609	0.8750	0.8762	64	1.7901	0.7217	0.7238	98	......	0.6140	0.6166
31	1.2719	0.8696	0.8708	65	1.8125	0.7180	0.7201	99	......	0.6114	0.6139
32	1.2832	0.8642	0.8654	66	1.8354	0.7143	0.7165	100	......	0.6087	0.6112
33	1.2946	0.8589	0.8602	67	1.8590	0.7107	0.7128				

NOTE.—For more complete data see Table 5, *Circ.* 410, Bureau of Standards, "National Standard Petroleum Oil Tables," also ASTM Standard D287.

To convert degrees Baumé Bureau of Standards to specific gravity at 60/60 F use the formulas

$$S = \frac{140}{130 + °\text{Bé}} \text{ for liquids lighter than water,}$$

and

$$S = \frac{145}{145 - °\text{Bé}} \text{ for liquids heavier than water.}$$

To convert degrees Baumé Bureau of Standards to weight per cubic foot at 60/60 F use the formulas

$$y = \frac{62.37 \times 140}{130 + °\text{Bé}} \text{ for liquids lighter than water,}$$

and

$$y = \frac{62.37 \times 145}{145 - °\text{Bé}} \text{ for liquids heavier than water.}$$

The above conversion modulus is that known as the "U. S. Bureau of Standards." A modification commonly used in the petroleum industry and elsewhere for liquids lighter than water employs $\frac{141.5}{131.5 + °\text{Bé}}$ instead of $\frac{140}{130 + °\text{Bé}}$. This modification, formerly known as the "Baumé Tagliabue scale," is now designated as the API scale. It has been adopted as standard by the American Petroleum Institute, and is now generally followed. Owing to the previous use of both scales it is always advisable to specify which is used in stating Baumé readings. The general practice with liquids heavier than water is to state specific gravity rather than degrees Baumé, or API. The relation between specific gravity, specific heat, and temperature for petroleum oils is given in Fig. 32 on page 209. The specific gravities of oil corresponding to different degrees Baumé are given in Tables XXXI and XXXII on pages 222 and 223.

The specific gravity of gases relative to water at 30.1 F may be calculated from the formula

$$S = \frac{0.001496pM}{t + 460} \text{ (see also Table XXIII, page 166)}$$

where p is the absolute pressure in pounds per square inch, M the molecular weight, and t the temperature in degrees Fahrenheit.

The weight per cubic foot of gases may be calculated in a similar way from the formula

$$y = \frac{62.4 \times 0.001496pM}{t + 460}.$$

Obviously, where the specific gravity relative to water of any fluid is known, it can be converted to weight per cubic foot (or y) by multiplying by 62.4. Data concerning the molecular weight, density relative to air, etc., for some of the more common gases are given in Table XXIII, page 166. The densities and specific volumes of steam under different conditions are given in the

steam tables on pages 230 to 243. The specific gravities relative to water of certain common gases and steam at different temperatures are given in Fig. 19 which was taken from the work of Wilson, McAdams, and Seltzer. The data on viscosity and density of brine solutions given in Figs. 20 and 21 were taken from the paper

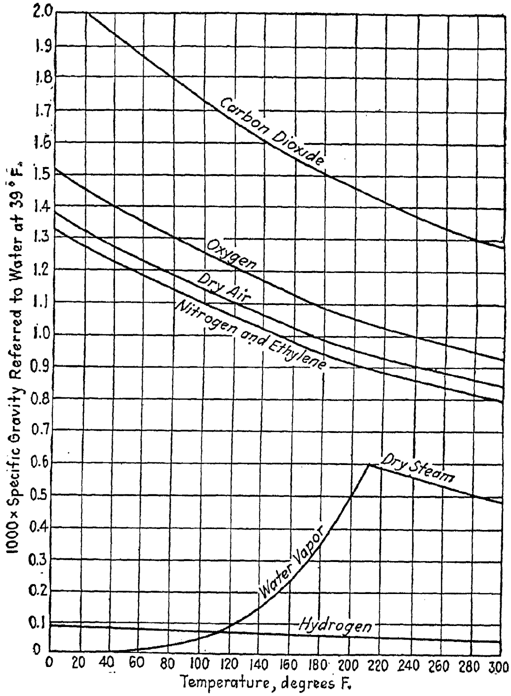


FIG. 19.—Temperature-density curves for typical gases. (Pressure is constant at 1 atm except for water vapor below 212 F.)

of McAdams on "The Flow of Liquids." The temperature *vs.* specific-gravity curves for various liquids given in Fig. 22 were taken mostly from the paper by Wilson, McAdams, and Seltzer. Obviously, the density or specific volume of steam and air can be taken to best advantage from the steam and air tables in general use. Steam tables will be found on pages 230 to 243 and air data

in Table XVIII on page 146 and Figs. 24 and 26 on pages 152 and 155, respectively.

Elbow Friction.—Wilson, McAdams, and Seltzer also applied their method of solution to existing data regarding elbow friction in pipes and demonstrated that elbow friction varies as a function of dvS/z, dvy/z, etc. This relation for screwed elbows is plotted

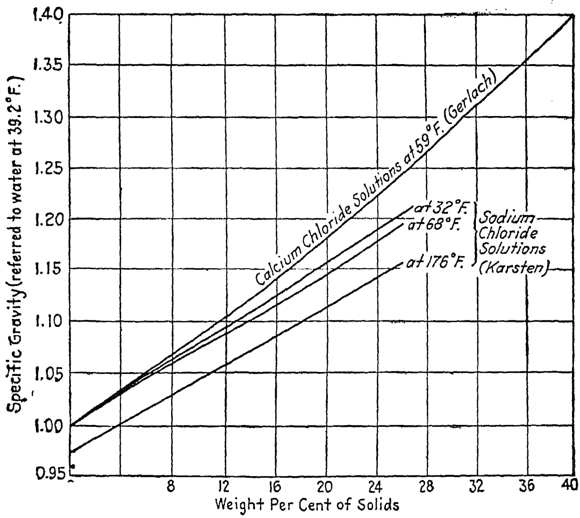


FIG. 20.—Effect of concentration and of temperature upon specific gravities of water solutions of sodium chloride and of calcium chloride.

in Fig. 15*a* which also shows the friction factor f for straight pipe. Elbow friction is expressed as equal to the friction of a certain length of straight pipe measured in diameters. This arrangement is convenient since both elbow friction and friction factor can be determined from the same solution for dvy/z or equivalent expression.

Average Velocity in the Cross Section.—McAdams, in his paper on "The Flow of Fluids" presented before the American Society of Refrigerating Engineers (see footnote on page 107), also worked out a relation between average velocity for the entire cross section of a pipe and maximum velocity at the center line for different values of dvS/z. This relation is shown in Fig. 15*a*. In this paper

McAdams also gave a curve for "tuberculated iron pipe" which is included in Fig. 15*a*.

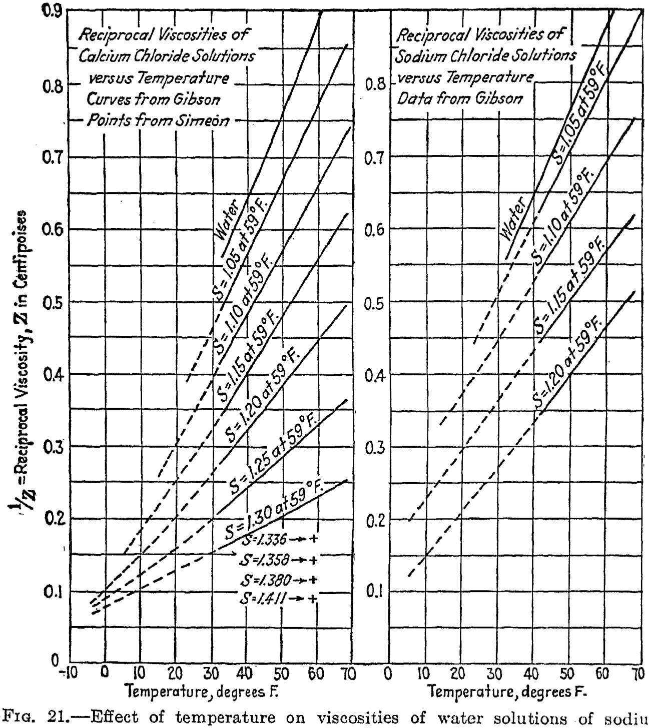


FIG. 21.—Effect of temperature on viscosities of water solutions of sodium chloride and of calcium chloride.

Formulas for f Curves.—The following formulas used in plotting the corresponding curves of Fig. 15 were taken from page 114 of the reference in footnote (*b*) on page 107:

Formula for curve A:

$$f = 0.0018 + 0.00662\left(\frac{z}{dvS}\right)^{0.355}.$$

Formula for curve B:

$$f = 0.0035 + 0.00594\left(\frac{z}{dvS}\right)^{0.424}.$$

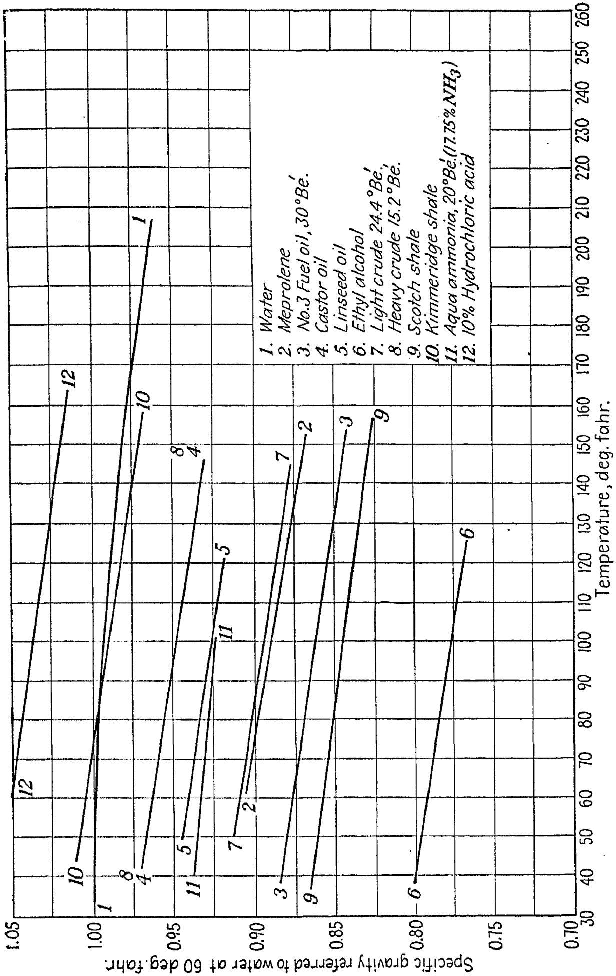


FIG. 22.—Temperature-specific-gravity curves for typical liquids.

Curve C is straight line from $dvS/z = 1.0$.

Curve D for viscous flow is straight line for which the formula is $f = 0.00207z/dvS$.

Curve E is straight line from $dvS/z = 4.0$.

Limitations of Formulas for Friction Loss in Pipes.—1. In applying friction-loss formulas to the flow of steam or gases it is necessary to take into account the pressure drop along the pipe. Where the pressure drop exceeds 10 to 15 per cent of the final absolute pressure, the density and velocity are appreciably changed. Under these conditions *average* values for density and velocity should be used or the expression $(p_1^2 - p_2^2)$ or its equivalent put in the formula (see pages 168 to 171). If this is done, the results will be reasonably accurate for pressure drops up to 40 or 50 per cent of the final absolute pressure.

2. Ordinary friction-loss formulas are not applicable to the flow of semi-solid plastic materials such as asphalt, clay suspensions, very viscous colloidal solutions, etc., where the laws of flow are modified by the tendency of the material to behave like a solid under certain conditions. For an application of the rational formula method to such problems, see (k) in footnote on page 107.

3. Allowance must be made where there is a tendency to reduce the cross-sectional area through adhesion of solid matter to the inside wall, or through the formation of tuberculations. The formation of tuberculations in water pipes also tends to increase the friction factor as explained on page 271. The middle curve for f on Fig. 15 is recommended for use for the flow of all fluids in "commercially smooth" pipes such as welded or seamless steel, cast iron, etc., while the lower line applies to "very smooth" pipes such as drawn brass, copper, or lead pipes or glass tubes. Obviously, values still higher than those given by the middle line must be used for corroded or tuberculated pipes, or conduits lined with pebbles, or pipes whose interior appears quite rough to the naked eye. The upper line for f in Fig. 15 gives values for moderately tuberculated iron pipe, which are also applicable to brine pipes in refrigerating systems, etc.

4. The critical region shown in Fig. 15 denotes a transition state from viscous flow to turbulent flow. Flow conditions in this region are not settled and may change from one state to the other owing to fluctuating flow, external disturbances, or changing condition of the interior of the pipe. It is safe practice, therefore, to employ in this region the higher values of f given by the upper line.

5. An upper limit of the friction factor for turbulent flow of a fluid in a rough conduit probably exists but has not been definitely determined. Pigott and Kemler [see footnotes (*i*) and (*j*) on page 107] who examined the results of more than 10,000 individual tests conclude that unless the roughness becomes of such an order of magnitude as to cause effects such as occur when there are contractions and enlargements, the friction factor will not become greater than 0.0135 using the scale for friction factor given in Fig. 15. For corrugated or creased pipe bends and corrugated straight pipe which are sometimes used to secure greater flexibility, f will probably lie between 0.010 and 0.0135.

Sample Problems.—The use of the rational formulas for friction loss in pipes described in the preceding paragraphs can best be illustrated by sample problems. The following are examples of a few typical cases:

Example 1. *Steam.*—A uniflow engine using 500 lb of steam per hour exhausts into a condenser through a standard-weight 6-in. pipe 25 ft long. There is one elbow in the line from the engine to the condenser. The pressure in the condenser is 1 psi abs. Quality of steam is 96 per cent. What is the friction pressure loss in the pipe?

Density of steam at 100 per cent quality is 0.003 lb per cu ft from steam tables on pages 230 to 243.

Density y at 96 per cent quality is $0.003 \div 0.96 = 0.00312$ lb per cu ft. $W = 500$ lb per hr.

Inside diameter d of a 6-in. standard-weight (Schedule 40) pipe = 6.065 in. from Table X, page 366, and $d^5 = 8{,}206$.

Temperature is 101.8 F from Steam Tables on page 236.

Viscosity $z = 0.0103$ centipoise (from Fig. 17*b*).

$$\frac{W}{dz} = \frac{500}{6.065 \times 0.0103} = 8{,}020.$$

Friction factor $f = 0.0062$ from Fig. 15*a*.

Elbow equivalent = 26 diameters = 13 ft from Fig. 15*a*.

Total equivalent length of straight pipe = 25 + 13 = 38 ft.

$$\text{Pressure drop } p_\lambda = \frac{\quad}{yd^5} = \frac{0.0062 \times 37.9 \times 500^2}{74{,}500 \times 0.00312 \times 8{,}206} = 0.031 \text{ psi.}$$

Check.—The pressure drop shown by the Fritzsche chart (Fig. 37) is 0.09 lb per 100 ft of pipe, or $0.09 \times 0.379 = 0.034$ psi for an equivalent length of 37.9 ft. The pressure drop shown by the Unwin chart (Fig. 36) is 0.07 lb per 100 ft of pipe, or $0.07 \times 0.379 = 0.027$ psi for 37.9 ft. The value obtained by the rational formula is seen to lie between those obtained by the Fritzsche and Unwin formulas. Probably the Fritzsche and Unwin formulas are not particularly accurate at this very low density, since they have to cover the entire range of steam pressures, temperatures, and velocities as well as pipe sizes with rather limited adjusting factors. The absence of a quality correction for moisture on the Unwin chart accounts for a part of the discrepancy.

The pressure drop involved in this problem is too small to necessitate the use of average values for pressure and density. From Table XIV, page 100, the equivalent length of one 90-deg standard screwed elbow is 15.2 ft, which in this case is within 15 per cent of that obtained from Fig. 15*a*. The following problem for a higher steam pressure shows a better agreement between the rational and empirical methods of solution.

Example 2. *Steam.*—What is the pressure drop in an equivalent length of 100 ft of 10-in. Schedule 40 pipe carrying 50,000 lb of dry saturated steam per hour at 250 psi abs? What is the straight pipe equivalent of one 90-deg flanged elbow?

Specific volume V is 1.843 cu ft per lb from steam tables on page 239, and the temperature is 401 F.

Inside diameter of pipe d is 10.02 in. from Table X on page 366, and d^5 is 101,000.

$$z = 0.021 \text{ centipoise (from Fig. 17}b\text{).}$$

$$\frac{W}{dz} = \frac{50,000}{10.02 \times 0.021} = 237,600$$

$$f = 0.004 \text{ and elbow equivalent} = 34 \text{ diameters, from Fig. 15}a.$$

$$p_\lambda = \frac{fLVW^2}{74,500d^5} = \frac{0.004 \times 100 \times 1.84 \times 50,000^2}{74,500 \times 101,000}$$

$$= 0.246 \text{ psi per 100 ft of pipe.}$$

Check.—The pressure drop shown by the Fritzsche chart (Fig. 37, page 251) is 0.20 lb per 100 ft of pipe. The pressure drop shown by the Unwin chart (Fig. 36, page 249) is 0.23 lb per 100 ft of pipe. This tends to support the statement of Gebhardt in "Steam Power Plant Engineering," 6th ed., page 752: "Babcock's, Spitzglass' and Fritzsche's coefficients give practically the same results for moderate rates of discharge and pipe diameters under 10 in., but for larger pipe diameters and high rates of discharge, Fritzsche's coefficient appears to give results more in accord with actual performance."

NOTE.—Unwin's and Carpenter's coefficients differ by approximately 1 per cent from that of Babcock mentioned above by Gebhardt.

The straight pipe equivalent from Table XIV, page 100, for a 10-in. flanged elbow is 18.7 ft. The value obtained from Fig. 15*a* for an elbow equivalent of 31 diameters is 19.4 ft, taking three-fourths the loss for a screwed elbow.

Example 3. *Water.*—A new 12-in. Schedule 40 pipe is delivering water at 100 F at the rate of 2,000 gpm. What is the friction pressure loss for each 100 ft of pipe and for each 90-deg elbow?

$$G = 2,000$$

$$y = 62.0 \text{ lb per cu ft (from Table XL, page 267).}$$

$$z = 0.70 \text{ centipoise (from Fig. 16, page 119).}$$

$$d = 11.938 \text{ in. (from Table X, page 366).}$$

$$d^5 = 242,470 \text{ (from Table X, page 366).}$$

$$\frac{Gy}{dz} = \frac{2,000 \times 62.0}{11.938 \times 0.70}$$

$$f = 0.0042 \text{ (from Fig. 15, page 108).}$$

$$= \frac{0.0042 \times 100 \times 62 \times (2,000)^2}{1,159 \times 242,470}$$

$$= 0.371 \text{ psi per 100 ft of pipe,}$$

From Fig. 15*a* the length of straight pipe equivalent to one 90-deg elbow is 31 pipe diameters or $31 \times 11.938/12 = 31$ ft. The press loss per elbow is $31 \times 0.371/100 = 0.1143$ psi.

Check.—From Fig. 39*b*, page 273, based on the Saph and Schoder formula the pressure drop per 100 ft of straight pipe is 0.398 psi. From Table XIV, page 100, the equivalent length of one standard flanged 90-deg elbow is 22.5 ft. The rational formula probably gives the more correct result, since it takes into account the variation in density and viscosity with temperature.

Example 4. *Oil.*—A certain oil at 70 F has a time of 9,050 sec in a Saybolt Universal viscometer and a gravity corresponding to 15.2° Bureau of Standards Bé. It is desired to transmit 20 U.S. gpm at a temperature of 70 F through a 6-in. standard-weight steel pipe line. What will be the pressure drop in pounds per square inch per mile?

Solution.—$G = 20$. To obtain the value of z/S corresponding to the above time, refer to Fig. 18 on page 123. Since the abscissas scale does not go above 2,000 sec, divide 9,050 by 10 and multiply the corresponding reading on the ordinate scale by 10. $z/S = 1{,}990$. Note that this method of extrapolation is permissible only with straight lines on log-log coordinates, or with the tangent portions of curved lines.

The specific gravity S at 70 F $= \dfrac{140}{130 + 15.2} = 0.964$.

6.065 in. (from Table X, page 366).

8.206 (from Table X, page 366).

$$\frac{GS}{dz} \quad \frac{G}{d} \times \frac{S}{z} = \frac{20}{6.065} \times \frac{1}{1{,}990} = 0.00166.$$

f 3.06 (from Fig. 15*a*, page 108, extrapolation made by dividing the abscissas scale by 10 and multiplying the ordinate scale by 10 as explained above). This shows the given conditions to be in the lower ranges of the viscous-flow region.

$$p_\lambda = \frac{\qquad}{18.6d^5} = \frac{3.06 \times 5{,}280 \times 0.964 \times 20^2}{18.6 \times 8{,}206}$$
$$= 40.8 \text{ psi per mile.}$$

Alternate Solution.—Since this problem is in the viscous flow region, it can be solved also by Poiseuille's formula (see page 106) $p_\lambda = 0.000273zLG/d^4$. From the relation $z/S = 1{,}990$ and the specific gravity S corrected to 70 F by tables VIII and IX, z is determined as 1,923 centipoises. Substituting in the formula

$$p_\lambda = \frac{0.000273 \times 1{,}923 \times 5{,}280 \times 20}{1{,}353} \quad 41 \text{ psi per mile.}$$

Example 5. *Oil.*—Turbulent flow is particularly desirable in a line carrying heated oil. What are the diameter of pipe and the minimum velocity with which oil should be pumped to ensure turbulent flow under the following conditions: $G = 200$ gpm, Bureau of Standard Bé when heated $= 28°$, Saybolt Universal time when heated $= 300$ sec?

Specific gravity $(S) = \dfrac{140}{130 + 28} = 0.886.$

$$\frac{z}{S} = 66 \text{ (from Fig. 18, page 123).}$$

Viscosity $(z) = 66S \qquad \times 0.886 = 57.2$ centipoises.

From Fig. 15*a*, $\frac{GS}{dz}$ must exceed 1.0 for turbulent flow.

Solving for d in $GS/dz = 1.0$.

$$d = \frac{GS}{z} = \frac{200}{66} = 3.03 \text{ in.}$$

Since the diameter should not exceed 3.03 in., 3-in. standard-weight (Schedule 40) pipe having an actual inside diameter of 3.068 in. is just too large. Therefore $2\frac{1}{2}$-in. standard-weight pipe having an inside diameter of 2.469 in. and a cross-sectional area of 4.788 sq. in. should be used. The corresponding velocity will be $v = (200 \times 231)/(4.788 \times 12 \times 60) = 13.4$ ft per sec. Or in case the rate of pumpage will slightly exceed 200 gpm, turbulent flow will be insured in the 3-in. pipe, at a considerably lower friction loss.

Example 6. *Air.*—Dry air at 80 F and an initial pressure of 150 psi abs is to be discharged through a pipe line with a pressure drop of not to exceed 10 psi. The pipe line is 200 ft long with three 90-deg elbows. What size standard-weight steel pipe will be required if the discharge is 1,000 cu ft of free air per minute measured under standard conditions of 60 F and 29.921-in. barometer?

The average pressure is $(150 + 140) \div 2 = 145$ lb absolute.

The density of air (y) under standard conditions is 0.07638 lb per cu ft (from Table XVIII, page 146).

Weight of discharge $= 0.07638 \times 1{,}000 = 76.38$ lb per min.

The volume at 145 psi abs and 80 F is found by the perfect gas equation page 144) as follows:

$$V = \frac{WRT}{p} = \frac{76.38 \times 0.371 \times (460 + 80)}{145} = \quad ; \text{ cfm.}$$

The average density $y = 76.38/105.5 = 0.724$ lb per cu ft in pipe.

It is now expedient to employ a cut and try calculation to determine the correct pipe diameter, allowing an average elbow equivalent of 30 pipe diameters. First assume a $1\frac{1}{2}$-in. pipe and solve for dvy/z:

Actual inside diameter $d = 1.61$ in. from Table X, page 366.
Actual inside pipe area $A = 0.01414$ sq ft, Table X, page 366.
Viscosity $z = 0.0187$ centipoises from Fig. 17*a*, page 120.

$$\text{Velocity } v = \frac{105.5}{60 \times 0.01414} = 124.3 \text{ ft per sec.}$$

$$\frac{dvy}{z} = \frac{1.61 \times 124.3 \times 0.724}{0.0187} = 7.748.$$

$$f = 0.0042 \text{ (from Fig. 15}a\text{).}$$

$$L = 200 + \frac{3 \times 30 \times 1.61}{12} = 212.1 \text{ ft.}$$

$$p_\lambda = \frac{fLyv^2}{193.2d} = \frac{0.0042 \times 212.1 \times 0.724 \times (124.3)^2}{193.2 \times 1.61}$$
$$= 32.0 \text{ psi.}$$

This is a higher pressure drop than desired. Repeat the above process for the next larger size pipe which is 2 in., and a pressure drop of 9.6 psi is obtained; 2-in. pipe is, therefore, the proper size to use.

Alternate Solution.—A direct solution for Example 6 can be made using the alternate method shown in Fig. 15*b* and Table XV. In this case the known factors are p_1, p_λ, Q, s, z, and T_c. Because of elbows in the line the equivalent

length L has to be estimated. The equations for "d unknown" for gas or air measured at 60 F and 14.7 psi abs in Table XV apply.

$$p_1 = 150 \quad p\lambda = 10 \quad p_2 = p_1 - p\lambda = 150 - 10 = 140$$
$$Q = 1{,}000 \times 60 = 60{,}000 \text{ cu ft per hr.}$$

The specific gravity of air at 60 F and 14.7 psi abs = 1.0.

$$L = 200 + \text{allowance for 3 elbows} = \text{say } 230 \text{ ft.}$$
$$z = 0.0187 \quad T_c = (460 + 80) = 540.$$

From Table XV,

$$N' = \frac{1{,}400Qs}{2T_c \sqrt[4]{LzQ/(p_1^2 - p_2^2)}} = \frac{1{,}400 \times 60{,}000 \times 1.0}{0.0187 \times 540 \sqrt[4]{\dfrac{230 \times 0.0187 \times 60{,}000}{22{,}500 - 19{,}600}}}$$
$$= 2{,}700{,}000.$$

From Fig. 15*b*, $C = 190$, and from Table XV

$$d = 0.178 \sqrt[4]{\frac{CLzQ}{p_1^2 - p_2^2}} = 0.178 \sqrt[4]{\frac{190 \times 230 \times 0.0187 \times 60{,}000}{22{,}500 - 19{,}600}}$$
$$= 2.03 \text{ in.}$$

The equivalent length for 2-in. elbows may then be checked (see Table XIV, page 100) and in this case the allowance was adequate. Therefore the 2-in. pipe size should be used.

Discussion of Sample Problems.—The sample problems given above show in general a fair agreement between the rational formula employed in Fig. 15 and the various empirical formulas used as checks. The greatest variations occurred in connection with elbow equivalents. It is believed that, in general, the results obtained from the rational formula are more nearly correct, since this formula takes into account all of the variables while the empirical formulas do not. The chief advantage of the various empirical formulas is ease of solution. Where a considerable degree of accuracy is desired, use of the rational formula is recommended.

FLUID FLOW IN PIPES OF ANNULAR CROSS SECTION

An investigation of the flow of air, oil, and water through pipes of annular cross section is reported by H. D. Atherton in the 1926 *Trans. ASME*, page 145. The results are presented in a form similar to that used above in the rational flow formulas.

MECHANICAL MIXTURES OF LIQUIDS

Temperature of Mechanical Mixtures.—Problems involving the resulting temperature and final condition when various substances at different temperatures are mixed mechanically are often

met with in engineering work. They are best treated by first determining the heat in Btu that would be available for use if the temperature of all of the substances were brought to 32 F, and using this heat (positive or negative) to raise (or lower) the total weight of the mixture to its final temperature and condition. Another method of solving such problems is by equating the heat absorbed to the heat rejected and solving for t, the resulting temperature. It is often difficult to decide upon which side of the equation a material should be placed. In such a case a trial calculation should be made, and the temperature determined by the trial will settle this question.

In a mixture of substances which pass through a change of state during the mixing process it is almost necessary to make a trial calculation. Take, for example, a mixture of steam with other substances. The steam all may be condensed and the resulting water cooled also; the steam all may be condensed only; or the steam may be only partially condensed. The equations in each case would be different.

If 1 lb of dry saturated steam at a temperature t_1 is condensed and then the temperature of the condensed steam is lowered to a temperature t_2, the amount of heat H' given off would be

$$H' = L_1 + c(t_1 - t_2),$$

where L_1 is the latent heat corresponding to the temperature t_1 and c is the specific heat of water. If the steam were condensed only, the heat given off would be

$$H' = L_1,$$

and the temperature of the mixture is the temperature corresponding to the pressure. If the steam is only partly condensed let q' equal the percentage of steam condensed. Then,

$$H' = \frac{q'L_1}{100}$$

and the temperature of the mixture is the temperature corresponding to the pressure.

The general laws of thermodynamics do not apply in the case of mixtures, as the equations become discontinuous.

The general expression for heat absorbed in passing from a solid to a gaseous state may be stated as follows:

Let c_1, c_2, c_3 be the specific heats of the material in the solid, liquid, and gaseous states, respectively. Let w be the weight

of the material, t the initial temperature, t_1 the temperature of the melting point, t_2 the temperature of the boiling point, t_3 the final temperature, H_f the heat of liquefaction, and L the heat of vaporization. Then,

$$H' = w[c_1(t_1 - t) + H_f \qquad\qquad - t_1) + L + c_3(t_3 - t_2)].$$

Example.—Find the final temperature and condition of the mixture after mixing 10 lb of ice at 20 F, 20 lb of water at 50 F, and 2 lb of steam at atmospheric pressure. Mixture takes place at the pressure of the steam. The specific heat of ice may be taken as 0.5 and the heat of liquefaction as 144 Btu (see page 267). The properties of saturated and superheated steam are given in the tables on pages 230 to 243.

Solution 1:

Heat to raise ice to 32 F = 10 × 0.5(32 − 20)	= 60.0.
Heat to melt ice = 10 × 144	= 1440.0.
Total heat necessary to change the ice to water at 32 F	= 1500.0 Btu
Heat given up by water when temperature is lowered to 32 F = 20 × (50 − 32)[1]	= 360.0.
Heat in steam above 32 F (from tables) = 1150.3 × 2	= 2300.6.
Total heat given up in lowering water and steam to 32 F	= 2660.6 Btu
Heat available for use = 2660.6 − 1500	= 1160.6 Btu
Degrees this heat will raise the mixture[1] 1160.6 ÷ 32	= 36.3

Final temperature of mixture = 36.3 + 32 = 68.3 F.

Answer.—32 lb water at 68.3 F.

Solution 2.—Assume that the steam is all condensed and that the final temperature of the mixture is t. Then the heat necessary to raise the ice to the melting points equals

$$10 \times 0.5(32 - 20).$$

The heat necessary to melt the ice equals 10 × 144; the heat necessary to raise the melted ice to the temperature of the mixture equals $10(t - 32)$; the heat necessary to raise the water to the temperature of the mixture equals $20(t - 50)$; the heat given up by the steam in changing to water at the temperature of the boiling point equals 2 × 970.4, and the heat given up by the condensed steam when its temperature is lowered to the temperature of the mixture equals $2(212 - t)$.

Combining the preceding parts into one equation, we have

$$10 \times 0.5(32 - 20) + (10 \times 144) + 10(t - 32) + 20(t - 50) = (2 \times 970.4) + 2(212 - t).$$

$$60 + 1{,}440 + 10t - 320 + 20t - 1{,}000 = 1{,}940.8 + 424 - 2t$$
$$32t = 2{,}184.8.$$
$$t = 68.3 \text{ F}.$$

[1] This calculation is made on the assumption that the specific heat of water is unity throughout the range between 32 and 212 F. In the case of mixtures at pressures above atmosphere, it is desirable to use steam-table values for heat of the liquid where extreme accuracy is required.

Since t is less than the temperature of the boiling point corresponding to the pressure at which the mixture takes place, all the steam is condensed.

Answer.—32 lb of water at 68.3 F.

DENSITY AND PROPORTIONS OF LIQUID MIXTURES

Relation of Density, Weight, and Volume in Solutions.—Let d, w, v, and d', w', v' represent the respective density, weight, and volume of either of two liquids, and D, W, and V the density, weight, and volume resulting from a mixture of the two.

$$d = \frac{w}{v} \quad d' = \frac{w'}{v'} \quad D = \frac{W}{V},$$

or

$$v = \frac{w}{d} \quad v' = \frac{w'}{d'} \quad V = \frac{W}{D}.$$

$$V = v + v' = \frac{w}{d} + \frac{w'}{d'} = \frac{wd' + w'd}{dd'}.$$

$$W = w + w' = dw + d'v'.$$

$$D = \frac{W}{V} = \frac{dv + d'v'}{\frac{wd' + w'd}{dd'}} = \frac{dd'\,(dv + d'v')}{wd' + w'd},$$

$$D = \frac{dd'(w + w')}{wd' + w'd},$$

also

$$D = \frac{W}{V} = \frac{dv + d'v'}{v + v'}.$$

The above formulas furnish a ready means of determining the density, weight, or volume of any mixture of two liquids, or if any two of these properties are known for both the mixture and one of its constituents, these properties of the other constituent can be determined.

Dilution to a Definite Specific Gravity.—To find the amount of water to add to a solution to reduce it to any desired specific gravity, deduct the desired specific gravity from that of the heavier liquid, the difference giving the number of volumes of water to be mixed with the number of volumes of the solution denoted by the fractional or decimal portion of the required specific gravity.

Example.—It is desired to prepare a solution having a specific gravity of 1.30 by adding water to a solution whose specific gravity is 1.70. Deducting 1.30 from 1.70 leaves 0.40, which means that 40 volumes of water must be added to 30 volumes of the heavier solution.

When specific gravities are less than that of water, they must first be subtracted from unity before proceeding. For instance, if a solution having a specific gravity of 0.90 is to be diluted with water to increase the specific gravity

to 0.98; subtracting 1.00 − 0.90 = 0.10; 1.00 − 0.98 = 0.02; then 0.10 − 0.02 = 0.08. Hence, 8 volumes of water are required to 2 of the original solution in order to produce a solution having a specific gravity of 0.98.

Rectangle Method for Diluting or Concentrating Solutions.—This is a graphical method for determining the proportions in which to mix two solutions of different concentrations in order to obtain a solution of any desired concentration. Both solutions must of necessity be comprised of either one or both of the same two liquids. The figures expressing the percentage concentration of the two solutions are written in the two left-hand corners of a rectangle, and the figure expressing the desired concentration is placed on the intersection of the two diagonals of this rectangle (see Fig. 23). Subtract the figure at the intersection of the diagonals from those at the left and write the results, irrespective

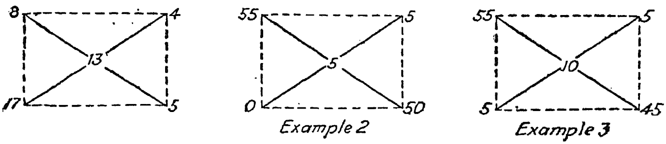


FIG. 23.—Rectangle method for diluting or concentrating solutions.

of whether plus or minus, at the right on the corresponding diagonals. The results indicate in what proportions, by weight, the solutions whose concentrations are given on the other ends of the respective horizontal lines must be taken to obtain a solution of the desired concentration.

Example 1.—It is desired to make a 13 per cent solution by mixing an 8 per cent and a 17 per cent solution in the proper proportions. Referring to the rectangle for Example 1, 5 parts by weight of the 17 per cent solution added to 4 parts by weight of the 8 per cent solution will give 9 parts by weight of 13 per cent solution. If a definite amount of 13 per cent solution, say 100 lb, is required, it will consist of $100 \times \frac{4}{9} = 44.5$ lb of the 8 per cent solution, and $100 \times \frac{5}{9} = 55.5$ lb of the 17 per cent solution. Check: $44.5 + 55.5 = 100$.

Example 2.—It is desired to dilute a 55 per cent solution by adding clear water so as to make a 5 per cent solution. Clear water may be considered a solution of 0 per cent concentration. Referring to the rectangle for Example 2, 5 parts by weight of 55 per cent solution added to 50 parts by weight of water give 55 parts by weight of 5 per cent solution. Any definite weight of 5 per cent solution can be prepared as explained under Example 1.

Example 3.—It is desired to dilute 200 lb of 55 per cent solution by adding the required amount of 5 per cent solution to obtain a 10 per cent solution. Referring to the rectangle for Example 3, 5 parts by weight of 55 per cent solution added to 45 parts by weight of 5 per cent solution will give 50 parts by weight of 10 per cent solution. But 5 parts by weight of 55 per cent solution must weigh

200 lb in this instance and one part by weight will weigh 200/5 = 40 lb. Hence, the 45 parts by weight of 5 per cent solution will weigh 45 × 40 = 1,800 lb. Therefore 1,800 lb of 5 per cent solution must be added to 200 lb of 55 per cent solution to obtain 2,000 lb of 10 per cent solution. Obviously, this method is readily applicable to determining the weight of pure water which must be added to any weight of concentrated solution in order to obtain a solution having some particular concentration.

PROPERTIES OF AIR

Composition.—The atmosphere of the earth is a mixture of several gases and vapors, the proportions of which vary somewhat in different localities and under different weather conditions. In general, the proportions of oxygen and nitrogen, the two most important constituents of dry air, are approximately as follows:

	By weight, percentage	By volume, percentage
Nitrogen	76.9	79.1
Oxygen	23.1	20.9
Approximate total	100.0	100.0

Carbon dioxide and water vapor are also contained in air in varying amounts, and there are, in addition, small quantities of other gases, such as argon, ozone, and neon which are of less importance and are not ordinarily considered in air calculations. In addition to the gases mentioned, a certain amount of dust particles is usually present in suspension. Air is not a chemical combination, but is a mechanical mixture of these gases. The properties of dry air under standard pressure conditions are given in Table XVIII.

LAWS OF GASES

Boyle's Law.—At constant temperature, the volume of a perfect gas is inversely proportional to the absolute pressure, or $p_1V_1 = p_2V_2$ in which

p_1 = initial absolute pressure, psi.
V_1 = initial volume, cu ft.
p_2 = final absolute pressure, psi.
V_2 = final volume, cu ft.

This law expresses the fact that if the pressure on a certain volume of gas is doubled, for instance, the volume will be one-half the original volume, provided the temperature remains constant.

Deviation of Natural Gas from Boyle's Law.—The pressure-volume relations of all gases deviate from the generally accepted law of Boyle as just defined. When pressures exceed about 50 psi, this deviation is enough to warrant consideration in gas measurement. An imaginary gas that conforms to Boyle's law is considered an *ideal* gas. Many common gases conform approximately to Boyle's law at ordinary ranges of pressure and temperature, but no real gas follows it exactly at higher pressures and temperatures. The deviation is manifest in that a given amount of gas at low pressure will compress into less volume at some higher pressure than would be expected from the "perfect gas" equation. This is designated as *supercompressibility* which is the reciprocal of the *superexpansibility* manifest if the gas is reexpanded to the original conditions.

In the case of natural gas the supercompressibility, depending on the constituents of the gas, may be of the order of 7 to 12 per cent in compressing from atmosphere to 500 psi and perhaps 15 to 25 per cent between atmosphere and 1,000 psi. The extent of deviation depends on the relative quantities of the various hydrocarbons and other gases constituting the natural gas. The deviations of these individual components differ widely from each other so that no one set of definite values can be assigned which will apply strictly to all natural gases; hence values may need to be determined for each natural gas. For instance, a natural gas that contains much ethane and propane will have a larger percentage deviation than one consisting mostly of methane. With gas having the same initial and final temperatures, the extent of deviation tends to decrease with increase in temperature. Thus raising the temperature base 20 F may reduce the supercompressibility at 500 psi by about 1 per cent, and at 1,000 psi by about 2 per cent.

The deviation of natural gas from Boyle's law has been investigated by the Bureau of Mines[1] and others,[2] and much has

[1] (*a*) "Compressibility of Natural Gas and Its Constituents, with Analyses of Natural Gas from 31 Cities in the United States," by G. A. Burrell and I. W. Robertson, *Tech. Paper* 158, Bureau of Mines, 1917.

(*b*) "Deviation of Natural Gas from Boyle's Law," by T. W. Johnson and W. B. Berwald, *Tech. Paper* 539, Bureau of Mines, 1932.

[2] (*a*) "Compressibilities of Gases," by S. F. Pickering, *Misc. Pub.* 71, Bureau of Standards, November, 1925.

(*b*) "Air Apparatus and Method for Determining the Compressibility of a Gas and the Correction for Supercompressibility," by H. S. Bean, *Jour. Research*, Vol. 4, No. 5, Bureau of Standards, May, 1930.

been written on the subject.[1] Rather intricate natural laws are involved and considerable kinematics and thermodynamics, all of which enter into the design of compressing stations as well as pipe lines. Space limitations do not permit treating the subject here in full detail, and those interested should refer to the references listed.

Charles's Law.—At constant volume, the pressure of a perfect gas is directly proportional to the absolute temperature; or at constant pressure the volume is directly proportional to the absolute temperature. This is expressed in symbols as follows:

$$\frac{p_1}{T_1} = \frac{p_2}{T_2} \quad \text{and} \quad \frac{V_1}{T_1} = \frac{V_2}{T_2}$$

in which T_1 and T_2 are the initial and final absolute temperatures, respectively. [The absolute temperature in degrees Fahrenheit is $(460 + t)$ where t is the observed temperature in degrees Fahrenheit.]

Equation of a Perfect Gas.—The equation of a perfect gas is obtained by combining the laws of Boyle and Charles. The equation for 1 lb of the gas is $p_1V_1/T_1 = p_2V_2/T_2 = R$, in which R is a constant for each gas. For a perfect gas, R is inversely proportional to the molecular weight of the gas. The numerical value is given by the relation $R = 10.72/m$, where m is the molecular weight of the gas.

NOTE.—Where P is expressed in pounds per square foot, $R = 1{,}544/m$.

The molecular weights of various gases are given in Table XXIII, page 166. The equation for 1 lb of gas is usually written in the form $pV = RT$, in which the subscripts are dropped to denote any simultaneous conditions of pressure, volume, and temperature. The equation for any number of pounds of gas is written $pV = WRT$ where W is the weight in pounds of the quantity of gas which has a pressure, volume, and absolute temperature of p, V, and T, respectively.

Considering dry air as a perfect gas, its constant R, as determined from the molecular weight relation, is 0.371 which checks with

[1] (a) "Deviations of Natural Gas from Ideal Gas Laws," by George Granger Brown, pamphlet published 1940 by Clark Bros. Co., Inc., Olean, N.Y.

(b) "Air and Gas Compression," by T. T. Gill, John Wiley & Sons, Inc., New York, 1941.

(c) "An Analysis of Gas-pipe-line Economics," by H. C. Lehn, *Trans.* ASME Vol. 65, No. 5, pp. 445–460, July, 1943. See Appendix 1 for instructions on how to include the effect of supercompressibility in the general flow formulas.

the values obtained by experiment. The equation for 1 lb of dry air then becomes $pV = 0.371T$, and the equation for any number of pounds $pV = 0.371WT$.

NOTE.—Where P is expressed in pounds per square foot, $R = 53.37$ ft-lb.

In a similar way, the equation for density of dry air under any conditions is obtained by taking V as equal to 1 cu ft and solving for W as follows: $W = p/0.371T = 2.6982p/T$. Or for the case where B represents absolute pressure in inches of mercury (or barometric reading): $W = 1.3253B/T$. In the formulas of this paragraph, W represents the density of dry air in pounds per cubic foot. The densities of dry air at atmospheric pressure and various temperatures given in Table XVIII were calculated by these formulas. "A Table of Thermodynamic Properties of Air," which gives the thermal properties of dry air in a form similar to the steam tables, has been worked out by Keenan and Kaye.[1]

Joule's Law.—When a perfect gas expands doing no external work, the temperature remains constant. For example, in the equation $p_1V_1/T_1 = p_2V_2/T_2$, if $T_1 = T_2$, the equation becomes $p_1V_1 = p_2V_2$. This is identical with the equation of Boyle's law.

The Two Specific Heats of Air.—The specific heat of any substance may be defined as the amount of heat in Btu required to change the temperature of 1 lb of the substance 1 F. Owing to the fact that air is an expansive gas, it has two specific heats which must be considered: (1) the specific heat at constant pressure; and (2) the specific heat at constant volume.

When a gas is heated at constant pressure, its volume increases against that pressure, and external work is done in consequence. The external work may be computed by multiplying the pressure by change in volume. When heated at constant volume, no external work is done, *i.e.*, no movement is made against an external resistance. If the gas be perfect, no disintegration work is done, and the specific heat at constant volume is a *true* specific heat. The specific heat at constant pressure is, however, the one more commonly encountered in engineering work. The numerical values of the two specific heats, in a perfect gas, must differ by the heat equivalent to the external work done during heating at constant pressure. Numerical values for the two specific heats of air are given in the next two paragraphs.

[1] See ASME *Jour. of Applied Mechanics*, Vol. 10, No. 3, pp. *A*-123 to *A*-130, September, 1943.

TABLE XVIII.—PROPERTIES OF DRY AIR UNDER STANDARD PRESSURE CONDITIONS

(One atmosphere = 14.696 psi = 29.921 in. Hg)

Temperature	Density	Volume of 1 lb.	Percentage of volume at 70° F.	Instantaneous specific heat[1]	B.t.u. absorbed by 1 cu. ft. per degree Fahrenheit	Temperature	Density	Volume of 1 lb.	Percentage of volume at 70° F.	Instantaneous specific heat[1]	B.t.u absorbed by 1 cu. ft. per degree Fahrenheit
° F.	Pounds per cu. ft.	Cu. ft. per pound	Percentage	B.t.u. per pound per ° F.	B.t.u. per cu. ft. per ° F.	° F.	Pounds per cu. ft.	Cu. ft. per pound	Percentage	B.t.u. per pound per ° F.	B.t.u. per cu.ft. per ° F.
0	0.08636	11.579	86.760	0.2411	0.0208	210	0.05926	16.877	126.46	0.2430	0.0144
5	0.08542	11.707	87.719	0.2412	0.0206	220	0.05838	17.129	128.35	0.2431	0.0142
10	0.08451	11.833	88.663	0.2412	0.0204	230	0.05754	17.382	130.24	0.2432	0.0140
15	0.08362	11.959	89.607	0.2413	0.0202	240	0.05671	17.633	132.12	0.2433	0.0138
20	0.08275	12.085	90.551	0.2413	0.0200	250	0.05591	17.885	134.01	0.2433	0.0136
25	0.08189	12.212	91.503	0.2413	0.0198	260	0.05514	18.136	135.89	0.2435	0.0134
30	0.08108	12.330	92.387	0.2414	0.0196	270	0.05438	18.390	137.79	0.2436	0.0132
35	0.08024	12.463	93.383	0.2414	0.0194	280	0.05365	18.642	139.68	0.2436	0.0131
40	0.07944	12.582	94.275	0.2415	0.0192	290	0.05293	18.896	141.58	0.2437	0.0129
45	0.07865	12.715	95.271	0.2415	0.0190	300	0.05223	19.148	143.46	0.2438	0.0127
50	0.07788	12.840	96.208	0.2416	0.0188	325	0.05057	19.778	148.19	0.2440	0.0123
55	0.07712	12.967	97.160	0.2416	0.0186	350	0.04900	20.408	152.91	0.2443	0.0120
60	0.07638	13.094	98.111	0.2417	0.0185	375	0.04755	21.030	157.80	0.2445	0.0116
65	0.07584	13.184	98.793	0.2417	0.0183	400	0.04617	[illegible]	162.36	0.2447	0.0113
70	0.07493	13.346	100.00	0.2418	0.0181	425	0.04483	22.301	167.10	0.2449	0.0110
75	0.07423	13.472	100.94	0.2418	0.0179	450	0.04361	22.930	171.81	0.2452	0.0107
80	0.07354	13.598	101.89	0.2418	0.0178	475	0.04245	23.557	176.51	0.2454	0.0104
85	0.07287	13.725	102.84	0.2419	0.0176	500	0.04134	24.190	181.25	0.2456	0.0102
90	0.07221	13.849	103.77	0.2419	0.0175	525	0.04029	24.820	185.97	0.2459	0.0099
95	0.07155	13.976	104.72	0.2420	0.0173	550	0.03929	25.452	190.71	0.2461	0.0097
100	0.07091	14.102	105.66	0.2420	0.0172	575	0.03834	26.085	195.45	0.2463	0.0094
110	0.06967	14.355	107.56	0.2421	0.0169	600	0.03744	26.709	200.12	0.2465	0.0092
120	0.06846	14.607	109.45	0.2422	0.0166	650	0.03575	27.972	209.59	0.2469	0.0088
130	0.06730	14.859	111.34	0.2423	0.0163	700	0.03421	29.231	219.02	0.2474	0.0085
140	0.06618	15.110	113.22	0.2424	0.0160	800	0.03149	31.756	237.94	0.2483	0.0078
150	0.06509	15.363	115.11	0.2425	0.0158	900	0.02917	34.282	256.87	0.2492	0.0073
160	0.06404	15.615	117.00	0.2426	0.0155	1000	0.02718	36.805	275.77	0.2501	0.0068
170	0.06302	15.868	118.89	0.2427	0.0153	1500	0.02024	49.407	370.29	0.2560	0.0052
180	0.06204	16.118	120.77	0.2427	0.0151	2000	0.01612	62.034	464.81	0.2656	0.0043
190	0.06108	16.371	122.67	0.2428	0.0148	2500	0.01340	74.701	559.72	0.2773	0.0037
200	0.06016	16.622	124.55	0.2429	0.0146	3000	0.01145	87.260	653.83	0.2926	0.0034

[1] See explanation on p. 145.

$$\text{density} = \frac{1.3253B}{459.2 + t} = \frac{2.6983p}{459.2 + t}.$$

Where density = pounds per cubic foot. B = absolute pressure in inches Hg. p = absolute pressure in pounds per square inch. t = temperature in degrees Fahrenheit.

Specific Heat of Air at Constant Pressure.—The specific heat of dry air at constant pressure was given by Renault as 0.2375. More recent investigators have given it a slightly higher value.

The specific heat of dry air at constant pressure was given by F. G. Swann as $0.24112 + 0.000009t$ and the specific heat of water vapor as $0.4423 + 0.00018t$, where t is the observed temperature in degrees Fahrenheit. The specific heat of moist air with any degree of saturation may then be found by multiplying the weight of dry air by its specific heat and adding to this product the weight of water vapor times its specific heat, and dividing the sum by the weight of the mixture. The specific heat C_p of dry air at constant pressure and at high temperatures is given by Pye (see "Mechanical Engineers' Handbook," by Lionel S. Marks, 2d. ed., p. 378) as

$$C_p = \frac{6.93 \times 10^6 + 0.1254T^2}{2,895 \times 10^4}$$

where T is the absolute temperature in degrees Fahrenheit. The specific heat at constant pressure, as computed by the above two formulas, coincides at 1307.9 F. Above this temperature, Pye's values are higher than Swann's, while below it the reverse condition obtains. In compiling the data on dry air given in Table XVIII, Swann's values for specific heat have been used up to 1307.9 F, and Pye's values at higher temperatures. The variation of the specific heat of dry air at constant pressure for different ranges of pressure within the temperature limits of 68 to 212 F is given by Holborn and Jacob (*ZVDI*, Vol. 58, p. 1436) as:

Pressure, psi abs.	14.2	356	711	1,422	2,133	2,844.
C_p.............	0.2415	0.2490	0.2554	0.2690	0.2821	0.2925.

For most engineering calculations within the ordinary range of pressures and temperatures, the specific heat of dry air at constant pressure may be taken as 0.2415. Those wishing more exact information can refer also to "The Specific Heats of Certain Gases over Wide Ranges of Pressures and Temperatures," by F. O. Ellenwood, Nicholas Kulick, and Norman R. Gay,[1] or to the "ASHVE Guide."

Specific Heat of Air at Constant Volume.—The specific heat of dry air at constant volume is less than the specific heat at constant pressure by an amount representing the external work done in the constant pressure case. The heat equivalent of this external work may be computed as follows:

Let w represent the external work in foot-pounds when the temperature of 1 lb of air is raised 1 F at constant pressure.

[1] *Cornell Univ. Bull.* 30, October, 1942, Engineering Experiment Station, Ithaca, N.Y.

The external work w is equal to the pressure times the change in volume. Using the subscript 0 to denote initial conditions and the subscript 1 for final conditions:

$$w = P(V_2 - V_1). \quad \text{From Charles's law } \frac{V_1}{V_2} = \frac{T_1}{T_2}$$

and $\quad V_1 = V_2 \dfrac{T_1}{T_2}.$

Substituting in the first equation,

$$w = P\left(V_2 - V_2 \frac{T_1}{T_2}\right) = \frac{PV_2}{T_2}(T_2 - T_1) = \frac{PV_2}{T_2}$$

since, in this case, $(T_1 - T_0) = 1$, and dropping the subscripts,

$$w = PV/T.$$

In a previous paragraph on the "Laws of Gases" it was shown that for dry air $PV/T = R = 53.37$ ft-lb. Hence, $w = R = 53.37$ ft-lb or, converting into heat units, $w = 53.37/778 = 0.0686$ Btu. The specific heat at constant volume is then 0.0686 Btu less than the corresponding specific heat at constant pressure. If the specific heat at constant pressure is taken as 0.2415, the specific heat at constant volume is $0.2415 - 0.0686$ or 0.1729 Btu per lb.

Ratio of Specific Heats.—The numerical ratio between the two specific heats of a sensibly perfect gas, denoted by the symbol k and referred to as the adiabatic exponent, is of prime importance in thermodynamics. Some of the applications of this ratio are used on pages 73 to 80 in connection with the critical pressure in nozzles and with the flow of gases and steam through orifices and nozzles. It appears again on pages 254 to 264 in the discussion of fluid flow at the accoustic velocity.

The numerical value of k for air under ordinary conditions is determined as $k = 0.2415/0.1729 = 1.40$. Values of k for several common gases and vapors are given in Table XXIII, page 166. For others, the value of k may be computed as the ratio of specific heat at constant pressure to the specific heat at constant volume. For saturated and superheated steam the values of k are 1.135 and 1.30, respectively.

Water Vapor.—Water vapor is an important constituent of the atmosphere. It is the most variable in quantity of all the atmospheric elements, its amount depending largely on weather conditions. In the northern part of the United States the moisture content of the atmosphere is very great. In New York, for example, it varies from 0.5 to 7 grains per cu ft. Water vapor, strictly speaking, is nothing other than steam at very low pressures,

and *its properties are identical with those of steam.* This fact should always be borne in mind when dealing with the subject of atmospheric moisture. Another conception that should be thoroughly understood is that of *Dalton's law of partial pressures.* According to this law, in any mixture of gases, each gas has a partial pressure of its own which is entirely independent of the partial pressures of the other gases.

The above points may be illustrated by the following specific problem: Assume 1 cu ft of saturated air at 70 F and atmospheric pressure of 14.6963 psi abs. The partial vapor pressure corresponding to a temperature of 70 F (refer to steam tables on page 231) is 0.3626 psi, and the weight of vapor per cubic foot is 0.001148 lb. The partial air pressure is $14.6963 - 0.3626 = 14.3337$ psi. From Boyle's law the weight of dry air present in 1 cu ft of the mixture then, is $0.07493 \times \frac{4.333}{4.696}$ 0.07308, where 0.07493 lb is the weight of 1 cu ft of dry air at 14.6963 psi abs and 70 F, see Table XVIII. The weight of 1 cu ft of saturated air under these conditions, then, is the sum of the weights of dry air and water vapor present, or $0.001148 + 0.07308 = 0.07423$ lb. The weight of vapor water per pound of dry air is $0.001148 \div 0.07308 = 0.01571$ lb. The weight of water vapor present in moist air is frequently expressed in grains. There are 7,000 grains in 1 lb av, so in this instance the number of grains of moisture present in 1 cu ft of saturated air is $0.001148 \times 7{,}000 = 8.0360$ grains. To complete the illustration, the following values were read from the tables named as a check against the above calculations for conditions of 70 F and 14.6963 psi abs (29.921 in. of mercury absolute):

	Calculated value	Value from tables	
		Table or figure number	Value
Vapor pressure, pounds per square inch absolute	0.3626	Steam tables,	0.3626
Grains of moisture per cubic foot of mixture.....	8.036	Fig. 26	8.00
Pounds of vapor per cubic foot of mixture.......	0.001148	Fig. 26	0.001143[1]
Pounds of vapor per pound of dry air...........	0.01571	Fig. 24	0.0157
Weight of 1 cu. ft. of mixture, pounds..........	0.07423	Fig. 24	0.07387[2]
Pounds of dry air in 1 cu. ft. of mixture.........	0.07308	By subtraction	0.07273

[1] Grains of moisture ÷ 7,000.

[2] Calculated from data on Fig. 24 as follows: Part by weight of water = 0.0157/1.0157 = 0.01546; weight of 1 cu ft of mixture is $0.001143 \div 0.01546 = 0.07387$ lb.

The partial vapor pressure and the corresponding relative weights of water vapor and air can be calculated readily for any degree of saturation in a way similar to that outlined above. In the above problem if the air had been 70 per cent saturated, the weight of water vapor per cubic foot would have been 0.001148 $\times$ 0.7 = 0.0008036 lb and the partial vapor pressure 0.3626 $\times$ 0.7 = 0.2538 psi. The method of calculating the other quantities then follows directly from the example for saturated air.

For every temperature there is a corresponding partial pressure of water vapor at which the vapor is in a *saturated* state, its condition then being exactly similar to that of saturated steam, *i.e.*, with the maximum number of molecules occupying a unit space. When the water vapor is in a saturated condition, the air is also spoken of as being saturated, since it then contains the maximum weight of vapor which it can hold at that temperature. If the temperature of the air is higher than that corresponding to the partial pressure of the water vapor, the vapor is *superheated*. If the temperature drops below the saturation point, some of the vapor is condensed and the vapor pressure is lowered to that corresponding to the new temperature. The saturation temperature is termed the "dew point." The partial pressure of saturated vapor increases as the temperature increases. Consequently, air at higher temperatures is capable of holding a greater weight of water per cubic foot. It should be remembered that water vapor exists independently of the air, except for the temperature effect of the latter, and the vapor may be thought of as occupying the given volume at its own partial pressure. The state of intimate mixture of the air and vapor causes their temperatures to be always the same.

Moisture in Compressed Air.—The presence of moisture in compressed air is undesirable both because of pipe-line troubles and excessive wear in air-operated tools caused by washing out the lubricating oil. Other troubles develop in paint-spraying equipment and through frosting produced by sudden expansion of the air. Hence the use of aftercoolers in connection with air compressors is recommended as a means of moisture removal.[1]

Effect of Change in Barometric Pressure.—Suppose that air in which the partial vapor pressure is e_o is compressed isothermally at a temperature t from a barometric pressure B_o to a barometric

[1] See "Compressed Air Data," by F. W. O'Neil, published by *Compressed Air Magazine*, New York.

pressure B. Then the partial pressures of both air and vapor are increased proportionally and the new partial vapor pressure is $e = e_o B/B_o$. The temperature corresponding to saturation at e is the temperature of the dew point at pressure B.

Relative and Absolute Humidity.—Atmospheric moisture is termed "humidity." Absolute humidity is the actual vapor content expressed in grains per cubic foot or per pound of air. The ratio of the vapor content to the vapor content of saturated air at the same temperature, expressed in percentage, is called the "relative humidity." For example, given a sample of air at 70 F having an absolute humidity of 4 grains per cu ft. Since saturated air at 70 F contains 8 grains per cu ft, the relative humidity is 50 per cent. Pounds of water vapor per pound of dry air corresponding to various *relative humidities* are shown in Fig. 24.

Total Heat of Air.—The total heat above 0 F of air containing aqueous vapor is the sum of the heat of the air and the heat of the vapor. The latter has three components: the heat of the liquid, the heat of vaporization, and the superheat. The vapor is always in a superheated condition, unless the air is at the saturation point.

In dealing with air containing vapor it is often convenient to use the units of weight instead of volume as a basis for calculations. The total heat above 0° in 1 lb of dry air at temperature t_a is equal to

$$H = C_{pa}(t_a - 0)$$

in which t_a is the air temperature and $C_{pa} = 0.2415$, the specific heat of air at constant pressure.

Let W_w = the weight of water vapor contained in 1 lb of a mixture of air and water vapor. Then for saturated atmosphere

$$H = (1 - W_w) \times C_{pa}(t_a - 0) + W_w(h' + r)$$

in which h' = heat of the liquid above 0° for the water vapor

and r = latent heat of the water vapor.

For atmosphere below saturation (and, therefore, containing superheated vapor) at temperature t_a

$$H = [(1 - W_w) \times C_{pa}(t_a - 0)] + W_w[h' + r + C'_{ps}(t_a - t_d)]$$

in which t_d is the temperature at the dew point and C'_{ps} is the specific heat of water vapor at constant pressure.

For a more vigorous treatment of the thermal properties of moist air, see the "ASHVE Guide."

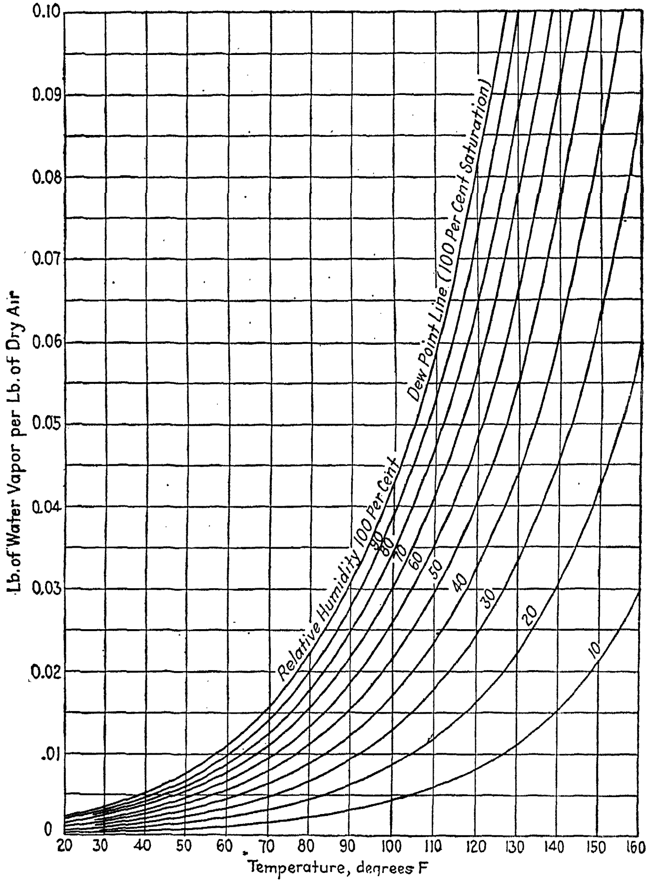


Fig. 24.—Pounds of water vapor per pound of dry air at various relative humidities. (Barometric pressure 29.921 in.)

Adiabatic Saturation.—When air below saturation is brought into intimate contact with water, there is always a tendency for some of the water to vaporize, adding to the moisture content

of the air. If no heat is added from an outside source and none removed, the heat of vaporization for the moisture which is added will be supplied entirely at the expense of the heat of the air and of the superheat of the original quantity of water vapor. The process will continue until the saturation point is reached. A process of this nature taking place without a transfer of heat to or from an outside source is called "adiabatic" and the final temperature which is reached is, therefore, termed the "temperature of adiabatic saturation" or "wet-bulb temperature." Its depression below the original temperature of the air will depend upon the amount of moisture which was added to bring the air to saturation. If the air is saturated, no moisture can be added, and the wet-bulb and dry-bulb temperatures coincide.

The heat used in the vaporization of the moisture which was added is exactly equal to the heat given up by the air and by the water vapor which it contained originally, assuming that the water which was added was at the temperature of adiabatic saturation. The action may be expressed algebraically as follows:[1]

Let t = temperature of the air, degrees F.

t' = temperature of adiabatic saturation, degrees F.

W' = weight of water vapor mixed with 1 lb of dry air at saturation at temperature t'.

W = weight of water vapor mixed with 1 lb dry air at temperature t.

$W' - W$ = weight of water added per lb of dry air.

r = latent heat of vaporization at temperature t.

C_{ps} = specific heat of water vapor at constant pressure.

C_{pa} = specific heat of dry air at constant pressure.

$$(W' - W)r = C_{ps}W(t - t') + C_{pa}(t - t'), \tag{1}$$

or

$$W = \frac{rW' - C_{pa}(t - t')}{r + C_{ps}(t - t')}. \tag{2}$$

Measurement of Humidity.—The principle stated in the preceding paragraph affords a convenient means for measuring humidity, through the use of the wet- and dry-bulb thermometer. The instrument consists of two mercury thermometers, the bulb of one of which is covered with cotton wicking. The end of the wicking extends into a bottle of water and the entire length is kept wet by absorption. As the water is evaporated from the

[1] From "Rational Psychrometric Formulae," by W. H. Carrier, *Trans. ASME*, 1911.

wicking, its temperature is lowered to the temperature of adiabatic saturation or "wet-bulb" temperature. By reading both thermometers when they have reached a constant point, the wet-bulb depression is obtained and the moisture content of the air W can be found from equation (2), above.

Distinction should be drawn between the wet-bulb temperature and the dew point, which was defined previously. The former temperature is produced by adding moisture to the air and causing its temperature to drop by reason of the giving up of heat to vaporize the water. The dew point, on the other hand, is reached by removing heat from the air without changing its moisture content. In order to obtain accurate results with a wet-bulb thermometer, it is necessary that the air surrounding the wet bulb be in motion, so that the maximum evaporation may be secured. For this reason, the best form of wet- and dry-bulb thermometer is the sling psychrometer illustrated in Fig. 25. In this instrument, the wet- and dry-bulb thermometers are mounted on a metal strip pivoted to a handle. In using the instrument, the wick surrounding the wet bulb is moistened and the instrument is whirled rapidly and read at intervals until there is no further drop in the wet-bulb temperature. Somewhat more accurate results are obtained with the aspiration psychrometer in which a continuous current of air is drawn over the wet-bulb thermometer by means of a small fan driven by clockwork.

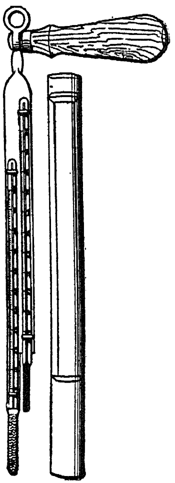


FIG. 25.—Sling psychrometer.

It is necessary that the water used to moisten the wet bulb of the sling psychrometer be at approximately the wet-bulb temperature; otherwise the time required to bring the water to the wet-bulb temperature might be so great that parts of the wicking would become dry.

The psychrometric chart in Fig. 26 is constructed for use with the sling psychrometer.[1] This chart gives the moisture content of air in grains per cubic foot. If the weight of water vapor per pound of dry air is wanted, it can be calculated readily, as explained in the example on page 149 from the grains per cubic foot value.

[1] From "Fan Engineering," Buffalo Forge Company.

Example of Use of Psychrometric Chart.—Given a dry-bulb temperature of 80 F and a wet-bulb temperature of 70 F, find the relative and absolute humidity and the dew point. From the

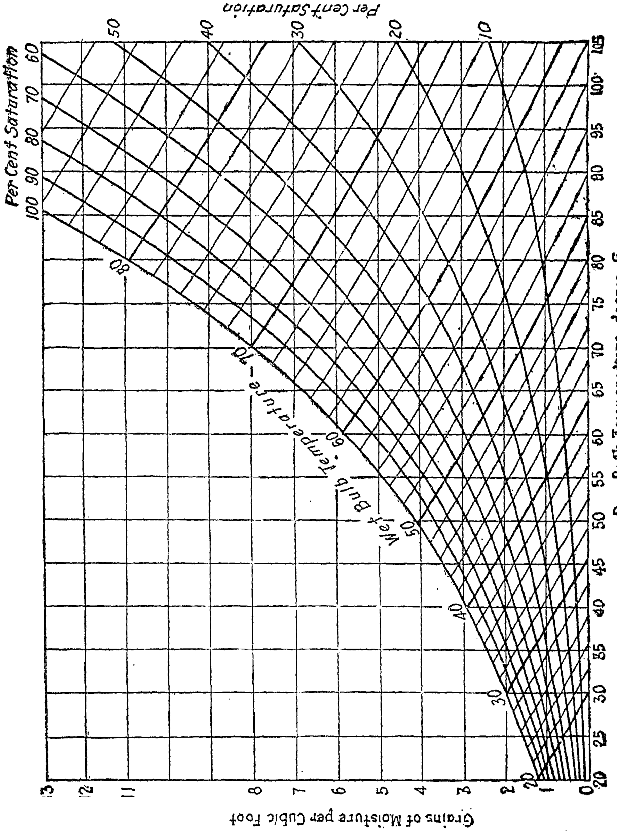


Fig. 26.—Psychrometric chart.

80 F point on the horizontal scale follow the vertical line to its intersection with the diagonal line representing the wet-bulb temperature of 70 F. Passing horizontally to the left from this point to the left-hand scale, we find that the absolute humidity is

6.65 grains per cu ft. To find the relative humidity we note that this same point lies between the 60 and 70 per cent relative humidity lines (the curved lines extending upward to the right) and that the relative humidity is 62 per cent. To find the dew point, follow left horizontally from this same point to the curved line of wet-bulb temperatures, called the "saturation line." The dew point is 64.5 F.

GASES

The fundamental laws of gases are discussed under the subject of air, starting on page 142. The data pertaining directly to air are given in that connection, while the additional information required to deal with the usual gases of engineering work is furnished in this section. The most common gases encountered by the engineer are those used for heating, illuminating, operating gas engines, and similar purposes. The ordinary varieties of these are coal gas, water gas and carbureted water gas, natural gas, oil gas, blast furnace gas, producer gas, and coke-oven gas. Those commonly distributed for domestic consumption are coal gas, carbureted water gas, oil gas of the artificial varieties, and natural gas.

Common Fuel and Illuminating Gases.—The name "illuminating gas," used at one time to designate the common commercial product, is descriptive of the use for which gas was originally intended. For a great many years after the first commercial production of illuminating gas, it was used for no other purpose than that which its name implies. From an economy standpoint it was out of the question to consider it as a fuel at that time. Manufacturing methods were crude and expensive, necessitating a high price for gas, while wood, the prevailing fuel, was plentiful and cheap. The flat-flame burner, which was the only gas lamp known at that time, produced light through combustion of the illuminating constituents of the gas. Then came the incandescent gas lamp of the mantle type, of which the Welsbach burner is an example. The incandescent lamp consumes gas in the same manner as a range or water-heater burner, depending on the heat generated in a Bunsen-type burner to heat a refractory mantle to incandescence. In a Bunsen (or atmospheric burner) a certain proportion of air is drawn in and mixed with the gas before combustion to produce a blue, nonluminous flame. Such a flame contains none of the incandescent carbon particles which furnish

the greater part of the light produced by a luminous flame. In a Bunsen flame the hydrocarbons are burned before they can decompose, owing to the intermingling of air and gas before combustion sets in, and, as a consequence, no luminous carbon particles are present. Such a burner will not smoke or deposit soot upon the mantle of an incandescent lamp or a cooking vessel, as would happen if a luminous flame were used, and combustion is more likely to be complete. The amount of heat produced by the consumption of equal amounts of gas is practically the same whether the gas be burned in a luminous or Bunsen burner, provided complete combustion is obtained in both cases.

The *illuminating power* of a gas comes from the hydrocarbons, called "illuminants," contained in the gas. The luminosity of a flame depends on the presence of solid particles, the density of the burning gas, and its temperature. The solid particles are very finely divided carbon which are heated to incandescence before they meet sufficient air for their combustion. These carbon particles are produced by the decomposition of hydrocarbons in the gas while in the center of the flame and out of contact with air, but subject to the temperature of combustion on the surface of the flame. In general the more dense the gases undergoing combustion, the more luminous they are when heated to high temperatures, and anything which tends to raise the temperature of the flame increases the luminosity, other things being equal. Smoke and soot are produced by the escape of unburned particles of carbon. The standard of illuminating value in general use is the sperm candle of a size running six to the pound and burning at the rate of 120 grains an hour. The illuminating power of a gas is expressed by stating the number of times a flame burning at the rate of 5 cu ft of gas per hr is greater than the standard candle.

The *heating value* of a gas is expressed in Btu per cubic foot. In the case of hydrogen and substances containing hydrogen, a distinction must be drawn between gross or total heating value, and net or available heating value. The complete combustion of such substances produces water which must be evaporated into steam at the expense of a part of the heat produced in combustion. It is customary in all ordinary work to express quantities of heat in terms of gross value, unless otherwise stated. The higher heating value is determined by cooling the products of combustion to the original temperature of air and gas. Since the amount of heat rejected by the water vapor depends on the temperature

to which the cooling is carried, it is possible that various interpretations of the lower heating value may be made. Lucke and Flather state ("Text Book of Engineering Thermodynamics") that it is sufficiently close for engineering work to accept, as the difference between the high and low heat values, 970.4 times the weight of water vapor formed per fuel unit burned.

NOTE.—The latent heat of water vapor at atmospheric pressure is 970.4 Btu per lb, and each pound of hydrogen burned will form 9 lb of water vapor; hence

$$\begin{aligned}\text{High heat value} - \text{low heat value} &= 970.4 \times (\text{weight of water vapor formed})\\ &= 9 \times 970.4 \times (\text{weight of hydrogen per fuel unit}).\end{aligned}$$

See also the method for computing moisture loss given in connection with Table XXI.

Standard Conditions.—The standard cubic foot of artificial gas is measured at a temperature of 60 F when under an absolute pressure of 30 in. of mercury column at 32 F which is equivalent to 30.028 in., or 14.73 psi abs, at 60 F. The gas is considered as saturated with water vapor at 60 F. It should be noted that these standard conditions for artificial gas differ slightly from those commonly used in the natural-gas industry given on page 163. Owing to this discrepancy and to a desire to use the same standard conditions wherever possible throughout this handbook in formulas for the flow of air as well as artificial and natural gas, a mean condition of 14.7 psi abs at 60 F has been employed.

The *density* of a gas is usually expressed as the specific gravity relative to air under the same conditions of temperature and pressure. The specific gravities relative to air of the more common individual gases are given in Table XXIII, while those for the commercial heating and illuminating gases are given in Table XXI. Specific gravities relative to water at 39 F of some of the common individual gases are shown in Fig. 19 over a considerable temperature range.

Coal gas is produced by the destructive distillation of bituminous coal, at high temperature, in a retort or oven externally heated. Coal gas is usually the most profitably manufactured of the artificial gases, owing to the credit obtained from the sale of by-products and residual coke. Nine to ten thousand cubic feet of gas are obtained per ton of coal distilled. Owing to municipal or state regulation of heating value and candlepower, it is generally necessary to employ some method of enrichment whereby these qualities are increased. The most common form of enrichment

is by the addition of a portion of carbureted water gas which brings the mixed gas to the proper quality to meet these requirements. Two typical percentage compositions of coal gas by volume are:

Carbon dioxide (CO_2)............	1.60	2.71[1]
Oxygen (O)......................	0.39	0.78
Benzine vapor (C_6H_6)............	0.50	4.51 illuminants
Heavy hydrocarbons (C_2H_4).......	4.25	
Carbon monoxide (CO)...........	8.04	8.38
Hydrogen (H_2)...................	47.04	47.81
Methane (CH_4)...................	36.02	32.00
Nitrogen..........................	2.16	3.81
Total..........................	100.00	100.00

[1] From Bureau of Mines, *Bull.* 6.

The average heating value runs from 500 to 600 Btu per cu ft and the illuminating value about 15 candles. Owing to its large percentage of hydrogen, coal gas has a low specific gravity of approximately 0.45 (air = 1).

Water gas is produced by the decomposition of steam by contact and union with carbon at a high temperature. *Carbureted water gas* is made from water gas by adding to and mixing with a certain proportion of oil gases, the richness of the carbureted gas depending on the proportions in which the water and oil gases are mixed. In this way a wide range of luminosity and heating values may be obtained, which make it a simple and flexible means of meeting quality requirements. Water gas is usually made from gas coke, although anthracite also can be used for this purpose. Good operating results should show a fuel consumption of 30 to 35 lb of coke per 1,000 cu ft of gas made. Uncarbureted water gas has a calorific value of only about 300 Btu per cu ft and no illuminating value. It may be used for lighting, however, by causing it to heat to incandescence some solid substance, as a Welsbach or other incandescent mantle. An illuminating gas having a higher calorific value is made from water gas by adding to it hydrocarbon gases or vapors which are obtained from crude petroleum or its distillate known as "gas oil." The gas so made should be called "carbureted water gas," but it is ordinarily known in the United States simply as "water gas." Gas oil is most commonly used for this purpose. The results obtained from an oil depend primarily upon its specific gravity and composition,

TABLE XIX.—PROPERTIES OF HYDROCARBONS FOUND IN NATURAL GAS AND CASING-HEAD GAS[1]

	Methane	Ethane	Propane	Butane	Pentane	Hexane	Heptane	Octane
Formula	CH_4	C_2H_6	C_3H_8	C_4H_{10}	C_5H_{12}	C_6H_{14}	C_7H_{16}	C_8H_{18}
Molecular weight	16.03	30.05	44.07	58.08	72.10	86.12	100.13	114.15
Specific gravity of liquid	....	0.432 = 194° Bé	0.515 = 142° Bé	0.585 = 109° Bé	0.630 = 92.2° Bé	0.670 = 78.9° Bé	0.697 = 70.9° Bé	0.718 = 65.0° Bé
Specific gravity of gas	0.555	1.049	1.526	2.008	2.496	2.982	3.467	3.952
Boiling point at atmospheric pressure	−165 C −265 F	−93 C −135 F	−45 C −49 F	+1 C 34 F	36.3 C 97 F	69 C 156 F	98.4 C 200 F	125.5°C 258 F
Pressure to liquefy at 60 F, psi	....	475	105	35	6.5	1.8	0.5	0.15
Vapor pressure 70 F in percentage of atmosphere	100+	100+	100+	100+	55	10	2.7	0.7
Gallons per 1,000 cu ft at B.P. reduced to 60 F	....	22.13	27.01	31.28	36.13	40.56	45.34	50.11
Weight 1,000 cu ft vapor at B.P. reduced to 60 F, lb	42	79.7	116	152.6	189.7	226.6	263.5	300
Shrinkage in volume by 1 gal liquid removed per 1,000 cu ft	....	....	....	....	2.8 per cent	2.5 per cent	2.2 per cent	2.0 per cent
Maximum possible removable gal. per 1,000 cu ft at 70 F gal	....	....	....	....	19.87	4.06	1.22	0.35
High heat value, Btu per cu ft	1,065	1,861	2,685	3,447	4,250	5,012	5,780	6,542
Btu per lb	25,360	23,350	23,150	22,590	22,400	22,120	21,935	21,807
Cubic feet air to burn 1 cu ft gas	9.57	16.72	23.92	31.10	38.28	46.46	53.6	60.8
Carbon percentage	75.0	80.0	81.8	82.8	83.3	83.7	84.0	84.2
Explosive mixture percentage in air, Maximum	14.5	5.0	3.5	3.0	2.5	2.2	1.9	1.6
Minimum	5.6	3.0	2.1	1.6	1.3			

[1] Reproduced by permission from *Bull.* 25 of the Kansas City Testing Laboratory.

but an average production of 1,000 cu ft from 3 gals of oil is considered good practice. The illuminating value of carbureted water gas may run as high as 20 candles, and the heating value 640 Btu per cu ft, although in actual practice these values are reduced to meet the local requirements. It is common practice in many plants to make both coal and water gas and mix them in the proper proportions to meet the local requirements. A typical composition of carbureted water gas by volume is given in Table XXI.

Natural Gas.—Pure natural gas is odorless and colorless, burns with a luminous flame, and is highly explosive when mixed with air. Its chief constituent is marsh gas, or methane, a member of the paraffin series. Natural gas is classified as either "wet" or "dry," according to its content of gasoline. Wet gas contains not only ethane, propane, butane, and pentane, the lighter members of the methane series which predominate in the dry gas, but some heavier hydrocarbons. Dry gas contains chiefly methane, the lighest known hydrocarbon, which has a specific gravity of 0.559. Natural gas usually occurs in connection with petroleum, but it is also found in places far removed from oil fields. A considerable amount of casing-head gasoline is obtained from natural gas by compression, refrigeration, or absorption methods. The composition of natural gas varies considerably in different fields. Typical compositions by volumes of the natural gas used in eight cities of the United States are as follows:

City	Methane, percentage CH_4	Ethane, percentage C_2H_6	Nitrogen, percentage N_2
Pittsburgh, Pa......	79.2	19.6	1.2
Louisville, Ky......	77.8	20.4	1.8
Buffalo, N. Y.......	79.9	15.2	4.9
Cincinnati, Ohio....	79.8	19.5	0.7
Cleveland, Ohio.....	80.5	18.2	1.3
Springfield, Ohio....	80.3	14.7	5.0
Columbus, Ohio	80.4	18.1	1.5
Ch[illegible], Okla.	75.4	17.7	6.6
Detroit, Mich......	74.0	14.5	11.2

These analyses were made by the ordinary combustion method, and, hence, show only the two predominating paraffin hydrocarbons.

The usual high heat value is from 800 to 1,100 Btu per cu ft, (see Table XXI), and the specific gravity (relative to air) 0.60 to

TABLE XX.—ABOUT NATURAL GAS AND ITS USEFULNESS

(An average sample of natural gas has 950 Btu per cu ft)

1 lb mill coal will evaporate 9 lb water.
1 gal oil will evaporate 100 lb water.
1 cu ft gas will evaporate 0.85 lb water.
1 ton coal used under boilers = 18,500 cu ft of gas.
1 bbl oil (42 gal) under boilers = 5,000 cu ft of gas.
40 to 50 cu ft of gas per hr = 1 boiler hp.

Gas engines:
Highest grade gas engines develop 1 brake hp on 8,500 Btu per hr.
Average large engine develops 1 hp on 10,500 Btu per hr.
Oil-well engine develops 1 hp on 20,000 Btu per hr.
In a small steam turbine plant 20 cu ft gas per kw-hr is a fair average.
It requires 40,000 cu ft of gas to pump 1,000,000 gal of water against 200-ft head.

Brick plants—gas used per thousand brick made:
1,800 cu ft for power.
1,800 cu ft for drying.
15,000 cu ft for kilns.

Ice plants:
2,000 ft gas per ton of refrigeration.

Zinc plants:
15,000 cu ft for roasting per ton of metal produced.
65,000 cu ft for smelting per ton of metal produced.
20,000 cu ft for power and miscellaneous uses per ton of metal produced.

Cement plants:
60 to 100 cu ft per bbl for power.
80 to 100 cu ft per bbl for roasters.
1,800 to 2,600 cu ft per bbl for kilns.

Salt plants:
Direct-fire pans, 9,000 cu ft per ton.
Steam pans, 10,000 cu ft per ton.
Single-effect vacuum pan, 15,000 cu ft per ton.
Double-effect vacuum pan, 10,000 cu ft per ton.
Triple-effect vacuum pan, 6,000 cu ft per ton.

Flour mills:
200 to 400 cu ft per bbl.

Gas compressors:
Horsepower required to compress 1,000 cu ft of gas per min:

To 15 lb	50 hp
To 30 lb	85 hp
To 45 lb	111 hp
To 60 lb	134 hp
To 80 lb	117 hp (2 stages)
To 100 lb	151 hp (2 stages)
To 200 lb	212 hp (2 stages)

Horsepower required to compress 1,000 cu ft of gas per hr:

To 15 lb	1 hp
To 30 lb	1.75 hp
To 45 lb	2.25 hp
To 60 lb	2.75 hp

0.70. In exceptional cases the specific gravity may run as high as 0.8 or 1.0 and the heating value up to 1,500 Btu per cu ft. Such gases are usually rich in casing-head gasoline. Properties of the constituents of natural gas are given in Table XIX. No generally accepted set of standard conditions has been adopted in the natural-gas industry, although temperature of 60 F and pressure ¼ lb or 4 oz above an assumed mean atmospheric pressure of 14.4 psi is used to a considerable extent. This is equivalent to 520 F abs and 14.65 psi abs.

References for the properties, production, and distribution of natural gas will be found in "Measurement, Compression, and Transmission of Natural Gas," by Lester Clyde Lichty, John Wiley & Sons, Inc., New York; "Problems in Natural Gas Engineering," by Thomas R. Weymouth, *Trans. ASME*, Vol. 34, 1912; "Gas Engineers' Handbook" of the Pacific Coast Gas Association, McGraw-Hill Book Company, Inc., New York; "American Gas Practice, Vol. 2, Distribution and Utilization of City Gas," by Jerome J. Morgan, published by Jerome J. Morgan, Maplewood, N.J. Considerable interesting data regarding the amount of natural gas required to accomplish different tasks are given in Table XX which is reproduced, by permission, from *Bull.* 25 of the Kansas City Testing Laboratory.

Oil Gas.—This is the least used of the artificial gases under consideration. Strictly speaking, it is the gas resulting from the destructive distillation of oil (liquid hydrocarbons). Pintsch gas, which is used in coach lighting, and Blau gas (liquid), which is used for domestic purposes, are the principal kinds of pure-oil gas; but under the heading of oil gas must be included that manufactured and used in California and other coast states, which is no small industry. Pacific coast oil gas is made by a process similar to the manufacture of carbureted water gas, except that oil instead of solid fuel is used to heat the apparatus. In Pintsch and Blau gases the high-heat value ranges from 1,350 to 1,500 Btu and the illuminating value from 50 to 60 candles. California oil gas has an average heating value of 680 Btu and an illuminating value of 20 candles. It is made from crude oil obtained in the fields close to the point of manufacture, so that a low price prevails. Gas coal is not found in the Pacific states, and oil gas is, of necessity, the prevailing kind. A large quantity of lampblack is produced in the manufacture of California oil gas, which is made into briquets, constituting a valuable by-product.

TABLE XXI.—TYPICAL COMPOSITIONS AND PROPERTIES OF FUEL AND ILLUMINATING GASES

	Percentage composition by volume										Density,[2] lb per cu ft	Specific gravity (air = 1)	Specific heat at constant pressure		High-heat value[3]	
	Hydrogen, H_2	Methane, CH_4	Ethane, C_2H_6	Illuminants,[1] C_nH_{2n}	Propane, C_3H_8	Butane, C_4H_{10}	Carbon monoxide, CO	Carbon dioxide, CO_2	Nitrogen, N_2	Oxygen, O_2			Btu per lb per F	Btu per cu ft per F[2]	Btu per lb	Btu per cu ft[2]
Specific heat in Btu per lb per degree F at constant pressure	3.42	0.593	0.400	0.400	0.390	0.390	0.243	0.210	0.247	0.217						
Oil gas	32.0	48.0	.....	16.5	.....	.....	.....	.....	3.0	0.5	0.03666	0.48	0.63	0.023	23,077	846
Refinery gas:																
Dubbs	5.0	33.0	15.0	33.0	14.0	.....	.....	.....	.....	.....	0.07332	0.96	0.44	0.040	20,458	1,500
Houdrie	....	28.6	17.5	0.6	33.4	19.9	.....	.....	.....	.....	0.11533	1.51	0.41	0.047	17,342	2,000
Natural gas		87.0	9.6	.....	.....	.....	.....	0.4	3.0	.....	0.04659	0.61	0.54	0.025	23,932	1,115
		74.0	14.5	.....	.....	.....	.....	0.1	11.2	0.2	0.05230	0.685	0.38	0.020	19,311	1,010
		.....	79.4	.....	20.0	.....	.....	.....	0.6	.....	0.08604	1.13	0.40	0.034	21,513	1,851
Pintsch gas	12.6	45.4	.....	35.7	.....	.....	0.6	0.7	3.0	2.0	0.06416	0.84	0.47	0.031	23,379	1,500
Blast-furnace gas	2.0	2.0	.....	.....	.....	.....	26.0	12.0	58.0	.....	0.07638	1.00	0.25	0.019	1,309	100
Producer gas, coal	12.0	2.6	.....	0.4	.....	.....	29.0	4.0	52.0	.....	0.06645	0.87	0.28	0.019	2,453	163
Coke-oven has	50.0	36.0	.....	4.0	.....	.....	6.0	1.5	2.0	0.5	0.02902	0.38	0.73	0.021	20,779	603
Coal-retort gas	52.5	31.4	.....	2.2	.....	.....	8.6	1.5	3.5	0.3	0.03208	0.42	0.70	0.021	17,924	575
Blue water gas	51.3	.....	.....	.....	.....	.....	43.4	3.5	1.3	0.5	0.04048	0.53	0.46	0.018	7,411	300
Carbureted water gas	35.6	16.5	.....	14.6	.....	.....	28.0	3.8	1.0	0.5	0.04888	0.64	0.46	0.023	11,457	560

[1] Includes ethylene C_2H_4, propylene C_3H_6, etc.

[2] For gas measured under standard conditions, see "Piping Handbook," pp. 158, 163.

[3] Owing to the varying hydrogen content of different types of gas the low-heat value will vary considerably, depending also on the flue gas temperature. *Correction for loss due to water vapor* can be made by the following formula (see ASME Power Test Code for Stationary Steam Generating Units, and "Computation of Heat in Moisture," by C. H. Berry, *Power*, Vol. 61, Mar. 17, 1925, p. 410):

$$\text{Heat loss in Btu per fuel unit} = 9H_2\,(1{,}066 + 0.5tg - ta), \text{ when } tg \text{ is more than 550F}$$
$$= 9H_2\,(1{,}089 + 0.46tg - ta), \text{ when } tg \text{ is less than 550 F}$$

where H_2 = weight of hydrogen in the fuel unit (lb H_2 per lb of gas, or lb H_2 per cu ft of gas as the case may be).

tg = temperature of the flue gas, degrees F.

ta = temperature of the atmosphere, degrees F.

For low heat values of various gases at different stack temperatures see Table XXII.

Composition, Heating Value, and Physical Constants.—These properties for common fuel and illuminating gases will be found in Table XXI. Because of the varying hydrogen content of different types of gas, the realizable heating value may differ considerably. The available heat, Btu per cubic foot, for given flue-gas temperatures when burning different types of gas under theoretically perfect conditions is given in Table XXII.

TABLE XXII.—AVAILABLE HEAT (LOW HEAT VALUE) BTU PER CUBIC FOOT WHEN BURNING DIFFERENT GASES WITH COMPLETE COMBUSTION AND THE THEORETICAL AMOUNT OF AIR

	Natural	Natural gas	Coke-oven gas	Coke-oven gas	Carbureted water gas	Blue water gas
High heat value..............	1232	967	600	490	534	310
Available heat (low-heat value)						
Flue temperature, 300 F.....	1050	825	510	420	450	270
Flue temperature, 400 F.....	1025	800	500	405	440	260
Flue temperature, 600 F.....	970	760	475	380	420	245

The properties of *other miscellaneous gases* commonly encountered will be found in Table XXIII.

FLOW OF GAS AND AIR IN PIPES

BASIC FLOW FORMULAS

In the general discussion of friction loss in pipes (see pages 81 to 87, with symbols defined on page 82) it was shown in equation (51), the basic formula for volume flow through round pipes, that

$$p_\lambda = \frac{174fyLF^2}{d^5} \quad \text{or} \quad F = 0.0757\sqrt{\frac{p_\lambda d^5}{fyL}}.$$

It is assumed that the pipe is horizontal and that there is no transmission of heat to or from the gas or air during flow. In the foregoing equation, F is expressed in cubic feet per *second.*

In dealing with *compressed air*, however, it is customary to measure flow in cubic feet per *minute* of either compressed air, or free air under standard conditions corresponding to a 29.921-in. barometer, *i.e.*, atmospheric pressure of 14.696 psi abs, and at a temperature of 60 F. For convenience, atmospheric pressure is taken as 14.7 psi abs. The symbol Q_a is used in this section to

TABLE XXIII.—PROPERTIES OF MISCELLANEOUS GASES[1]

Gas	Chemical symbol	Number of atoms	Molecular weight, Approximate	Molecular weight, Exact, $O_2 = 32$	Weight in pounds of 1 cu ft at atmospheric pressure, At 62 F	Weight in pounds of 1 cu ft at atmospheric pressure, At 32 F	Specific gravity (air = 1)	Gas constant, R, Note[2]	Specific heat, Btu per lb per deg F, c_p	Specific heat, Btu per lb per deg F, c_v	Specific heat, Btu per cu ft per deg F at atmos. pres. and 62 F, c_p	Specific heat, Btu per cu ft per deg F at atmos. pres. and 62 F, c_v	$k = \frac{c_p}{c_v}$
Helium	He	1	4.0	4.0	0.0105	0.0112	0.137	386.0	1.25	0.75	0.0131	0.0079	1.66
Argon	Ar	1	40.0	39.9	0.1048	0.1112	1.378	38.70	0.124	0.075	0.0131	0.0079	1.66
Air	....	..	29.0	28.95	0.0761	0.0807	1	53.34	0.241	0.171	0.0183	0.0130	1.40
Oxygen	O_2	2	32.0	32	0.0840	0.0892	1.105	48.25	0.217	0.155	0.0182	0.0130	1.40
Nitrogen	N_2	2	28.0	28.02	0.0737	0.0783	0.970	54.99	0.247	0.176	0.0182	0.0130	1.40
Hydrogen	H_2	2	2.0	2.016	0.00529	0.00562	0.0696	765.86	3.42	2.44	0.0181	0.0129	1.40
Nitric oxide	NO	2	30.0	30.04	0.0789	0.0838	1.038	51.40	0.231	0.165	0.0183	0.0130	1.40
Carbon monoxide	CO	2	28.0	28.00	0.0734	0.0780	0.968	55.14	0.243	0.172	0.0180	0.0126	1.41
Hydrochloric acid	HCl	2	36.5	36.45	0.0958	0.1017	1.260	42.35	0.191	0.136	0.0183	0.0130	1.40
Carbon dioxide	CO_2	3	44.0	44.00	0.1156	0.1227	1.520	35.09	0.210	0.160	0.0243	0.0185	1.31
Nitrous oxide	N_2O	3	44.0	44.03	0.1157	0.1229	1.522	35.03	0.221	0.171	0.0256	0.0198	1.26
Sulphur dioxide	SO_2	3	64.0	64.06	0.1684	0.1786	2.213	24.10	0.154	0.123	0.0260	0.0207	1.25
Ammonia	NH_3	4	17.0	17.06	0.04483	0.0476	0.590	90.50	0.523	0.399	0.0234	0.0178	1.31
Acetylene	C_2H_2	4	26.0	26.02	0.0684	0.0725	0.899	59.34	0.350	0.270	0.024	0.0185	1.28
Methyl chloride	CH_3Cl	5	50.5	50.47	0.1326	0.1407	1.744	30.59	0.24	0.20	0.032	0.0265	1.20
Methane	CH_4	5	16.0	16.03	0.0421	0.0447	0.554	96.31	0.593	0.450	0.025	0.019	1.32
Ethylene	C_2H_4	6	28.0	28.03	0.0738	0.0780	0.969	55.08	0.40	0.33	0.029	0.024	1.20

[1] Reproduced, by permission, from Marks's "Mechanical Engineers' Handbook."

[2] For use in perfect gas equation where P is given in pounds per square foot. If p is given in pounds per square inch, divide these values by 144.

denote cubic feet per *minute* of free (atmospheric) air or gas, while Q_c is used to designate cubic feet per *minute* of compressed air or gas under designated conditions. The symbols y_a and y_c are used to designate the density in pounds per cubic feet of free and compressed dry air, respectively, and s, the specific gravity of dry air at 14.7 psi abs and 60 F, is taken as unity and used as a base for gas-flow computations.

In dealing with *gas* it is customary to express flow in cubic feet per *hour* under standard conditions of measurement, which also approximate 14.7 psi abs at 60 F. Standard conditions for manufactured gas differ slightly from those for natural gas (see pages 158 and 163), the former being slightly above 14.7 psi abs and the latter slightly below, whereas air is taken at 14.696 psi abs. The differences, however, are so slight that they well can be ignored in computing pipe-line flows. Accordingly the tables and flow constants given in this handbook have been computed for "standard conditions" of 14.7 psi abs at 60 F so that the one set of values will serve for air and for both natural and manufactured gas. The symbol Q is used in this section to denote cubic feet per *hour* of free gas measured under standard conditions.

The *specific gravity* of a gas with respect to dry air under standard conditions (approximately 14.7 psi abs at 60 F) is designated as s. The corresponding density of the gas is $y = sy_a$. For any flow conditions other than standard, the corresponding density of the gas can be determined from its specific gravity under standard conditions and the density of air at the respective flow conditions as follows: $y = sy_c$. The density of *air* for any conditions can be determined from Boyle's law and Table X, or from the equation of a perfect gas, see page 144. Or the density of the *gas* for any conditions can be determined directly from the equation of a perfect gas if its gas constant or molecular weight is known.

The relation between the *volumes* of compressed and free air or gas is obtained from the perfect gas equation (see page 144), from which

$$Q_0 = Q_c \frac{p_c}{p_0} \times \frac{T_0}{T} \tag{58a}$$

and

$$p_c V_c = \frac{p_c}{y_c} = 0.371 T_c. \tag{58b}$$

If Q_0 is measured under atmospheric standard conditions denoted by subscript a, then $p_a = 14.7$ and $T_a = 460 + 60 = 520$, and

$Q_a = Q_c \times \frac{p_c}{14.7} \times \frac{520}{T_c}$ and in case Q_c is measured at 60 F, the last term becomes unity and

$$Q_a = Q_c \frac{p_c}{14.7}. \tag{58c}$$

The *weight* of air or gas flow M in pounds per minute can be computed from the volume flow as follows:

$$M = sy_aQ_a = sy_cQ_c. \tag{58d}$$

From Table XVIII, page 146, the density y_a of dry air under standard conditions is 0.07638 lb per cu ft. Substituting in the above equation and noting that $s = 1$ for air,

$$M = s0.07638Q_a = s0.07638Q_c \times \frac{p_c}{14.7} \times \frac{520}{T_c} = \frac{2.7sp_cQ_c}{T_c}. \tag{58e}$$

Using air as a base, the weight also can be determined directly from the perfect gas equation, $pV = WRT$ (see page 144), by substituting Q_c for V, and M for W, giving R its numerical value of 0.371, and solving for M,

$$M = \frac{sp_cQ_c}{0.371T_c} = \frac{2.7sp_cQ_c}{T_c}. \tag{58f}$$

The flow Q_c of *compressed* air or *high*-pressure gas in *cubic feet per minute* instead of in cubic feet per *second* F can be determined by substituting $Q_c/60$ for F in equation (51) and transposing as follows:

$$p_\lambda = \frac{174fyLF^2}{d^5} = \frac{174fsy_cL(Q_c/60)^2}{d^5} = \frac{fsy_cLQ_c^2}{20.7d^5}, \tag{59a}$$

$$Q_c = 4.55\sqrt{\frac{p_\lambda d^5}{fsy_cL}}. \tag{59b}$$

The *weight* M of air or gas in *pounds per minute* can be determined by substituting $Q_c = M/sy_c$ in equation (59) as follows:

$$p_\lambda = \frac{fsy_cL}{20.7d^5} \times \left(\frac{M}{sy_c}\right)^2 = \frac{fLM^2}{20.7sy_cd^5}, \tag{60a}$$

$$M = 4.55\sqrt{\frac{sy_cp_\lambda d^5}{fL}} \tag{60b}$$

where y_c is the density for air corresponding to average absolute pressure and temperature conditions in the line.

In dealing with the flow through pipes of expansive gases, including air, where there is any significant pressure drop, it is

necessary to take into account the change in density with its resulting effect on volume and velocity. Where flow is expressed in cubic feet of compressed air or gas, as in equation (59) or (60), this can be done to best advantage by assigning a value to y, or its equivalent sy_c, which corresponds to the average pressure in the line under conditions of maximum flow.

If flow is expressed in terms of weight, or of volume of *free* air or *low*-pressure gas, however, the effect of pressure drop can be taken into account conveniently through substituting the following relations in the flow formula to represent average conditions in the line. If p_1 is the initial pressure and p_2 the final pressure in the line, both expressed in pounds per square inch absolute, then the average flow pressure is $p_c = (p_1 + p_2)/2$, and the pressure drop is $p_\lambda = p_1 - p_2$. The fact that flow is being expressed in cubic feet measured under standard conditions makes it possible to effect further simplifications through introducing numerical values for p_a and y_c.

Where flow is expressed in *cubic feet per minute* of *free* air or *low*-pressure gas measured under *any* standard conditions designated by the subscript 0, the following substitutions from equation (58) can be made in equation (59*a*). Substituting for Q_c,

$$p_\lambda = \frac{fsy_cL}{20.7d^5} \times \left(\frac{Q_0 p_0 T_c}{p_c T_0}\right)^2,$$

and substituting $y_c/p_c = 1/0.371T_c$,

$$p_\lambda = \frac{fsLp_0^2T_c^2Q_0^2}{20.7 \times 0.371T_c p_c T_0^2 d^5} = \frac{p_0^2}{T_0^2} \times \frac{fsT_cLQ_0^2}{7.68p_c d^5} \qquad (61a)$$

and

$$Q_0 = 2.77\frac{T_0}{p_0}\sqrt{\frac{p_c p_\lambda d^5}{fsT_cL}}, \qquad (61b)$$

but $p_c = (p_1 + p_2)/2$ and $p_\lambda = p_1 - p_2$, hence

$$Q_0 = 1.96\frac{T_0}{p_0}\sqrt{\frac{(p_1^2 - p_2^2)d^5}{fsT_cL}} \qquad (61c)$$

$$d = \left[\frac{fsT_cL}{3.84(p_1^2 - p_2^2)} \times \left(\frac{p_0Q_0}{T_0}\right)^2\right]^{1/5}. \qquad (61d)$$

Where *standard conditions* are those usual for compressed air or gas (see page 167), which approximate 14.7 psi abs at 60 F, and the flow temperature also is assumed to be 60 F, a considerable simplification can be made through substituting the following numerical values in equation (61): $p_0 = p_a = 14.7$; $T_0 = T_a = T_c$

$= 460 + 60 = 520$; and Q_a = cu ft of gas or air per *minute* measured under *standard* conditions, from which

$$p_\lambda = \frac{fsLQ_a^2}{18.5p_c d^5}, \tag{62a}$$

$$Q_a = 4.3\sqrt{\frac{p_c p_\lambda d^5}{fsL}}, \tag{62b}$$

and substituting for p_c and p_λ as before,

$$Q_a = 3.04\sqrt{\frac{(p_1^2 - p_2^2)d^5}{fsL}}, \tag{62c}$$

$$d = \left[\frac{fsLQ_a^2}{9.24(p_1^2 - p_2^2)}\right]^{1/5}. \tag{62d}$$

The *weight M* of gas or air in *pounds per minute* can be expressed in terms of free air or gas by substituting in equation (60).

$$y_c = \frac{p_c}{p_a} y_a = \frac{p_c}{14.7} \times 0.07638, \text{ from which}$$

$$p_\lambda = \frac{fLM^2}{20.7s\left(\frac{0.07638p_c}{14.7}\right)d^5} = \frac{9.3fLM^2}{sp_c d^5}, \tag{63a}$$

$$M = 0.328\sqrt{\frac{sp_c p_\lambda d^5}{fL}}, \tag{63b}$$

and substituting for p_c and p_λ as before,

$$M = 0.232\sqrt{\frac{(p_1^2 - p_2^2)sd^5}{fL}}. \tag{63c}$$

Where flow is expressed as *cubic feet* per *hour* of air or gas measured under *any* standard conditions designated by the subscript 0, the following adjustments can be made directly from equation (61) using the relation that $Q_0 = Q_{60/60}$

$$p_\lambda = \frac{p_0^2}{T_0^2} \times \frac{fsT_cLQ_{60}^2}{3{,}600 \times 7.68p_c d^5} = \frac{p_0^2}{T_0^2} \times \frac{fsT_cLQ_{60}^2}{27{,}650p_c d^5} \tag{64a}$$

$$Q_{60} = 166\frac{T_0}{p_0}\sqrt{\frac{p_c p_\lambda d^5}{fsT_cL}}, \tag{64b}$$

$$Q_{60} = 117.5\frac{T_0}{p_0}\sqrt{\frac{(p_1^2 - p_2^2)d^5}{fsT_cL}}, \tag{64c}$$

$$d = \left[\frac{fsT_cL}{13{,}800(p_1^2 - p_2^2)} \times \left(\frac{p_0Q_{60}}{T_0}\right)^2\right]^{1/5}. \tag{64d}$$

Where flow is expressed in *cubic feet* per *hour* of *free* air or *low-pressure* gas measured under standard conditions approximating 14.7 psi abs and 60 F with a flow temperature of 60 F the following adjustments can be made directly from equations (62):

$$p_\lambda = \frac{fsLQ^2}{3{,}600 \times 18.5 p_c d^5} = \frac{fsLQ^2}{66{,}500 p_c d^5}, \tag{65a}$$

$$Q = 258 \sqrt{\frac{p_c p_\lambda d^5}{fsL}}, \tag{65b}$$

and

$$Q = 258 \sqrt{\frac{(p_1 + p_2)(p_1 - p_2)d^5}{2fsL}} = 182 \sqrt{\frac{(p_1^2 - p_2^2)d^5}{fsL}}, \tag{65c}$$

$$d = \left[\frac{fsLQ^2}{33{,}100(p_1^2 - p_2^2)}\right]^{1/5}. \tag{65d}$$

Where flow is expressed in *pounds* per hour, $W = 60M$ can be substituted in equation (63) as follows:

$$p_\lambda = \frac{9.3 fLW^2}{3{,}600 s p_c d^5} = \frac{fLW^2}{387 s p_c d^5}, \tag{66a}$$

$$W = 19.7 \sqrt{\frac{s p_c p_\lambda d^5}{fL}}, \tag{66b}$$

$$= 13.9 \sqrt{\frac{(p_1^2 - p_2^2)sd^5}{fL}}. \tag{66c}$$

Flow near Atmospheric Pressure.—Where gas or air flows through a pipe or round duct at a pressure only slightly above atmosphere, as is the case in low-pressure gas distribution or in ventilating work, the following substitutions can be made to simplify the formula and reduce it to the terms commonly used in these applications. Since the pressure above atmosphere is small, it is convenient to express pressure drop as h in *inches of water column* instead of as p_λ in pounds per square inch. From Table I page 30, $p_\lambda = 0.03613h$. Also near atmospheric pressure it is approximately true that $p_c = p_a = 14.7$. Substituting these relations in equation (61) it follows that, for *any* temperature,

$$p_\lambda = 0.03613h = \frac{14.7^2}{T_0^2} \times \frac{fsT_cLQ_0^2}{7.68 \times 14.7d^5},$$

$$h = \frac{53}{T_0^2} \times \frac{fsT_cLQ_0^2}{d^5}, \tag{67a}$$

$$Q_0 = \frac{T_0}{7.28} \sqrt{\frac{hd^5}{fsT_cL}}. \tag{67b}$$

In *ventilating* work where standard conditions for air are 14.7 psi abs at 70 F instead of at 60 F, the absolute temperature of 530 F corresponding to 70 F can be substituted for T_0 and s dropped for *air*. This gives

$$Q_a = \frac{530}{7.28}\sqrt{\frac{hd^5}{fT_cL}} = 72.8\sqrt{\frac{hd^5}{fT_cL}}. \qquad (67c)$$

If standard conditions, as in the *gas* industry, are approximately 14.7 psi abs at 60 F and the flow temperature also is assumed to be 60 F, the equation can be simplified still further by substituting $T_0 = T_c = 520$, which gives for gas *or* air,

$$h = \frac{53}{520} \times \frac{fsLQ_a^2}{d^5} = \frac{fsLQ_a^2}{9.81d^5}, \qquad (68a)$$

$$Q_a = 3.13\sqrt{\frac{hd^5}{fsL}} \qquad (68b)$$

and since the flow in *pounds* per minute $M = 0.07638Q_a$,

$$h = \frac{fsL}{9.81d^5} \times \left(\frac{M}{0.07638}\right)^2 = \frac{17.5fsLM^2}{d^5}, \qquad (69a)$$

$$M = 0.239\sqrt{\frac{hd^5}{fsL}}$$

NOTE: In *ventilating* work where *standard conditions* for air are 14.7 psi abs at 70 F instead of at 60 F, a density of 0.07493 lb per cu ft should be substituted for 0.07638 in the above formulas (see Table XVIII, page 146).

Likewise in the *gas* industry where flow is desired in *cubic feet* per *hour* instead of per *minute*, the corresponding substitutions can be made in equations (64) for *any* standard conditions:

$$h = \frac{p_0^2}{T_0^2} \times \frac{fsT_cLQ_{60}^2}{0.03613 \times 27{,}650p_cd^5} = \frac{p_0^2}{T_0^2} \times \frac{fsT_cLQ_{60}^2}{1{,}000p_cd^5}. \qquad (70a)$$

$$Q_{60} = 31.6\frac{T_0}{p_0}\sqrt{\frac{hp_cd^5}{fsT_cL}}. \qquad (70b)$$

If standard conditions are taken as approximately 14.7 psi abs at 60 F and the flow temperature also is assumed to be 60 F, equation (70) can be simplified by substituting $T_0 = T_c = 520$, and $p_0 = p_c = 14.7$ as follows:

$$h = \frac{14.7^2}{520^2} \times \frac{fs520LQ^2}{1{,}000 \times 14.7d^5} = \frac{fsLQ^2}{35{,}400d^5}, \qquad (71a)$$

$$Q = 188\sqrt{\frac{hd^5}{fsL}}. \qquad (71b)$$

And where weight of flow is desired in pounds per hour, $W = 0.07638Q = 60M$ can be substituted in equation (68) or (69), from which

$$h = \frac{fsLW^2}{205d^5} \tag{72a}$$

$$W = 14.3\sqrt{\frac{hd^5}{fsL}}. \tag{72b}$$

In low-pressure work the *velocity head* or change in static pressure required to produce a given velocity may be determined from the fundamental relation $H = v^2/2g$ as follows: If H represents the head in feet of gas or air and h the corresponding inches of water column, then $H = \frac{h}{12} \times \frac{62.4}{y}$ where y is the weight of gas or air in pounds per cubic foot at the given static pressure, temperature, and humidity. Substituting $H = \frac{v^2}{2g} = \frac{h}{12} \times \frac{62.4}{y}$ and transposing and solving for h,

$$h = 0.1925y\frac{v^2}{2g} = 0.00299yv^2, \tag{73a}$$

and

$$v = 18.27\sqrt{\frac{h}{y}} \tag{73b}$$

where v is the velocity in feet per *second.* If it is desired to solve in terms of v_m, the velocity in feet per *minute*, $v_m/60$ can be substituted for v as follows:

$$h = \frac{0.00299yv_m^2}{3{,}600} = 832 \times 10^{-9}yv_m^2, \tag{74a}$$

$$v_m = 1{,}096\sqrt{\frac{h}{y}}. \tag{74b}$$

In *ventilating* work the formula may be reduced to standard air conditions of 70 F and 14.7 psi abs by substituting for y its numerical value of 0.07493 which gives

$$h = 62.4 \times 10^{-9}v_m^2, \tag{75a}$$

and

$$v_m = 4005\sqrt{h}. \tag{75b}$$

Kinetic Energy Changes.—In the flow of gas or air where there is a considerable energy conversion from static pressure to velocity head, particularly with high-velocity flow at low pressure, the effect on the computed result may be enough to warrant taking the

energy changes into account. The general principles involved are discussed under Energy on pages 7 and 8 and under Velocity Produced by a Given Change in Heat Content on pages 70 and 71. Several articles on the application of these principles to the flow of gas and air through pipes are available in the literature[1] and can be referred to by those interested.

Adjusting Flow Factor to Test Results.—The principal point of variance between the formulas proposed by different investigators of the flow of air, gas, and steam through pipes lies in their method of fitting test results into the basic formula structure. Early tests were limited in scope and the initial tendency was to assign for use in the basic flow formula the same coefficients of friction and flow factors for all internal diameters and roughness of pipe. Later this was found to give discordant results and various methods of adjustment were tried in fitting the formula to other conditions. The three principal means of adjustment are listed below in connection with some of the well-known formulas in which each is employed:

1. *Flow Factor a Function of Pipe Diameter.*—In the Unwin and Spitzglass formulas (see pages 175, 184) the flow factor is made a function of pipe diameter. Another way of accomplishing an equivalent result is arbitrarily to assign a different flow factor to each pipe size (see page 192).

2. *Fractional Exponents.*—In the Williams-Hazen and the Saph-Schoder formulas for the flow of water (see pages 270, 276), the Fritzsche formulas for air, gas, or steam (see pages 177, 246), the Weymouth formula for gas (see page 186), and the Harris formula for air (see page 181), fractional exponents are assigned to d, v, or Q as the case may be.

3. *Rational Solution.*—In the rational formula method (see pages 107 to 137) the flow factor f is a function of pipe diameter, average velocity, density, and the absolute viscosity of the fluid. In one well-known form this functional relationship is known as a Reynolds number. Owing to the rational formula being the most logical and universally applicable method, it is used to an increasing extent in the gas industry.[2] In solving gas-flow problems by

[1] See "Pressure Drop in the Flow of Compressible Fluids," by Walter E. Lobo, Leo Friend, and G. T. Skaperdas, *Ind. Eng. Chem.*, Vol. 34, No. 7, pp. 821–823, July, 1942, which references four other articles.

[2] The following references are suggested as of particular interest:

(*a*) "Gas Engineering Flow Formulae and the Reynolds Number," by Wilbert J. Huff and Lloyd Logan, *AGA Proc.*, 1935, pp. 687–696. This paper

the rational method numerical values for the friction factor f, obtained from Fig. 15*a*, are substituted directly in the corresponding velocity, volume, or weight of flow formulas such as equations (48), (49), (56), (57), (64), (65), (66), or (72), or the method shown in Fig. 15*b* and Table XV can be used (see pages 113 to 118).

SPECIFIC FLOW FORMULAS—GAS AND AIR

In the order of their importance, the empirical formulas principally used at the present time for computing the flow of gas are those attributed to Weymouth, Spitzglass, Fritzsche, and Unwin. These formulas are applicable also to compressed air, and in some cases to ventilating work. The Harris formula, which is widely used for compressed air, closely resembles the Weymouth formula and gives much the same results within its own field. All these well-known formulas are presented in this section in a succession of rearrangements intended to facilitate solving for pressure drop, for flow in different terms, or for pipe diameter as the respective independent variable. Mention is made also of the possibilities of extending the so-called "rational formula" based on the Reynolds criterion into the gas and air field. The older gas-flow formulas of limited application, which now are of little more than historical interest, are reviewed briefly at the close of this section on gas and air flow.

PRINCIPAL FORMULAS IN COMMON USE

Unwin Formulas.—For what he termed "exceptionally smooth" pipes Unwin[1] determined a friction factor for *air* from tests on the Paris compressed-air mains which gives somewhat lower pressure

---

contains a bibliography of 37 references.

(*b*) "Flow of Natural Gas through High-pressure Transmission Lines," a joint report by T. W. Johnson and W. B. Berwald, *Monograph* 6, Bureau of Mines. Copies may be obtained from the American Gas Association, 420 Lexington Ave., New York, N.Y.

(*c*) "Gas Flow Computations," by P. McDonald Biddison, Natural Gas Section, American Gas Association, 1941. Reprints are available also from the AGA.

(*d*) "Viscosity of Natural Gas," by W. B. Berwald and T. W. Johnson, *Tech. Paper* 555, Bureau of Mines.

(*e*) "Handbook of Welded Steel Pipe," published, 1942, by Welded Pipe Division of California Corrugated Culvert Co., Berkeley, Calif.

[1] "Flow of Gas in Mains and Distribution at High Pressure," by W. C. Unwin, *Proc. Inst. Gas Engrs.*, London, published in *Journal of Gas Lighting, Water Supply*, etc., June, 21, 1904, pp. 852–867.

drops than he found to exist in gas mains (see below). Unwin's friction factor for air as expressed for use in the general formula is $f = 0.0025\left(1 + \frac{3.6}{d}\right)$. Substituting this value in equations (62) and (63) and noting that $s = 1$ for air and $y_a = 0.07638$ give the following results for air:

Unwin Formulas for Compressed-air Flow—Standard Conditions

(For definition of symbols, see page 82)

$$p_\lambda = \frac{\left(1 + \frac{3.6}{d}\right) L Q_a^2}{7{,}400 p_c d^5} \qquad p_\lambda = \frac{\left(1 + \frac{3.6}{d}\right) L M^2}{43 p_c d^5}.$$

$$Q_a = 86 \sqrt{\frac{p_c p_\lambda d^5}{\left(1 + \frac{3.6}{d}\right) L}} \qquad M = 6.56 \sqrt{\frac{p_c p_\lambda d^5}{\left(1 + \frac{3.6}{d}\right) L}}$$

$$= 43 \sqrt{\frac{(p_1^2 - p_2^2) d^5}{\left(1 + \frac{3.6}{d}\right) L}} \qquad = 4.63 \sqrt{\frac{(p_1^2 - p_2^2) d^5}{\left(1 + \frac{3.6}{d}\right) L}}$$

If standard conditions are other than 14.7 psi abs at 60 F, or the flow temperature is not 60 F, the corresponding substitutions can be made in equation (61), instead of (62).

On an assumption of a rougher pipe interior, Unwin assigned a somewhat higher friction factor for *gas* than for air or steam. Expressed in terms of f for substitution in the general flow formulas of this handbook, this can be reduced to $f = 0.0044\left(1 + \frac{1.714}{d}\right)$. Substituting this expression for f in equations (62), (63), (65), and (66) gives the following results for gas:

Unwin Formulas for Gas Flow—Standard Conditions

(For definition of symbols, see page 82)

$$p_\lambda = \frac{\left(1 + \frac{1.714}{d}\right) s L Q_a^2}{4{,}200 p_c d^5}. \qquad p_\lambda = \frac{\left(1 + \frac{1.714}{d}\right) s L Q}{151 \times 10^5 p_c d^5}.$$

$$Q_a = 64.8 \sqrt{\frac{p_c p_\lambda d^5}{\left(1 + \frac{1.714}{d}\right) sL}} \qquad Q = 3{,}885 \sqrt{\frac{p_c p_\lambda d^5}{\left(1 + \frac{1.714}{d}\right) sL}}$$

$$= 45.8 \sqrt{\frac{(p_1^2 - p_2^2) d^5}{\left(1 + \frac{1.714}{d}\right) sL}} \qquad 2{,}745 \sqrt{\frac{(p_1^2 - p_2^2) d^5}{\left(1 + \frac{1.714}{d}\right) sL}}.$$

$$p_\lambda = \frac{\left(1 + \frac{1.714}{d}\right) LM^2}{24.45 s p_c d^5}. \qquad p_\lambda = \frac{\left(1 + \frac{1.714}{d}\right) LW^2}{88{,}000 s p_c d^5}.$$

$$M = 4.95 \sqrt{\frac{s p_c p_\lambda d^5}{\left(1 + \frac{1.714}{d}\right) L}} \qquad W = 296.5 \sqrt{\frac{s p_c p_\lambda d^5}{\left(1 + \frac{1.714}{d}\right) L}}$$

$$= 3.49 \sqrt{\frac{(p_1^2 - p_2^2) s d^5}{\left(1 + \frac{1.714}{d}\right) L}}. \qquad = 209.5 \sqrt{\frac{(p_1^2 - p_2^2) s d^5}{\left(1 + \frac{1.714}{d}\right) L}}$$

Fritzsche Formulas.—Modified slightly and expressed in English units for substituting in the general formulas of this handbook, Fritzsche's[1] friction factor can be written for *any* standard conditions designated by the subscript 0.

Fritzsche Formulas for Gas or Air Flow in Cubic Feet per Hour for Any Standard Conditions at Any Flow Temperature

(For definition of symbols, see page 82)

$$f = 0.0048 \left[\frac{3{,}600 \times 53.33 T_0}{144 s p_0 Q_{60}}\right]^{1/7} = 0.01343 \left[\frac{T_0}{s p_0 Q_{60}}\right]^{1/7}$$

NOTE.—Examination of Fritzsche's expression for f shows it to involve empirically established relationships which represent the partial equivalent of a Reynolds criterion. Hence the Fritzsche formulas within their own field may be regarded as approaching the adaptability of the rational formula solution, but without the ability to take into account varying degrees of roughness of the pipe interior or to be extended to embrace fluids having viscosities other than those for which the empirical constants were determined.

Substituting the above value for f in equation (64) gives

$$p_\lambda = \frac{T_c L s^{0.857}}{204 \times 10^4 p_c d^5} \times \left[\frac{p_0 Q_{60}}{T_0}\right]^{1.857}.$$

$$Q_{60} = \frac{1{,}720}{s^{0.462}} \times \frac{T_0}{p_0} \left[\frac{(p_1^2 - p_2^2) d^5}{T_c L}\right]^{0.538}$$

$$d = \left[\frac{T_c L s^{0.857}}{102 \times 10^4 (p_1^2 - p_2^2)} \times \left(\frac{p_0 Q_{60}}{T_0}\right)^{1.857}\right]^{1/5}$$

NOTE.—$1/7 = 0.143$; $7/13 = 0.538$; $13/7 = 1.857$.

If desired, similar derivations can be made from equation (61) for flow in cubic feet per minute, or the flow can be converted to an hourly quantity and substituted in the above formulas.

[1] Fritzsche's formulas were published in German in 1908. For an account in English, see "Principles of Thermodynamics," by G. A. Goodenough, Henry Holt and Company, Inc., New York, 3d ed., 1925, p. 159.

Where *standard conditions* are those usual for compressed air or gas (see page 167) which approximate 14.7 psi abs at 60 F and the flow temperature also is assumed to be 60 F, a considerable simplification can be made through substituting the following numerical values in the foregoing formulas: $p_0 = p_a = 14.7$ psi abs; $T_0 = T_a = T_c = 460 \times 60 = 520$ F. Also under these conditions the Fritzsche friction factor simplifies to $f = 0.0224/(sQ)^{1/7}$; $f = 0.0125/(sQ_a)^{1/7}$; $f = 0.00862/M^{1/7}$; and $f = 0.0156/W^{1/7}$. These values for f can be substituted readily in the general flow formulas involving Q, Q_a, M, and W, respectively, to get

Fritzsche Formulas for Air and Gas Flow—Standard Conditions of 14.7 Psi at 60 F

(For definition of symbols, see page 82)

$$p_\lambda = \frac{s^{0.857} L Q_a^{1.857}}{1{,}480 p_c d^5}.$$

$$Q_a = \frac{51}{s^{0.462}} \left[\frac{p_c p_\lambda d^5}{L}\right]^{0.538} = \frac{35}{s^{0.462}} \left[\frac{(p_1^2 - p_2^2) d^5}{L}\right]^{0.538}.$$

$$d = 0.233 s^{0.1714} \left[\frac{L Q_a^{1.857}}{p_c p_\lambda}\right]^{0.2} = 0.267 s^{0.1714} \left[\frac{L Q_a^{1.857}}{p_1^2 - p_2^2}\right]^{0.2}.$$

$$p_\lambda = \frac{s^{0.857} L Q^{1.857}}{296 \times 10^4 p_c d^5}.$$

$$Q = \frac{3{,}050}{s^{0.462}} \left[\frac{p_c p_\lambda d^5}{L}\right]^{0.538} = \frac{2{,}100}{s^{0.462}} \left[\frac{(p_1^2 - p_2^2) d^5}{L}\right]^{0.538}.$$

$$d = 0.0507 s^{0.1714} \left[\frac{L Q^{1.857}}{p_c p_\lambda}\right]^{0.2} = 0.0583 s^{0.1714} \left[\frac{L Q^{1.857}}{p_1^2 - p_2^2}\right]^{0.2}.$$

$$p_\lambda = \frac{L M^{1.857}}{12.5 s p_c d^5}.$$

$$M = 3.89 \left[\frac{s p_c p_\lambda d^5}{L}\right]^{0.538} = 2.68 \left[\frac{(p_1^2 - p_2^2) s d^5}{L}\right]^{0.538}.$$

$$d = 0.603 \left[\frac{L M^{1.857}}{s p_c p_\lambda}\right]^{0.2} = 0.692 \left[\frac{L M^{1.857}}{s(p_1^2 - p_2^2)}\right]^{0.2}.$$

$$p_\lambda = \frac{L W^{1.857}}{24{,}900 s p_c d^5}.$$

$$W = 233 \left[\frac{s p_c p_\lambda d^5}{L}\right]^{0.538} = 161 \left[\frac{(p_1^2 - p_2^2) s d^5}{L}\right]^{0.538}.$$

$$d = 0.132 \left[\frac{L W^{1.857}}{s p_c p_\lambda}\right]^{0.2} = 0.152 \left[\frac{s L W^{1.857}}{s(p_1^2 - p_2^2)}\right]^{0.2}.$$

Like other fractional exponent formulas, the Fritzsche equations have to be solved with logarithms or a log-log slide rule. Figure 27 has been included to get around this difficulty, in so far as the equation for p_λ in terms of Q_a under standard conditions is concerned, by providing a chart from which a flow coefficient incor-

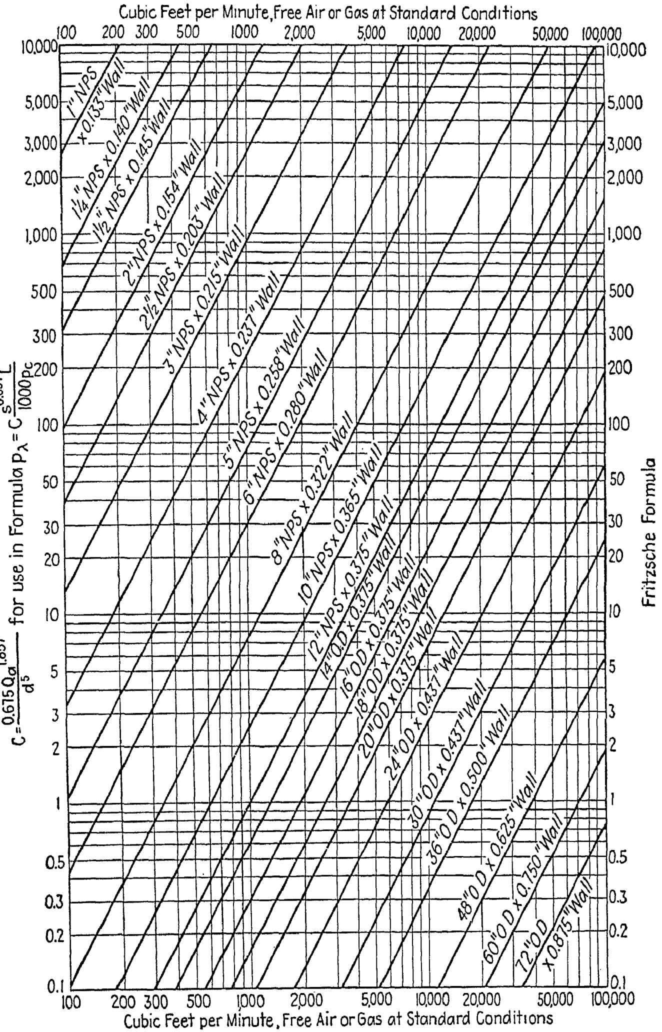


FIG. 27.—Chart for determining flow coefficient C used in simplifying the Fritzsche formula for air and gas where p_λ is given in terms of s, L, Q_a, and d.

porating the fractional power of Q_a can be read and substituted in the following simplified expressions:

$$p_\lambda = C \times \frac{\quad}{1.924p_c}$$

where

$$C = \frac{0.67[illegible]}{d^5}$$

or $p_\lambda = C \times \dfrac{s^{0.857}L}{1{,}000p_c}$, if the flow temperature is 60 F (520 F abs).

The flow coefficient C is read from the chart opposite the corresponding Q_a and pipe size intersection. For definition of other symbols see page 82. As in other applications of this type of formula where flow is associated with appreciable pressure drop, p_c should be taken as the average pressure in the line, *viz.*, $p_c = (p_1 + p_2)/2$, and T_c, if assumed other than 60 F, should be the average temperature in the line. In the case of air the specific gravity is unity and $s^{0.857}$ drops out of the formula. For gas where the specific gravity has to be taken into account, the tabulation values of $s^{0.857}$ given in Table XXIV will be an aid to solution.

TABLE XXIV.—VALUES OF $s^{0.857}$ FOR USE IN FRITZSCHE FORMULA FOR GAS

s	$s^{0.857}$	s	$s^{0.857}$	s	$s^{0.857}$
0.10	0.139	0.40	0.456	0.70	0.737
0.15	0.198	0.45	0.504	0.75	0.782
0.20	0.252	0.50	0.553	0.80	0.826
0.25	0.305	0.55	0.559	0.85	0.870
0.30	0.356	0.60	0.646	0.90	0.914
0.35	0.408	0.65	0.692	0.95	0.957

Fritzsche Formulas for Low-pressure Air and Gas Flow

(For definition of symbols, see page 82)

The Fritzsche formulas can be converted for ready use in computing the flow of air or gas in round ducts or pipes under low pressure, as in ventilating work or gas distribution. Pressures only slightly above atmosphere are measured in inches of water column. The conversion factor for changing inches of water column to pounds per square inch is 0.03613 (see Flow near Atmospheric Pressure, page 171), hence $p_\lambda = 0.03613h$. Substituting this value for p_λ in the respective Fritzsche formulas and considering $p_c = 14.7$,

$$h = \frac{s^{0.857}LQ_a^{1.857}}{785d^5};\quad Q_a = \left[\frac{786hd^5}{s^{0.857}L}\right]^{0.538};\quad d = \left[\frac{s^{0.857}LQ_a^{1.857}}{786h}\right]^{1/5}.$$

$$h = \frac{s^{0.857}LQ^{1.857}}{1,572,000d^5};\quad Q = \left[\frac{1,572,000hd^5}{s^{0.857}L}\right]^{0.538};\quad d = \left[\frac{s^{0.857}LQ^{1.857}}{1,572,000h}\right]^{1/5}.$$

$$h = \frac{LM^{1.857}}{6.62sd^5};\quad M = \left[\frac{6.62hsd^5}{L}\right]^{0.538};\quad d = \left[\frac{LM^{1.857}}{6.62sh}\right]^{1/5}.$$

$$h = \frac{LW^{1.857}}{13,240sd^5};\quad W = \left[\frac{13,240hsd^5}{L}\right]^{0.538};\quad d = \left[\frac{LW^{1.857}}{13,240sh}\right]^{1/5}.$$

The values given in some fan and blower catalogues are from 25 to 50 per cent higher than those obtained by the above formulas, probably to ensure the selection of an ample sized fan and motor. In ventilating work the conditions for *standard air* differ from those used in compressed air and gas work in that the air temperature is taken at 70 F rather than 60 F, the pressure being 14.7 psi abs (29.921 in. Hg) barometer in both cases. The above formulas were derived on the basis of 60 F air at 14.7 psi abs where the density y_a is 0.07638 lb per cu ft. For 70 F air the density y would be 0.07493 lb per cu ft (see Table XVIII, page 146), which can be substituted if desired as explained in connection with equation (67). The difference involved is small, however, in comparison with discrepancies between formulas and with test results and can well be neglected for most purposes.

The relation between *velocity head* and pressure in air or gas flow is discussed in connection with equation (73) on page 173.

Harris Formulas.—In the Harris[1] formula the friction factor f for *compressed air*, converted for substitution in the general flow formulas of this handbook, becomes $f = 0.00771/d^{0.31}$. This expression can be substituted for f in equations (61) to (66) and the specific gravity term s dropped in the case of air.

Since air flow usually is expressed in cubic feet or pounds per minute and this formula is commonly used for standard conditions of 14.7 psi abs at 60 F with the flow temperature likewise assumed to be 60 F, the following derivations from equations (62) and (63) will suffice:

Harris Formulas for Compressed-air Flow—Standard Conditions

(For definition of symbols, see page 82)

$$p_\lambda = \frac{LQ_a^2}{2,390p_cd^{5.31}}.\qquad p_\lambda = \frac{LM^2}{13.95p_cd^{5.31}}.$$

[1] See *Univ. of Missouri Bull.* 4, Vol. 1, 1912, or "Compressed Air," by Elmo G. Harris, McGraw-Hill Book Company, Inc., New York, *out of print.*

$$Q_a = 48.9\sqrt{\frac{p_c p_\lambda d^{5.31}}{L}} \qquad M = 3.74\sqrt{\frac{p_c p_\lambda d^{5.31}}{L}}$$

$$= 34.5\sqrt{\frac{(p_1^2 - p_2^2)d^{5.31}}{L}}. \qquad = 2.635\sqrt{\frac{(p_1^2 - p_2^2)d^{5.31}}{L}}$$

$$d = 0.232\left[\frac{LQ_a^2}{p_c p_\lambda}\right]^{0.188} \qquad d = 0.61\left[\frac{LM^2}{p_c p_\lambda}\right]^{0.188}$$

$$= 0.255\left[\frac{LQ_a^2}{p_1^2 - p_2^2}\right]^{0.188}. \qquad = 0.694\left[\frac{LM^2}{p_1^2 - p_2^2}\right]^{0.188}.$$

For those wishing to solve the Harris formulas without recourse to logarithms or a log-log slide rule, attention is called to Table VII on page 363 which gives values for $d^{5.33}$ and $\sqrt{d^{5.33}}$ which are close enough approximations to $d^{5.31}$ and $\sqrt{d^{5.31}}$ for most purposes. For instance, with a 6-in. pipe the discrepancy is about 2 per cent, which is much less than other variations to be expected.

Another convenient approximation of the Harris formulas is furnished in Table XXV, reproduced by permission of the Ingersoll Rand Company from "Compressed Air Data," by F. W. O'Neil (handbook published by *Compressed Air Magazine*, New York, N.Y.). The following variations of the same problem illustrate possibilities in using this table (see page 82 and Table XXV for definition of symbols):

Example 1.—Given a flow Q_a of 3,000 cfm of free air, an initial pressure of 120 psi abs, and an assumed pipe diameter of 6 in., what is the pressure drop per 1,000 ft of pipe?

Solution.—The ratio of compression is $r = 120/14.7 = 8.15$. Read opposite 3,000 cfm and below 6 in. in diameter a tabular value $N = 17.7$. The pressure drop per 1,000 ft of pipe is $N/r = 17.7/8.15 = 2.17$ psi.

Example 2.—Given a flow of 3,000 cfm, an initial pressure of 120 psi abs, and a desired pressure drop of about 2 psi per 1,000 ft of pipe, what diameter pipe will be required?

Solution.—Here again the ratio of compression $r = 8.15$. The approximate tabular number to look for will be $N = 2 \times 8.15 = 16.3$. In the table opposite 3,000 cfm it appears that a 6-in. pipe has an N of 17.7 which comes closest to the 16.3 desired. The answer is, use a 6-in. pipe.

Example 3.—Given a required flow of 3,000 cfm, a pipe diameter of 6 in., and a desired pressure drop of about 2.17 psi per 1,000 ft of pipe, what initial pressure will be required?

Solution.—In the table opposite 3,000 cfm for a 6-in. pipe, the value of N is read to be 17.7. The compression ratio corresponding to a pressure drop of 2 psi per 1,000 ft of pipe is $r = N/2.17 = 8.15$, from which $p_1 = 14.7 \times 8.15 = 120$ psi abs.

Example 4.—Given a 6-in.-diameter pipe, an initial pressure of 120 psi abs, and a pressure drop of about 2.17 psi per 1,000 ft of pipe, what will be the flow in cubic feet per minute of free air?

TABLE XXV.—FLOW OF COMPRESSED AIR THROUGH SCHEDULE 40 (STANDARD-WEIGHT) STEEL PIPE—HARRIS FORMULA[1]

Free air, cfm	Nominal diameter in inches									
	2	2½	3	3½	4	4½	5	6	8	10
320	61.1	23.8	7.5	3.5						
340	69.0	26.8	8.4	3.9	2.0					
360	77.3	30.1	9.5	4.4	2.2					
380	86.1	33.5	10.5	4.9	2.5					
400	94.7	37.1	11.7	5.4	2.7					
420	105.2	40.9	12.9	6.0	3.1					
440	115.5	44.9	14.1	6.6	3.4					
460	125.6	48.8	15.4	7.1	3.7	2.0				
480	137.6	53.4	16.8	7.8	4.0	2.2				
500	150.0	58.0	18.3	8.5	4.3	2.4				
525	165.0	64.2	20.2	9.4	4.8	2.6				
550	181.5	70.2	22.1	10.2	5.2	2.9				
575	197	76.7	24.2	11.2	5.7	3.1				
600	215	83.5	26.3	12.2	6.2	3.4				
625	233	92.7	28.5	13.2	6.8	3.7				
650	253	98.0	30.9	14.3	7.3	4.0	2.2			
675	272	105.7	33.3	15.4	7.9	4.3	2.4			
700	294	113.7	35.8	16.6	8.5	4.6	2.6			
750	337	130.5	41.1	19.0	9.7	5.3	2.9			
800	382	148.4	46.7	21.7	11.1	6.1	3.3			
850	433	168	52.8	24.4	12.5	6.8	3.8			
900	468	188	59.1	27.4	14.0	7.7	4.2			
950	541	209.4	65.9	30.5	15.7	8.6	4.7			
1,000	600	232.0	73.0	33.8	17.3	9.5	5.2	1.9		
1,050	658	256	80.5	37.8	19.1	10.4	5.8	2.1		
1,100	723	280.6	88.4	40.9	21.0	11.5	6.3	2.4		
1,150	790	306.8	96.6	44.7	22.9	12.5	6.9	2.6		
1,200	850	344.0	105.2	48.8	25.0	13.7	7.5	3.3		
1,300	.....	392.0	123.4	57.2	29.3	16.0	8.8	8.8		
1,400	.....	.....	.....	66.3	33.9	18.6	10.2	3.8		
1,500	.....	.....	.....	76.1	39.0	21.3	11.8	4.4		
1,600	.....	.....	.....	86.6	44.3	24.2	13.4	5.1		
1,700	.....	.....	.....	97.8	50.1	27.4	15.1	5.7		
1,800	.....	.....	.....	110.0	56.1	30.7	16.9	6.4		
1,900	.....	.....	.....	122	62.7	34.2	18.9	7.1	1.6	
2,000	.....	.....	.....	135	69.3	37.9	20.9	7.8	1.8	
2,100	.....	.....	.....	149	76.4	40.8	23.0	8.7	2.0	
2,200	.....	.....	.....	166	83.6	45.8	25.3	9.5	2.2	
2,300	.....	.....	.....	179	91.6	50.1	27.6	10.4	2.4	
2,400	.....	.....	.....	195	99.8	54.6	30.1	11.3	2.6	
2,500	.....	.....	.....	212	108.3	59.2	32.6	12.3	2.9	
2,600	.....	.....	.....	229	117.2	64.9	35.3	13.3	3.1	
2,700	.....	.....	.....	247	126	69.1	38.1	14.3	3.3	
2,800	.....	.....	.....	265	136	74.3	41.0	15.4	3.6	
2,900	.....	.....	.....	285	146	79.8	43.9	16.5	3.9	
3,000	.....	.....	.....	305	156	85.2	47.0	17.7	4.1	
3,200	.....	.....	.....	347	177	97.1	53.5	20.1	4.7	
3,400	.....	.....	.....	391	200	109.5	60.4	22.7	5.3	
3,600	.....	.....	.....	438	224	122.8	67.6	25.4	5.6	1.8
3,800	.....	.....	.....	488	250	137	75.5	28.4	6.6	2.0
4,000	.....	.....	.....	542	277	151	83.6	31.4	7.3	2.2
4,200	.....	.....	.....	.....	305	168	92.1	34.6	8.1	2.4
4,400	.....	.....	.....	.....	335	183	101.2	38.1	8.9	2.7
4,600	.....	.....	.....	.....	366	200	110.5	41.5	9.7	2.9
4,800	.....	.....	.....	.....	399	218	120.4	45.2	10.5	3.2
5,000	.....	.....	.....	.....	433	236	131	49.1	11.5	3.4
5,250	.....	.....	.....	.....	477	260	144	54.1	12.6	3.8

[1] Based on the formulas of Elmo G. Harris, *op. cit.* This table is reproduced by permission of the Ingersoll Rand Co. from "Compressed Air Data." The tabular values N divided by the ratio of compression r represent the pressure drop in psi per 1,000 ft of pipe. In symbols these relations are expressed as (see p. 82 for definition of symbols): $r = p_1/p_a$; $1{,}000p\lambda\ /L$ = the pressure drop per 1,000 ft of pipe; $N/r = 1{,}000p\lambda\ /L$; and $N = r\ 1{,}000p\lambda\ /L$. These relations can be used to solve problems as shown for examples in the text on p. 182. Note that N as used here is not a Reynolds number.

Solution.—The tabular number can be determined as follows: $r = 120/14.7 = 8.15$, and $N = r1{,}000p_\lambda/L = 8.15 \times 2.17 = 17.7$. Looking down the column for 6-in.-diameter pipe, and N value of 17.7 is found opposite 3,000 cfm. Hence the answer is 3,000 cfm.

Comparison of Example 1 of the foregoing tabular solution with results of various formulas:[1]

NOTE.—The specific gravity term s has been dropped from these solutions owing to its being unity for air.

Harris Formula (see page 181):

$$p_\lambda = \frac{LQ_a^2}{2{,}390p_cd^{5.31}} = \frac{1{,}000 \times 3{,}000^2}{2{,}390 \times 119 \times 14{,}300} = 2.21.$$

Unwin Formula (see page 176):

$$p_\lambda = \frac{\left(1 + \dfrac{3.6}{d}\right)LQ_a^2}{7{,}400p_cd^5} = \frac{1.593 \times 1{,}000 \times 3{,}000^2}{7{,}400 \times 119 \times 8{,}206} = 1.98.$$

Fritzsche Formula (see page 178):

$$p_\lambda = \frac{LQ_a^{1.857}}{1{,}480p_cd^5} = \frac{1{,}000 \times 3{,}000^{1.857}}{1{,}480 \times 119 \times 8{,}206} = 1.94.$$

NOTE.—The same value can be obtained by the chart solution of Fig. 27.

Spitzglass Formula (see page 185):

$$p_\lambda = \frac{3{,}600LQ_a^2}{2{,}333 \times 10^4p_cK^2} = \frac{3{,}600 \times 1{,}000 \times 3{,}000^2}{2{,}333 \times 10^4 \times 119 \times 67.97^2} = 2.53.$$

Weymouth Formula (see page 187):

$$p_\lambda = \frac{lQ^2}{1{,}572p_cd^{5.33}} = \frac{1{,}000}{5{,}280} \times \frac{(3{,}000 \times 60)^2}{1{,}572 \times 119 \times 14{,}970} \qquad 2.19.$$

Rational Formula [see equation (62*a*), page 170]:

$$p_\lambda = \frac{fLQ_a^2}{18.5p_cd^5} = \frac{0.0043 \times 1{,}000 \times 3{,}000^2}{18.5 \times 119 \times 8{,}206} = 2.14.$$

Spitzglass Formulas.—In 1912 Spitzglass[2] published the results of his tests for the Peoples Gas Light and Coke Company of

[1] For a comparison of the volume of gas flow computed by different formulas see "Flow of Natural Gas through High-pressure Transmission Lines," a joint report by T. W. Johnson and W. B. Berwald, *Monograph* 6, Bureau of Mines, 1935. Reprints may be obtained through the American Gas Association, 420 Lexington Ave., New York, N. Y.

[2] "Flow of Gas Formulae, Derived, Analyzed and Checked by Experimental Data, with Diagrams for Figuring the Flow of Gas in Street Mains and Services," by J. M. Spitzglass, *Am. Gas Light J.*, Vol. 96, 1912, pp. 269–271, 274–276, 290–296, 312–315.

Chicago and suggested a new formula for the flow of gas or air through pipes. Spitzglass concluded that the effect of internal friction on the viscosity of the fluid, which was neglected in the mean hydraulic radius (see page 85), must cause variation of the friction factor with pipe diameter. He reasoned that the resistance to flow caused by internal friction bore some relation to the cross-sectional area, whereas skin friction varied with the wetted perimeter. This pointed to the possibility that when the pipe diameter increased over a certain limit, the internal friction would become so preponderant that the friction factor, instead of diminishing, would increase with the diameter of the pipe. This consideration led Spitzglass to the adoption of a friction factor having the form $f = a\left(1 + \frac{b}{d} + cd\right)$ "which takes care of variation in both directions" as he put it. As expressed for use in the general flow formulas of this handbook, the Spitzglass friction factor for gas and air flow becomes $f = 0.0028\left(1 + \frac{3.6}{d} + 0.03d\right)$. This expression gives a rather rapid decrease in f with increase in diameter up to about 6-in. pipe size, a slow decrease in f from 6 in. to 10 or 12 in., and a slow rise in f from 12 in. up.

This form of expression does not lend itself readily to a direct solution for pipe diameter, but it can be solved conveniently for pressure drop or volume of flow. Solution for p_λ or Q is facilitated by grouping all the diameter terms into one expression, customarily designated as $K = \sqrt{d^5 \Big/ \left(1 + \frac{3.6}{d} + 0.03d\right)}$, for which numerical values for different pipe diameters can be read from tables (see Tables VII and VIII, pages 363 and 364). The Spitzglass formulas set up for use with K are as follows:

For pressures over 1 psi g:

$$p_\lambda = \frac{sLQ^2}{2{,}333 \times 10^4 p_c K^2}.$$

$$Q = 4{,}830K\sqrt{\frac{p_c p_\lambda}{sL}}$$

$$= 3{,}410K\sqrt{\frac{p_1^2 - p_2^2}{sL}}$$

For pressures not over 1 psi g:

$$h = \frac{sLQ^2}{126 \times 10^5 K^2}$$

$$Q = 3{,}550K\sqrt{\frac{h}{sL}}.$$

According to Johnson and Berwald in *Bur. Mines Monograph* 6,[1] the Spitzglass formula gives a somewhat lower gas flow than would

[1] See *Monograph* 6, "Flow of Natural Gas through High-pressure Trans-

be computed by the Weymouth formula.[1] The difference varies with pipe diameter, ranging from 10 per cent below Weymouth flow quantities for a 6-in. pipe, to 30 per cent below for a 24-in. pipe. Whereas the Weymouth formula is regarded by Berwald and Johnson as the most accurate of the empirical formulas, especially in the larger pipe sizes and with clean steel pipe, the deviation of the Spitzglass formula is on the conservative side. Hence some gas engineers prefer to use the Spitzglass formula where much rust, scale, or condensates are present in the line, especially in the diameters of cast-iron pipe 12 in. or smaller. The present trend, however, is toward the rational form of solution described on pages 107 to 137, see also discussion of its application to gas on pages 165 to 173.

Weymouth Formulas.—In 1912 Thomas R. Weymouth presented in an ASME paper his formulas which are generally considered to be the best suited of the empirical formulas for computing the flow of gas through intermediate and high-pressure transmission lines. Johnson and Berwald concluded from the results of flow tests made on 29 natural gas lines totaling 757 miles of pipe of 6 to 22 in. in diameter and operating at pressures of 30 to 600 psi that, "The curve based on the Weymouth formula . . . agreed closely with the average of the experimental results and satisfied the test data as well as any curve that could be drawn through the plotted points."

The Weymouth formulas employ fractional exponents and have to be solved with logarithms or a log-log slide rule except where numerical values for the fractional powers of d can be read from a table (see Table VII, page 363). As expressed for use in the general flow formulas of this handbook, the Weymouth friction factor becomes $f = 0.008/d^{1/3}$. The Weymouth formula for generalized conditions is expressed in the symbols of this handbook as follows (see page 82 for definition of symbols and note that l is used here to denote the length of the line in *miles* instead of L in feet and that the subscript 0 is used to denote *any* standard conditions):

---

mission Lines," Bureau of Mines, a joint report by T. W. Johnson and W. B. Berwald, 1935. Copies may be obtained through the American Gas Association, 420 Lexington Ave., New York 17, N. Y.

[1] "Problems in Natural-gas Engineering," by Thomas R. Weymouth, *Trans. ASME*, Vol. 34, pp. 185–206, 1912.

Weymouth Formulas for Gas or Air Flow in Cubic Feet per Hour for Any Standard Conditions at Any Flow Temperature

(See page 82 for definition of symbols, l is in miles)

$$p = \frac{sT_cl}{652p_cd^{16/3}}\left(\frac{p_0Q_{60}}{T_0}\right)^2.$$

$$Q_{60} = 18.062\frac{T_0}{p_0}\sqrt{\frac{(p_1^2 - p_2^2)d^{16/3}}{sT_cl}}.$$

$$d = \left[\frac{sT_cl}{326(p_1^2 - p_2^2)} \times \left(\frac{p_0Q_{60}}{T_0}\right)^2\right]^{3/16}.$$

NOTE.—$8/3 = 2.667$; $16/3 = 5\frac{1}{3} = 5.333$; $3/16 = 0.1875$. See Tables VII and VIII, pages 363 and 364, for inside diameter functions involving these powers.

Where *standard conditions* are those usual for gas or compressed air (see page 167) which approximate 14.7 psi abs at 60 F, and the flow temperature also is assumed to be 60 F, a considerable simplification can be made through substituting the following numerical values in the foregoing formulas:

$$p_0 = p_a = 14.7 \text{ psi abs};\ T_0 = T_a = T_c = 460 + 60 = 520 \text{ F}.$$

Weymouth Formulas for Gas or Air Flow in Cubic Feet per Hour for Standard Conditions and 60 F Flow Temperature

(See page 82 for definition of symbols, l is in miles)

$$p_\lambda = \frac{slQ^2}{1{,}572p_cd^{16/3}}.$$

$$Q = 28.05\sqrt{\frac{(p_1^2 - p_2^2)d^{16/3}}{sl}}.$$

$$d = \left[\frac{slQ^2}{786(p_1^2 - p_2^2)}\right]^{3/16}.$$

Flow capacities computed by the Weymouth formula for pipe sizes from ¾ in. to 8 in., inclusive, for a variety of intermediate pressures will be found in Table VI on page 1188.

Sample Problems in Gas Flow.—The following examples have been chosen as a means of demonstrating the principal gas flow formulas and of showing how to use the aids to solution provided in this handbook, at the same time affording a spot comparison of the results.[1] Owing to the frequency with which the expression $\sqrt{p_1^2 - p_2^2}$ appears in gas flow formulas, a schedule of values for the expression has been provided in Table XXVI.

[1] For a thoroughgoing comparison of gas flow formulas see, "Flow of Natural Gas through High-pressure Transmission Lines," *op. cit.*

TABLE XXVI.—COMPUTED VALUES OF THE EXPRESSION $\sqrt{p_1^2 - p_2^2}$ FOR USE IN GAS AND AIR-FLOW FORMULAS[1]

(NOTE.—For convenience in use, the column and line designations are given in psi *gauge*, whereas the tabular values were computed from the corresponding *absolute* pressures)

Tabular values = $\sqrt{p_1^2 - p_2^2}$, where p_1 and p_2 are absolute pressures in psi, *viz.*, gauge pressure + 14.7

Inlet pressure, psi, gauge	Discharge pressure, psi, gauge																						
	1	2	4	6	8	10	20	30	40	50	60	70	80	90	100	150	200	250	300	350	400	450	500
2	5.69																						
4	10.2	8.31																					
6	13.5	12.2	8.87																				
8	16.4	15.4	12.9	9.31																			
10	19.1	18.4	16.1	13.5	9.74																		
12	21.6	20.8	19.1	16.9	14.1	10.1																	
14	24.0	23.3	21.8	19.9	17.6	14.6																	
16	26.4	25.8	24.4	22.7	20.7	18.2																	
18	28.7	28.1	26.8	25.3	23.5	21.4																	
20	30.9	30.4	29.2	27.8	26.2	24.4																	
30	41.9	41.5	40.6	39.6	38.5	37.3	28.2																
40	52.4	52.1	51.4	50.6	49.8	48.8	42.3	31.5															
50	62.8	62.5	61.9	61.3	60.6	59.8	54.6	46.8	34.6														

60	73.0	72.8	72.3	71.8	71.2	70.5	66.2	59.9	50.9	36.3													
70	83.2	83.0	82.6	82.1	81.6	81.0	77.3	71.9	64.7	54.7	39.9												
80	93.4	93.2	92.8	92.4	91.9	91.4	88.1	83.5	77.3	69.2	58.2	42.4											
90	104	103	103	103	102	102	98.8	94.7	89.3	82.3	73.4	61.6	44.7										
100	114	114	113	113	112	112	109	106	101	94.7	87.0	77.3	64.7	46.8									
150	164	164	164	163	163	163	161	159	155	152	147	141	135	127	118								
200	214	214	214	214	214	213	212	210	208	205	201	197	193	187	182	138							
250	264	264	264	264	264	264	262	261	259	257	254	251	247	243	239	207	155						
300	314	314	314	314	314	314	313	312	310	308	306	303	300	297	293	268	230	170					
350	364	364	364	364	364	364	363	362	361	359	357	355	352	349	346	325	295	251	184				
400	414	414	414	414	414	414	413	412	410	409	408	406	404	401	399	381	355	319	270	197			
450	464	464	464	464	464	464	463	463	462	460	459	457	455	453	450	435	412	382	342	288	210		
500	515	514	514	514	514	514	514	513	512	511	509	508	506	504	502	488	468	441	407	363	305	221	
600	615	615	615	614	614	614	614	613	612	611	610	609	607	606	604	592	576	555	528	494	454	402	336
700	715	715	715	714	714	714	714	713	713	712	711	710	708	707	705	695	678	664	642	615	572	533	496
800	815	815	815	814	814	814	814	814	813	812	811	810	809	808	807	798	786	771	752	729	701	669	622
900	915	915	915	915	914	914	914	914	913	912	912	911	910	909	908	900	889	876	859	839	815	788	756
1,000	1,015	1,015	1,015	1,015	1,014	1,014	1,014	1,014	1,013	1,013	1,012	1,011	1,010	1,009	1,008	1,001	992	980	965	947	926	902	875

[1] Irrespective of the available pressure difference, the maximum capacity of a pipe line is limited by the acoustic velocity of the fluid at the downstream end of the line (see p. 254).

Example 1.—What is the carrying capacity in cubic feet of natural gas per hour measured at standard conditions of a 60-mile line of 8-in. Schedule 30 pipe if the initial pressure is 300 psi gauge, the final pressure is 30 psi gauge, the flow temperature is 60 F, and the specific gravity is 0.7?

Solution Aids.—The following aids to solution were used in connection with the formulas listed below:

$$\sqrt{p_1^2 - p_2^2} = \sqrt{(300 + 14.7)^2 + (30 + 14.7)^2} = 312 \text{ (from Table XXVI, page 188).}$$

$$p_1^2 - p_2^2 = 312^2 = 97{,}344.$$

$$l = 60 \text{ miles}, L = 60 \times 5{,}280 = 316{,}800 \text{ ft.}$$

$$d = 8.071; \sqrt{d^5} = 185.1; \sqrt{d^{1\,6/3}} = 262.1; K = 142.5 \text{ (from Table VII, page 363).}$$

$$d^3 = 525.7;\ d^4 = 4{,}243;\ d^5 = 34{,}248 \text{ (from Table IX, page 365).}$$

Unwin Solution (see page 176):

$$Q = 2{,}745 \sqrt{\frac{(p_1^2 - p_2^2)d^5}{\left(1 + \frac{1.714}{d}\right) sL}} = \frac{2{,}745 \times 312 \times 185.1}{\sqrt{\left(1 + \frac{1.714}{8.071}\right) 0.7 \times 316{,}800}}$$

$$= 306{,}000 \text{ cu ft per hr.}$$

Fritzsche Solution (see page 178):

$$Q = \frac{2{,}100}{s^{0.462}} \left[\frac{(p_1^2 - p_2^2)d^5}{L}\right]^{0.538} = \frac{2{,}100}{0.7^{0.462}} \left[\frac{97{,}344 \times 34{,}248}{316{,}800}\right]^{0.538}$$

$$= \frac{2{,}100 \times 146}{0.849} = 361{,}000 \text{ cu ft per hr.}$$

Spitzglass Solution (see page 185):

$$Q = 3{,}410K \sqrt{\frac{p_1^2 - p_2^2}{sL}} = \frac{3{,}410 \times 142.5 \times 312}{\sqrt{0.7 \times 316{,}800}} = 321{,}000 \text{ cu ft per hr.}$$

Weymouth Solution (see page 187):

$$Q = 28.05 \sqrt{\frac{(p_1^2 - p_2^2)d^{1\,6/3}}{sl}} = \frac{28.05 \times 312 \times 261.1}{0.7 \times 60} = 352{,}200 \text{ cu ft per hr.}$$

Rational Solution [using equation (65*c*) page 171, and Figs. 15*a* and 17*a*, pages 107 to 113]:

$$Q = 182 \sqrt{\frac{(p_1^2 - p_2^2)d^5}{fsL}} \qquad \frac{Qs}{dz} = \frac{350{,}000 \times 0.7}{8.071 \times 0.0105} = 2{,}890{,}000.$$

$$f = 0.00405 \text{ from Fig. } 15a.$$

$$= \frac{182 \times 312 \times 185.1}{\sqrt{0.00405 \times 0.7 \times 316{,}800}} = 350{,}000 \text{ cu ft per hr.}$$

NOTE.—Since Q is unknown, it is necessary to resort to cut-and-try methods as described on page 113 and assume a value for Q in order to determine the friction factor f. Under these circumstances it may be preferable to use the direct solution given below:

Rational Solution (using equations of Table XV*B*, and Figs. 15*b* and 17*a*, see pages 113 to 118):

$$N' = \frac{249{,}600(p_1^2 - p_2^2)sd^3}{Lz^2T_c} = \frac{249{,}600 \times 97{,}344 \times 0.7 \times 525.7}{316{,}800 \times 0.0105^2 \times 520}$$

$$= 493 \times 10^6, \text{ and from Fig. } 15b,\ C = 350.$$

$$Q = \frac{1{,}000(p_1^2 - p_2^2)d^4}{CLz} = \frac{1{,}000 \times 97{,}344 \times 4{,}243}{350 \times 316{,}800 \times 0.0105} = 354{,}000.$$

For an extra check on f as computed from Table XV*B*, page 116,

$$\frac{C^2Lz^2T_c}{15{,}704(p_1^2 - p_2^2)sd^3} = \frac{350^2 \times 316{,}800 \times 0.0105^2 \times 520}{15{,}704 \times 97{,}344 \times 0.7 \times 525.7} = 0.00397.$$

Example 2.—Required the diameter of a pipe 10,000 ft in length for a flow of 3,000 cu ft of free air per minute; initial pressure 120 psi abs with a pressure drop of 20 psi; flow temperature 60 F.

Harris Solution (see page 182):

$$d = 0.255\left[\frac{LQ_a^2}{p_1^2 - p_2^2}\right]^{0.188} = 0.255\left[\frac{10{,}000 \times 3{,}000^2}{120^2 - 100^2}\right]^{0.188} = 6.03 \text{ in.}$$

Fritzsche Solution (see page 178):

$$d = 0.267s^{0.1714}\left[\frac{LQ_a^{1.857}}{p_1^2 - p_2^2}\right]^{0.2} = 0.267 \times 1\left[\frac{10{,}000 \times 3{,}000^{1.857}}{120^2 - 100^2}\right]^{0.2} = 6.14 \text{ in.}$$

Weymouth Solution (see page 187):

$$d = \left[\frac{slQ^2}{786(p_1^2 - p_2^2)}\right]^{3/16} = \left[\frac{1 \times 10{,}000 \times (60 \times 3{,}000)^2}{786 \times 5{,}280(120^2 - 100^2)}\right]^{3/16} = 6.25 \text{ in.}$$

Rational Solution (using equations of Table XV*B* and Fig. 15*b*):

$$N' = \frac{1{,}400Qs}{zT_c\sqrt{LzQ/(p_1^2 - p_2^2)}} = \frac{1{,}400 \times 60 \times 3{,}000 \times 1}{0.0182 \times 520\sqrt{\dfrac{10{,}000 \times 0.0182 \times 60 \times 3{,}000}{120^2 - 100^2}}}$$

$= 2{,}662{,}000$ and (from Fig. 15*b*) $C = 180$.

$$d = 0.178\sqrt[5]{\frac{180 \times 10{,}000 \times 0.0182 \times 60 \times 3{,}000}{120^2 - 100^2}}$$

$= 6.06$ in.

Answer.—A 6-in. Schedule 40 (standard-weight) steel pipe having an inside diameter of 6.065 in. will be satisfactory.

OLD-STYLE FORMULAS OF LIMITED USE

The old-style formulas of limited use for computing the flow of gas are of two general types, one suited to problems in low-pressure distribution and the other to intermediate and high-pressure transmission lines. Each of these formulas was developed empirically years ago to fit some limited application and both types have been largely superseded now by the more universally applicable formulas described in the preceding pages. The older formulas mostly conform to the general flow equation but are deficient in that the flow factor was wrongly assumed to remain the same for all rates of flow and pipe diameters. The two types of formulas are summarized briefly in succeeding paragraphs for the information of those who have occasion to refer to them. For the sake of uniformity, all formulas are expressed here in the symbols defined on page 82, and the lengths of main are stated in feet (L) or miles (l) instead of in yards as was often done in early times.

Old-style Low-pressure Gas Flow Formulas.—The best known of the old-style "low-pressure" formulas are those attributed to Molesworth and Pole, although there were others including those of Clegg and Gill giving intermediate values and that of Humphries giving a greater flow. This type of formula is expressed as $Q = C\sqrt{hd^5/sL}$ where C is a coefficient combining the flow factor with the formula constants. The following values for C were given by Molesworth and Pole:

	Molesworth	Pole
Where L is in yards, $C =$	1,000	1,350
Where L is in feet, $C =$	1,732	2,338

These formulas had no means of compensating for change in friction factor with diameter, and the diversity of constants evidently came about through the authors having tested different size pipes, although the condition of the interior may have had something to do with the discrepancy. The simplicity of this type of formula greatly facilitates solution and it can be made to yield reasonably satisfactory results by varying the friction factor with diameter. The following values are suggested in the "American Gas Handbook"[1] for smooth pipes and usual conditions:

RECOMMENDED ADJUSTMENT OF CONSTANT C WITH PIPE DIAMETER

	Nominal pipe size, in.				
	¾ to 1	1¼ to 1½	2	3	4 and larger
Where L is in yards, $C =$	1,000	1,100	1,200	1,300	1,350
Where L is in feet, $C =$	1,732	1,905	2,078	2,252	2,338

Limitations of Low-pressure Formulas.—The low-pressure formulas for gas or air flow near atmospheric pressure all have a common limitation owing to their lack of a term to compensate for change in density (or volume) with change in pressure. This limitation applies to the general formulas given in equations (67) to (72) and to the low-pressure versions of the Fritzsche and Spitzglass formulas, as well as to "old-style" formulas such as those of Molesworth and Pole. These formulas are identified by

[1] Published by American Gas Journal, 53 Park Place, New York, N. Y.

the use of h to express pressure drop in inches of water column, and by the absence of p and y terms which are combined in the numerical coefficient.

Owing to the aforesaid limitations the low-pressure formulas are applicable only where the initial pressure does not much exceed 1 psi g (27.68 in. of water column). Furthermore, the over-all pressure drop for the system should not exceed two-thirds of the maximum pressure differential above atmosphere because of the adverse effect on metering as well as on the behavior of gas appliances and pilot lights if the differential fluctuates more than 50 per cent above or below some established average service pressure. For instance, if the established average service pressure is 6 in. of water column above atmospheric pressure, the maximum and minimum pressures should not be allowed to vary more than 3 in. of water column above or below the average, *viz.*, they should be confined to the range between 3 and 9 in. of water column.

Old-style High-pressure Gas Flow Formulas.—Several old-style formulas for the flow of high-pressure gas were in use at one time or another. Like the old-style low-pressure formulas, most of these were limited through having a constant friction factor for all pipe diameters. This type usually took the form $Q = C\sqrt{[(p_1^2 - p_2^2)d^5]/sL}$, although some were further limited in that they were set up for only one predetermined specific gravity and had the s term incorporated in C. Among these formulas were those of Cox,[1] Rix,[2] and Towl,[3] and another known as the Pittsburgh[4] formula. Towl subsequently modified his formula by varying the flow coefficient with pipe size and rate of flow.[5] Oliphant[6] brought in the effect of pipe diameter through an extra term involving $d^3/30$ so that where other formulas have $\sqrt{d^5}$, the Oliphant formula has $[\sqrt{d^5 + (d^3/30)}]$. Although these formulas appeared in the first three editions of "Piping Handbook," space limitations do not permit continuing the practice. Those

[1] "Flow of Gas through Pipes," by William Cox, *American Machinist*, Vol. 24, pp. 401–402, Mar. 20, 1902.

[2] "Compression and Transmission of Illuminating Gas," by E. A. Rix, *Proc. Pacific Coast Gas Assoc.*, Vol. 6, pp. 97–98, 1905.

[3,4] See "Measurement, Compression, and Transmission of Natural Gas," by L. C. Lichty, John Wiley & Sons, Inc., New York, 1924, p. 372.

[5] "f the Pipe Line Flow Factor and Its Relation to Density and Viscosity," by Forrest M. Towl, published in two parts 1934 and 1935, respectively, for personal distribution.

[6] "Production of Natural Gas in 1902," by F. H. Oliphant, U. S. Geological Survey. See also footnotes 2 and 3.

having occasion to refer to them can look in the earlier editions or in the original sources now listed in footnotes.

MISCELLANEOUS ITEMS CONCERNING THE FLOW OF GAS AND AIR

Several items concerning the flow of gas and air through pipes are collected in this section for convenient reference. In addition, attention is called to the following topics which are covered elsewhere as noted: Resistance of Bends, Fittings, and Valves (see page 95); Relative Carrying Capacity of Pipes of Different Diameters (see pages 87 to 89); Complex Pipe Lines and Divided Circuits or Loops (see pages 90 to 95); Flow through Nozzles and Orifices (see pages 74 to 81).

Pipe-line Storage Capacity.[1]—The storage capacity of a pipe line is the excess gas, over and above the average rate of supply, that can be packed in the line during the period when the demand is less than the supply. The supply rate is the rate of average daily delivery and, since the volume of gas contained in the line is a function of the pressure, it is necessary, in order to determine the storage capacity, to develop an expression for the total line contents, which will take into account the variable pressure conditions at all points along the line. Having such an expression, the total contents under the "packed" and "unpacked" conditions may be computed and the storage ascertained by taking their difference.

The unpacked condition of a pipe line obtains when the gas is flowing at the average daily rate, for when the consumption is below this point all excess gas is being stored, and this point, therefore, is the lower limit, or base of the peaks in the daily delivery curve for the day.

The upper limit, or packed-line condition, is not so readily determined, but if it is considered to be such that the intake pressure is at its maximum point, as fixed by considerations of safety, station-pump pressure limits, etc., and the flow is a mean between the minimum and average rates for the day, the result will be very nearly actual conditions, and whatever error may thereby be involved will be on the side of safety in estimating the storage capacity.

Figure 28 shows the variation of intake pressure p_1 for varying values of L with the discharge pressure p_2 constant. Two curves

[1] From "Problems in Natural Gas Engineering," by Thomas R. Weymouth, *Trans. ASME*, Vol. 34, 1912.

are plotted: The solid line represents the conditions with an assumed average rate of flow and the discharge pressure p_2 fixed by the minimum value allowable by the requirements of the distributing system. The dotted line represents the pressure conditions in the same line with the intake pressure p_1' at the maximum allowable value and the flow at an assumed mean of the minimum and average rates for the day. This latter curve represents the packed line, and the shaded area the quantity of gas stored in the line available for peak loads.

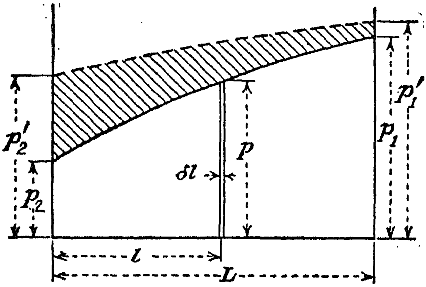


FIG. 28.—Variation of intake pressure p_1 for varying values of L length of pipe, with discharge pressure p_2 constant.

Let p = absolute pressure in pounds per square inch at any point; Q the discharge in cubic feet per hour; l or L the length of the line in miles; and d the internal diameter in inches; then for any given line and flow condition, from equation of original Weymouth paper reproduced on page 187, and assuming $s = 0.573$ in the line:

$$p^2 - p_2^2 = l\frac{Q^2}{37^2 d^{5\frac{1}{3}}} = Kl.$$

wherein
$$K = \frac{Q^2}{37^2 d^{5\frac{1}{3}}}.$$

Hence,
$$p = \sqrt{Kl + p_2^2}.$$

The gas contained in a length δl based on pressure $p_0 = 14.65$ will be in cubic feet measured at standard conditions (see page 167),

$$\delta V = 5{,}280A \times \delta l \times \frac{}{p_0},$$

where A is the cross-sectional area of the pipe in square feet. The total quantity of gas in the line under the given conditions will

then be

$$V = \int \delta V = 5{,}280 \frac{A}{p_0} \int_0^L \sqrt{Kl + p_2^2}\, \delta l$$
$$= \frac{3{,}520}{K} \frac{A}{p_0} [(KL + p_2^2)^{3/2} - p_2^3].$$

But $$KL = p_1^2 - p_2^2 \text{ and } K = \frac{p_1^2 - p_2^2}{L}.$$

Hence, $$V = \frac{3{,}520AL}{p_0(p_1^2 - p_2^2)} (p_1^3 - p_2^3)$$
$$= 3{,}520 \frac{AL}{p_0} \left[p_1 + p_2 - \frac{p_1 p_2}{p_1 + p_2} \right]$$
$$= 19.20 \frac{d^2 L}{p_0} \left[p_1 + p_2 - \frac{p_1 p_2}{p_1 + p_2} \right].$$

In the case of a complex system, it is necessary, first, to ascertain the common pressures at all junction points where the lines are tied together, and then to compute separately the capacity of each section. To do this it is necessary to use the actual diameter, or area, and length of each of the pipes in the loop. The gas content of a looped section thus becomes

$$V = \frac{3{,}520}{p_0} \left[p_1 + p_2 - \frac{p_1 p_2}{p_1 + p_2} \right] \sum AL,$$

or $$V = \frac{19.20}{p_0} \left[p_1 + p_2 - \frac{p_1 p_2}{p_1 + p_2} \right] \sum d^2L.$$

The gas content of the entire system will thus be the sum of the contents obtained for the several sections of line. If the quantity thus obtained for average flow conditions is deducted from that similarly computed for minimum flow with maximum intake pressure p_1', the result will be the total available storage capacity of the line.

If the letters used in the two preceding equations be taken to represent the pressure conditions of the unpacked lines and p_1' and p_2' those of the packed line, the available storage capacity of the system will be

$$S = \frac{3{,}520}{p_0} \left[p_1' - p_1 + p_2' - p_2 - \frac{p_1' p_2'}{p_1' + p_2'} + \frac{p_1 p_2}{p_1 + p_2} \right] \sum AL,$$

or

$$S = \frac{19.20}{p_0} \left[p_1' - p_1 + p_2' - p_2 - \frac{p_1' p_2'}{p_1' + p_2'} + \frac{p_1 p_2}{p_1 + p_2} \right] \sum d^2L.$$

The computation of pipe-line storage capacity often involves the principles of flow through complex pipe lines and divided circuits

or loops which are discussed on pages 90 to 95. The following example, taken from "Problems in Natural Gas Engineering," by Thomas R. Weymouth, serves to illustrate one of the more complicated situations involving the storage capacity of multiple loops.

Example.—Assume a complex line made up as shown in Fig. 29, and let it be required to find the equivalent line of single pipe and the intake, or initial pressure p_1 necessary to force 1,000,000 cu ft per hr measured under standard conditions, through the system, with a terminal pressure p_2 = 50 psi abs, a flow temperature of 40 F, and a specific gravity of 0.60. Taking first the section from A to B, we have 20 miles of 8-in. (net) pipe and 25 miles of 10-in. By the divided-circuit solution explained on page 93 the diameter of a pipe 25 miles long that

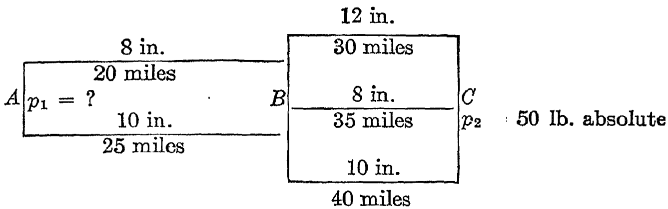


FIG. 29.—Diagram illustrating application of pipe-line formulas.

will be equivalent to the 20 miles of 8-in. pipe is, in inches, using the Weymouth exponents,

$$d_{eq.} = 8\left(\frac{25}{20}\right)^{3/16} = 8.342.$$

Section AB is then equivalent to two parallel or looped lines each 25 miles long and having respective inside diameters of 8.342 in. and 10 in.

For d = 8.342 in., $d^{8/3}$ = 286.2 in. (see Table VII, page 363 for $8/3$ powers.)
For d = 10.0, $d^{8/3}$ = 464.2

$\Sigma d^{8/3}$ = 750.4 in.

and $d_0 = (\Sigma d^{8/3})^{3/8} = (750.4)^{3/8} = 11.97$

or, section AB is equivalent to one pipe 11.97 in. in diameter and 25 miles long.

By the divided-circuit solution on page 93 the 8-in. pipe, 35 miles long, in section BC is equivalent to a 7.772-in. pipe 30 miles long, since

$$d_{eq.} = 8\left(\frac{30}{35}\right)^{3/16} = 7.772.$$

In the same manner, the 10-in. line 40 miles long is equivalent to a 9.475-in. line 30 miles long. Then summing up

for d = 12.0, $d^{8/3}$ = 754.8 (see Table VII, page 363 for $8/3$ powers).
for d = 7.772, $d^{8/3}$ = 237.0
for d = 9.475, $d^{8/3}$ = 402.0

$\Sigma d^{8/3}$ = 1,393.8

$$d_{eq.} = (\Sigma d^{8/3})^{3/8} = (1{,}393.8)^{3/8} = 15.10 \text{ in.}$$

Hence, section BC is equivalent to one pipe 15.10 in. in diameter, 30 miles long, The equivalent length of section BC in terms of the 11.97-in. pipe to which section AB was reduced is obtained by the complex-series method explained on page 95.

$$L_{eq.} = 30\left(\frac{11.97}{15.10}\right)^{5\frac{1}{3}} = 8.68 \text{ miles}$$

Thus, section BC is equivalent to a single line 11.97 in. in diameter and 8.68 miles long, and the whole system is equivalent to a single line 11.97 in. in diameter and 33.68 miles long.

For ascertaining the required initial pressure for a flow of 1,000,000 cu ft per hr, the known line constants are, therefore, $p_2 = 50$, $d = 11.97$, $L = 33.68$, $Q = 1{,}000{,}000$. By the Weymouth equation on page 195,

$$1{,}000{,}000 = 37\sqrt{\frac{p_1^2 - 50^2}{33.68}\,11.97^{5\frac{1}{3}}} \quad p_1 = 215.1 \text{ psi abs.}$$

The relative quantities of gas passed by the several pipes of the loops are to each other as the $\frac{8}{3}d$ power of their equivalent diameters for equal lengths. The pressure at junction B can be ascertained by means of the Weymouth equation on page 195, using for L and d the equivalent values derived above for section AB. This pressure is required to be known in order to determine the storage capacity of the line. After having ascertained the above quantities, the pressure drops being known, the computation can be checked by computing the flow through the several lines as they actually exist.

To compute the storage capacity of the system shown in Fig. 29, first consider section AB. Assume the maximum allowable pressure at A to be $p_1' = 225$ psi abs, and the flow $Q = 800{,}000$ cu ft per hr. This section was shown to be equivalent to a single line 11.97 in. in diameter and 25 miles long. Substituting these known values in the Weymouth equation, the pressure at B is found to be $p_2' = 172.7$ lb.

When the line is unpacked, *i.e.*, under average-flow conditions of 1,000,000 cu ft per hr, p_1 was shown to be 215.1 lb and by formula, p_2 is found to be 117.4 lb.

Substituting these values in the equation on page 196, together with the actual dimensions of the pipe in the section of line under consideration, the available storage capacity of this section AB in cubic feet is

$$\frac{19.20}{14.65}\quad 225 \cdot\cdot 215.1 + 172.7 \quad 117.4 - \frac{225 \times 172.7}{225 + 172.7}$$
$$-\frac{215.1 \times}{215.1} \qquad [(8^2 \times 20) + (10^2 \times 25)]$$

In like manner the storage capacity of BC is computed, the sum of the two results being the total storage capacity of the whole system available for peak demands.

Adiabatic Compression of Natural Gas.[1]—The following table gives the rise in temperature due to the adiabatic compression of natural gas. p_1 is the absolute initial and p_2 the absolute final pressure, p_2/p_1 being, therefore, the ratio of compression. The initial temperature of the gas is assumed to be 60 F.

[1] Reproduction from "National Pipe Standards," by permission of The National Tube Co. For data on the work requirement and temperature rise in adiabatic compression, see "A Series of Enthalpy-entropy Charts for Natural Gases," by George Granger Brown, *AIMME Tech., Publ.* 1747.

p_2/p_1	Rise in temperature, degrees Fahrenheit	p_2/p_1	Rise in temperature, degrees Fahrenheit	p_2/p_1	Rise in temperature, degrees Fahrenheit
1.0	0	6.0	238	14	386
1.5	47	6.5	251	16	412
2.0	82	7.0	263	18	435
2.5	110	7.5	274	20	456
3.0	135	8.0	285	25	503
3.5	157	8.5	296	30	543
4.0	177	9.0	305	35	578
4.5	194	10.0	324	40	609
5.0	210	11.0	341	45	638
5.5	224	12.0	357	50	664

Effect of Altitude on Gas Pressure.—The effect of altitude on low-pressure gas distribution becomes significant in hilly country, or where there are tall office buildings or waterless piston-type gas holders. On account of the altitude or stack effect, pressure on the mains is somewhat greater when such holders are nearly empty than when they are almost full. Likewise the available pressure differential is greater on the upper floors of tall buildings than it is at street level. In many cases the increase of pressure will more than cover the friction drop in a given riser. The difference is greater in the case of a gas of low specific gravity than with a heavier gas. Contrary to water distribution practice where head tanks are located on relatively high ground, waterless piston-type gas holders to be most effective should be located at points of low elevation.

The change in water column corresponding to a given difference in elevation can be computed conveniently as follows:

1 ft of air at 60 F $= \dfrac{0.07638}{62.37} = 0.0012246$ ft of water column.

100 ft of air at 60 F $0.0012246 \times 100 \times 12 = 1.47$ in. of water column.

100 ft of gas at 60 F $= 1.47s$ in. of water column where s is the specific gravity.

The altitude effect of a 100-ft change in elevation is, therefore, $1.47 - (1.47s) = 1.47(1 - s)$ in. of water column, where gas, air, and water columns all are at 60 F. Should any of these be at some other temperature, this can be taken into account readily in the foregoing solution through substituting corresponding values for the density of air or water (see pages 146 and 267, respectively). The altitude effects at 60 F for gas of different specific gravities

are given in Table XXVII. In the case of intermediate or high-pressure gas transmission lines with considerable pressure drop, the effect of altitude is of little consequence and usually is neglected. Those wishing to give the matter consideration, however, can refer to one of the Bureau of Mines reports covering this subject.[1]

TABLE XXVII.—ALTITUDE EFFECTS AT 60 F FOR GAS OF DIFFERENT SPECIFIC GRAVITIES IN INCHES OF WATER COLUMN PER 100 FT DIFFERENCE IN ELEVATION

Specific gravity..	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
Inches H_2O.....	1.32	1.18	1.03	0.88	0.74	0.59	0.44	0.29	0.15

OIL[2]

The lubricating oils and liquid fuels commonly encountered in engineering work are refined from "crude" oil or petroleum. The word petroleum is derived from two Latin terms, "petra" meaning rock and "oleum" meaning oil. Rock oil, which was an early name given it in America, is accounted for by the fact that certain shales and coals contain oil as a constituent. Petroleum is one of the family of bitumens which in their natural state assume many forms and are of world-wide distribution. It usually occurs in connection with pockets of natural gas. The occurrence of petroleum has been observed and recorded from the earliest times. The first extensive commercial exploitation of petroleum started about 1850.

[1] "Flow of Natural Gas through High-pressure Transmission Lines," *op. cit.*, pp. 72–75. See also Bureau of Mines, *Report of Investigations* 3153.

[2] The following references are listed as furnishing a more complete discussion of the properties of various oils than is possible here:

(*a*) "Handbook of the Petroleum Industry," Vol. I, David T. Day, Editor in Chief, John Wiley & Sons, Inc., New York, 1922.

(*b*) "Industrial Oil Engineering," by John Rome Battle, 3d ed., rev. 1926, J. B. Lippincott Company, Philadelphia.

(*c*) "A Handbook of Petroleum, Asphalt, and Natural Gas," by Roy Cross, *Bull.* 25, Kansas City Testing Laboratory, 1928.

(*d*) "National Standard Petroleum Oil Tables," *Circ.* 154, Bureau of Standards, 1924.

(*e*) "ASTM Standards on Petroleum Products and Lubricants" (methods of testing, specifications, definitions, charts, and tables), prepared by ASTM Committee D2. Issued annually by the American Society for Testing Materials, 260 S. Broad St., Philadelphia, Pa. See also ASA and API standards.

(*f*) "The Chemical Technology of Petroleum," by W. A. Gruse and D. R. Stevens, McGraw-Hill Book Company, Inc., New York, 1942.

Gasoline is now the most valuable product of petroleum. Gasoline was originally used for lighting purposes and for domestic stoves, but since the invention of the internal-combustion engine, production has increased tremendously to supply the fuel requirements of automobiles, motor boats, and airplanes. Prior to 1920 most of the gasoline produced was obtained by the fractional distillation of crude oil which yielded only 5 to 6 gal from a 42-gal barrel of crude oil.

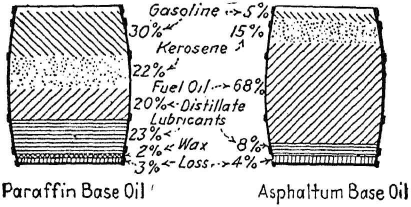


FIG. 30.—Typical examples of products resulting from ordinary refining of paraffin and asphaltum base oils (cracking process not used).

The following explanation of the fractional distillation of crude oil with typical percentages of products obtained is quoted from *U. S. Department of Agriculture Circular* 405:

> Crude oils obtained from different fields vary considerably. They are divided into two main groups, paraffin base oils and asphaltic base oils. The paraffin oils vary in color from a dark green to a light amber and are found principally in the Appalachian regions and mid-continent fields. The asphaltic types are heavier oils and are found in large quantities in California and the Gulf-coast regions. They are darker in color than the paraffin oils and vary from a red brown to black. A great many products are derived from crude oil by slowly heating it and collecting the products given off. (This is known as fractional distillation.) At the lower temperatures gasoline and kerosene are obtained, while at higher temperatures fuel oil, lubricating oils, lubricating grease, and other products are given off, each within a fixed range of temperature. The number of products obtained depends upon the market demand. Figure 30 shows a typical distillation of a paraffin oil and an asphaltic oil in which fuel oil was obtained. Either distillation might have been varied to give a different range of products. In the case of asphaltic oil, for example, the residue after the gasoline and kerosene had been driven off might have been used for road building purposes or further distilled for a number of grades of lubricants without producing any fuel oil.

By 1920 cracking processes had been introduced and developed to a point where the yield was over 10 gal per 42-gal barrel. In the cracking processes the heavier fractions like kerosene, gas oil, and fuel oil which were produced along with gasoline by fractional distillation are rerun in specially built stills under tremendous

heat and pressure, thus cracking the molecules and getting a further proportion of gasoline as a result. The recovery of casing-head gasoline from natural gas has been another important source of high-volatile fuel.

The polymerization process which is the reverse of cracking came into use in the early 1930's. It gathers petroleum refinery gases, formerly largely wasted and blown away, and converts them into liquid fuel. Hydrogenation, or the hydroforming process, originated in Europe and was perfected technically in the United States. Hydroforming was used during World War II to produce synthetic toluene for TNT at a rate equivalent to about twice that produced by the whole coal-tar industry. When the proper naphtha fraction is used, hydroforming will produce high-octane aviation gasoline.

By 1938 gasoline yield by all methods employed in the United States had risen to 18½ gal per 42-gal barrel of crude. In 1938 announcement was made of the Houdry catalytic cracking process which gives 33 to 34 gal of gasoline per barrel of crude. The development of isopropyl and alkylate for blending with leaded gasoline has contributed to the development of 100-octane fuel on a commercial scale.

Advances in the art of refining ultimately promise almost complete flexibility of output, thereby permitting production of any desired percentage of any one of the most important products and thus relieving storage problems created by variation in seasonal demands.

Liquid fuels and lubricants are generally distinguished one from another by their density or heaviness, commonly referred to as the "gravity," and by their viscosity.

Density of Oil.—It is standard American practice to indicate specific gravities of petroleum products at 60 F. The variation of density with temperature is important with petroleum products since most of them are sold by volume and the specific gravity is determined at the prevailing temperature rather than at 60 F. The Bureau of Standards has issued *Circ.* 410 containing tables which enable (1) reducing observed degrees API to degrees API at 60 F, (2) obtaining volume at 60 F occupied by unit volume at indicated temperature, and (3) reducing observed specific gravities to specific gravities at 60/60 F. Hydrometers used for determining the density of oil are graduated with either the Baumé scale (Bureau of Standards) or the API scale (American

Petroleum Institute). Both of these scales are arbitrary and differ but little from each other. They are identical at the 10-degree mark which corresponds to a specific gravity of 1.00, or the density

TABLE XXVIII.—DEGREES BAUMÉ, SPECIFIC GRAVITY, AND POUNDS PER GALLON CORRESPONDING TO EACH DEGREE API

Degrees A.P.I. modulus 141.5	Degrees Baumé modulus 140	Specific gravity at 60/60 F.	Pounds per gallon at 60 F	Degrees A.P.I. modulus 141.5	Degrees Baumé modulus 140	Specific gravity at 60/60 F	Pounds per gallon at 60 F	Degrees A.P.I. modulus 141.5	Degrees Baumé modulus 140	Specific gravity at 60/60 F	Pounds per gallon at 60 F
10	10.00	1.0000	8.328	40	39.68	0.8251	6.870	70	69.36	0.7022	5.845
11	10.99	0.9930	8.270	41	40.67	0.8203	6.830	71	70.35	0.6988	5.817
12	11.98	0.9861	8.212	42	41.66	0.8155	6.790	72	71.34	0.6953	5.788
13	12.97	0.9792	8.155	43	42.65	0.8109	6.752	73	72.33	0.6919	5.759
14	13.96	0.9725	8.099	44	43.64	0.8063	6.713	74	73.32	0.6886	5.731
15	14.95	0.9659	8.044	45	44.63	0.8017	6.675	75	74.31	0.6852	5.703
16	15.94	0.9593	7.989	46	45.62	0.7972	6.637	76	75.30	0.6819	5.676
17	16.93	0.9529	7.935	47	46.61	0.7927	6.600	77	76.29	0.6787	5.649
18	17.92	0.9465	7.882	48	47.60	0.7883	6.563	78	77.28	0.6754	5.622
19	18.90	0.9402	7.830	49	48.59	0.7839	6.526	79	78.27	0.6722	5.595
20	19.89	0.9340	7.778	50	49.58	0.7796	6.490	80	79.26	0.6690	5.568
21	20.88	0.9279	7.727	51	50.57	0.7753	6.455	81	80.25	0.6659	5.542
22	21.87	0.9218	7.676	52	51.56	0.7711	6.420	82	81.24	0.6628	5.516
23	22.86	0.9159	7.627	53	52.54	0.7669	6.385	83	82.23	0.6597	5.491
24	23.85	0.9100	7.578	54	53.53	0.7628	6.350	84	83.22	0.6566	5.465
25	24.84	0.9042	7.529	55	54.52	0.7587	6.316	85	84.20	0.6536	5.440
26	25.83	0.8984	7.481	56	55.51	0.7547	6.283	86	85.19	0.6506	5.415
27	26.82	0.8927	7.434	57	56.50	0.7507	6.249	87	86.18	0.6476	5.390
28	27.81	0.8871	7.387	58	57.49	0.7467	6.216	88	87.17	0.6446	5.365
29	28.80	0.8816	7.341	59	58.48	0.7428	6.184	89	88.16	0.6417	5.341
30	29.79	0.8762	7.296	60	59.47	0.7389	6.151	90	89.15	0.6388	5.316
31	30.78	0.8708	7.251	61	60.46	0.7351	6.119	91	90.14	0.6360	5.293
32	31.77	0.8654	7.206	62	61.45	0.7313	6.087	92	91.13	0.6331	5.269
33	32.76	0.8602	7.163	63	62.44	0.7275	6.056	93	92.12	0.6303	5.246
34	33.75	0.8550	7.119	64	63.43	0.7238	6.025	94	93.11	0.6275	5.222
35	34.74	0.8498	7.076	65	64.42	0.7201	5.994	95	94.10	0.6247	5.199
36	35.72	0.8448	7.034	66	65.41	0.7165	5.964	96	95.09	0.6220	5.176
37	36.71	0.8398	6.993	67	66.40	0.7128	5.934	97	96.03	0.6193	5.154
38	37.70	0.8348	6.951	68	67.39	0.7093	5.904	98	97.07	0.6166	5.131
39	38.69	0.8299	6.910	69	68.38	0.7057	5.874	99	98.06	0.6139	5.109
								100	99.05	0.6112	5.086

For more complete data see Table 5 of the "National Standard Petroleum Oil Tables," *Bur. Standards Circ.* 410. Also ASTM Standard D287.

of water. These scales are explained at greater length and the formula for converting degrees Baumé Bureau of Standards and API into specific gravity or density is given on page 126 in connection with the rational formula for friction loss in pipes. In order to overcome the confusion that has existed in the petroleum-oil

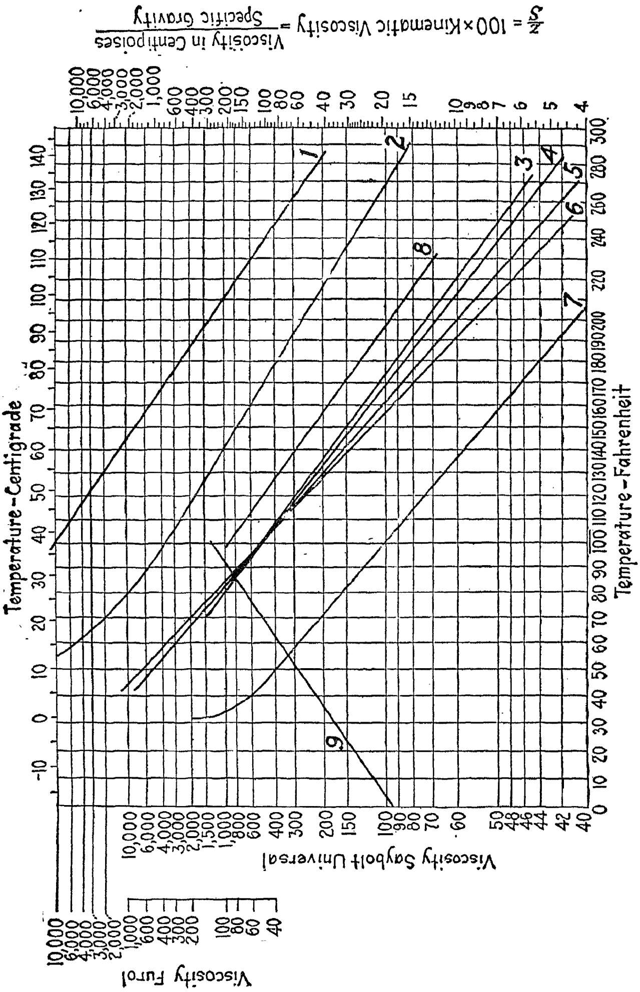


FIG. 31.—Temperature-viscosity relations for petroleum oils.

Blends may be determined on the basis of this chart. Given the viscosity-temperature lines of two oils, their blends will be inversely proportionate to the relative distances to the known lines. *Example:* 8 is a blend of two parts of 4 with one part of 2, the distance from 2 being twice as far as from 4.

To determine the proportions to be used to get a viscosity of 135 at 130 F using stocks having viscosities of 95 and 1,300 at 130. *Example:* Line 9 connects 95 viscosity with 1,300 viscosity crossing all intervening viscosities. Using the scale at the bottom as a percentage scale (0–100) the desired viscosity is shown as calling for 19 per cent of the heavy oil and 81 per cent of the lighter. Actual viscosities (not stock viscosities) must be had for accurate percentage results when using the chart for this purpose.

Conversely, the makeup of an oil may be known if its viscosity and its constituent oils are known. In fact, due to the straight-line characteristics of the chart, many problems which have previously depended on formulas more or less approximate can be settled directly by translation to the chart.

The use of absolute viscosities is becoming noticeable in the commercial as well as the engineering world. The right margin of the chart provides the values from which the absolute viscosity in terms of centipoises may be obtained by multiplying the value given by the specific gravity of the subject oil at the given temperature. To convert to poises (dynes) divide the centipoises by 100. *Example:* Determine the absolute viscosity in dynes of oil represented by line 5 at 120 F. Chart shows the kinematic viscosity of this oil at 120 F to be 56. Specific gravity at 120 F is 0.906. Therefore the absolute viscosity is 56 × 0.906 = 49.7 centipoises or 0.497 poise or dyne.

Specific Heat.—The following discussion of the specific heat, etc., of the petroleum oils, has been briefed from a paper on "Specific Heat—Specific Gravity—Temperature Relations of Petroleum Oils," by William Rankine Eckart, published in *Mechanical Engineering* for July, 1925, pages 535–540:

The value of the specific heat commonly used in practice for liquid hydrocarbons is 0.50 Btu per pound. As values as low as 0.40 and as high as 0.60 have been determined, an error may be involved within these limits as great as ±20 per cent. Any relationship, empirical or otherwise, which may be established so as to make it possible to confine this error within narrower limits should be of estimable value.

The material used for the preparation of the charts presented in this paper has been gathered from all available sources. It is of a truly representative character, and is not limited in any sense to the work of any single investigator, to any particular method, or to samples from any one field.

No positive relationship is claimed herein between the specific heat and specific gravity of petroleum oils, nor is inaccuracy imputed to data which do not agree with the average curves presented. From a practical and engineering standpoint, if not from that of the physicist, it is believed that results warrant the use of these mean data for purposes of design, where exact experimental data are not available for the particular petroleum oil being considered.

Method of Approach.—The study of the problem may be conveniently divided into five steps:

1. The variation of specific gravity with temperature.

2. The variation of specific heat with temperature.

3. The reduction of all data pertaining to specific gravity and specific heat to a standard temperature of 60 F and the determination of a mean relationship between these quantities at this temperature.

4. The preparation of a chart showing the mean relationship between specific gravity and specific heat for the range of the available data.

5. The extrapolation of these data to temperatures as high as 800 F, and the preparation of a chart showing the variation of both specific gravity and specific heat with temperature for this range, or as far as the critical temperature where this is lower than 800 F.

The above-mentioned steps are developed in detail in Eckart's paper until he presents the data in the final form, reproduced here as Fig. 32, regarding which he says:

Below 200 F the specific gravity curves agree with the Bureau of Standards Tables (*Circular* 154). . . . Within this range of temperature (0 to 200 F) the variation of specific gravity with temperature is to all intents and purposes a straight-line function. Wilson and Bahlke have shown that this linear relation does not extend to high temperatures and that, with the exception of the heavier oils, it is not safe to use the linear relation beyond 200 F.

To meet the necessity of data for the design of high-temperature apparatus Fig. 32 [Fig. 3 in Professor Eckart's paper] is presented in which the specific heat and specific gravity have been plotted against temperature up to 800 F. From this chart reasonably accurate values may be obtained of both the specific heat and specific gravity of any oil up to a temperature of 800 F (or up to the critical temperature if this is less than 800 F) when the specific gravity at 60 F is known.

Example.—The following is an example of the use of the specific-gravity–specific-heat temperature chart for petroleum oils shown in Fig. 32. A certain oil has a specific gravity of 0.64 at 60 F. What are its specific gravity and its specific heat at 300 F?

Solution.—Find the curve for 0.64 specific gravity at 60 F and follow it until it intersects the vertical ruling for 300 F. The specific gravity at 300 F is then seen to be 0.48 and the specific heat 0.75 Btu per lb per degree Fahrenheit.

It should be noted that the specific heats read from the chart are *instantaneous* rather than *mean* specific heats. If it is desired to calculate the quantity of heat required to raise the temperature of 1 lb of oil from temperature A to temperature B, it is necessary to determine the average or mean value of the specific heat between the limits A and B. In the above example, for instance, if it is desired to compute the number of Btu required to raise the temperature of 1 lb of that particular oil from 60 to 300 F, proceed as follows: The temperature increase is $300 - 60 = 240$ F. Divide this increase into any convenient number of steps, say six in this case, and average the specific heats read at the mid-points of the successive steps. Instantaneous specific heats will be read then

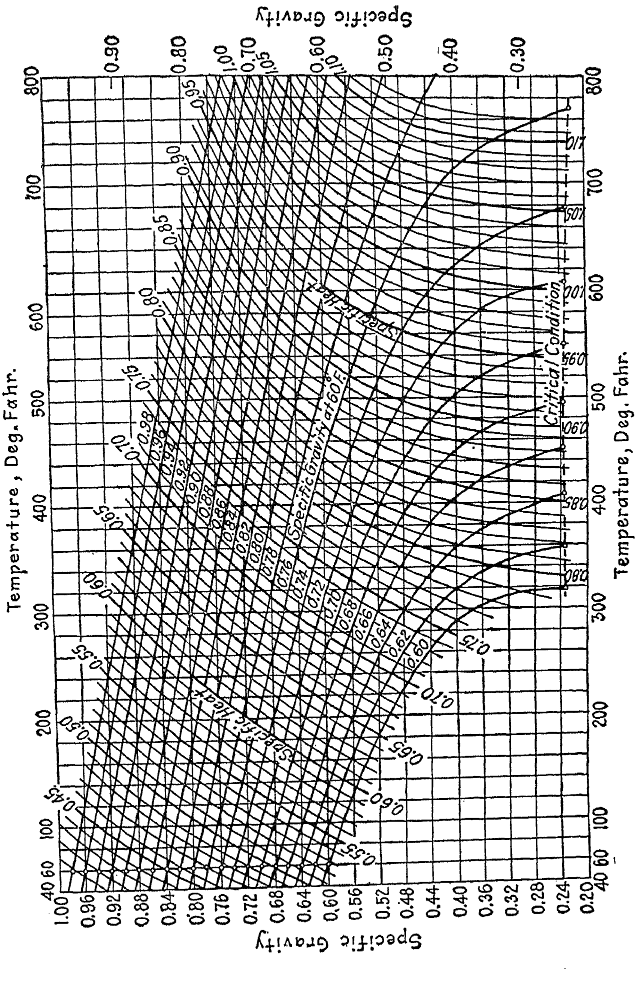


FIG. 32.—Relation between specific gravity, specific heat, and temperature for petroleum oils.

at 80, 120, 160, 200, 240, and 280 F from Fig. 32 and averaged as follows:

At 80 F the instantaneous specific heat is 0.534 Btu/lb/F
At 120 F the instantaneous specific heat is 0.573 Btu/lb/F
At 160 F the instantaneous specific heat is 0.613 Btu/lb/F
At 200 F the instantaneous specific heat is 0.652 Btu/lb/F
At 240 F the instantaneous specific heat is 0.691 Btu/lb/F
At 280 F the instantaneous specific heat is 0.730 Btu/lb/F
Average specific heat 60 to 300 F = 3.793 ÷ 6 = 0.632 Btu/lb/F.

NOTE.—It so happens that the example chosen falls in the portion of the curve where the specific-heat–specific-gravity relation is very nearly a straight-line function. Consequently, in this instance the average of the specific heats at 60 and 300 F happens to give an identical result with that obtained by the more lengthy solution above:

At 60 F the instantaneous specific heat is 0.514 Btu/lb/F
At 300 F the instantaneous specific heat is 0.750 Btu/lb/F
Average specific heat 60 to 300 F = 1.264 ÷ 2 = 0.632 Btu/lb/F.

The heat required to raise the temperature of 1 lb of this oil from 60 to 300 F then is 240 × 0.632 = 151.68 Btu.

Flow of Oil through Pipes.—The calculation of oil flow through pipes is much more complicated than in the case of water. Water is a fluid of well-defined and almost constant physical characteristics within ordinary temperature limits, oil is the opposite; no two oils are exactly alike, and even any one oil is subject to important physical changes under variable temperatures. While the flow of water through pipes offers in itself a sufficiently difficult problem, still the laws governing it have been determined empirically with an exactness sufficient to meet all ordinary practical requirements. The principal difference between oil and water from a pipe-flow point of view lies in the variable viscosity. Ordinary crude oil is not a simple homogeneous liquid such as water—it is a very complex substance composed of compounds of carbon and hydrogen which exist in petroleum in almost bewildering number.

For the reasons stated above, it is impossible to obtain any simple empirical equation which is universally applicable to the flow of oil in pipes. Satisfactory results can be obtained, however, by the use of the so-called "rational flow formulas" given on

TABLE XXIX.—FLOW OF OILS THROUGH COMMERCIAL PIPES
(Based on rational formula; see pages 107 to 137)

Size of pipe and average inside diameter, inches	Capacity, gallons per minute	Pressure drop in pounds per square inch per 100 ft. of pipe based on oils of 20° Bé. gravity — Viscosity in Saybolt Universal seconds: 100	200	300	400	500	600
½ 0.622	2	6.59	14.3	21.7	29.1	36.9	43.6
	5	21.1	34.8	53.4	71.5	89.5	107.0
	7	37.7	49.2	74.6	101.0	129.0	152.0
	10	69.8	85.8	107.0	144.0	177.0	217.0
	15	143.0	174.0	194.0	217.0	267.0	325.0
¾ 0.824	2	2.16	4.74	7.04	9.48	11.6	14.2
	5	5.6	11.5	17.1	23.1	28.9	35.2
	7	9.96	16.1	24.5	32.8	40.5	48.9
	10	18.4	22.3	34.9	46.0	58.5	70.9
	15	37.9	45.7	51.5	70.4	88.3	106.0
1 1.05	5	2.06	4.32	6.68	8.85	11.0	13.4
	10	5.9	8.74	13.4	17.9	22.0	26.6
	15	11.9	14.5	19.7	26.6	33.2	40.3
	20	19.7	23.8	26.8	35.4	44.1	53.5
	25	29.0	35.1	39.3	44.2	54.5	66.8
1½ 1.61	10	0.783	1.57	2.40	3.22	3.98	4.86
	20	2.61	3.17	4.85	6.52	8.02	9.70
	30	5.30	6.43	7.15	9.68	12.1	14.7
	40	8.72	10.6	11.9	12.8	16.0	19.4
	50	12.7	15.7	17.4	18.8	19.8	24.2
2 2.067	10	0.266	0.578	0.875	1.17	1.47	1.79
	20	0.79	1.15	1.75	2.34	2.92	3.52
	30	1.60	1.93	2.62	3.50	4.40	5.30
	40	2.63	3.18	3.56	4.66	5.85	7.10
	50	3.86	4.68	5.28	5.88	7.26	8.85
4 4.026	50	0.174	0.198	0.307	0.412	0.494	0.612
	100	0.550	0.668	0.744	0.810	1.01	1.22
	150	1.09	1.36	1.50	1.62	1.74	1.84
	200	1.81	2.22	2.49	2.68	2.85	2.98
	250	2.67	3.26	3.67	3.97	4.20	4.41
6 6.065	100	0.0788	0.0956	0.115	0.158	0.194	0.236
	200	0.259	0.315	0.359	0.388	0.408	0.480
	500	1.27	1.55	1.74	1.88	2.01	2.20
	700	2.29	2.80	3.15	3.36	3.60	3.78
	1000	4.24	5.21	5.82	6.32	6.68	7.05
8 8.03	200	0.0696	0.0834	0.0947	0.102	0.128	0.155
	500	0.340	0.418	0.459	0.500	0.530	0.556
	1000	1.13	1.37	1.55	1.67	1.78	1.87
	1500	2.27	2.74	3.07	3.30	3.59	3.75
	2000	3.88	4.55	5.15	5.56	5.92	6.22
12 12.05	1000	0.165	0.203	0.228	0.242	0.258	0.272
	2000	0.552	0.670	0.750	0.810	0.866	0.906
	3000	1.15	1.34	1.50	1.63	1.73	1.83
	4000	1.91	2.22	2.50	2.69	2.84	3.00
	5000	2.90	3.34	3.63	3.97	4.22	4.41

To change capacities from gallons per minute to barrels (42 gal) per hour, multiply pressure drops by 1.43. To change from pressure drops per square inch per 100 ft to pressure drops per square inch per mile, multiply by 52.8.

10. Inside-diameter functions of steel pipe (Tables VII to XII, pages 363 to 368).

11. Pipe-friction charts for flow of oil, based on rational formula, Figs. 33*a*, *b*, *c*, *d*, *e*, *f*, and *g*. See also Fig. 22 page 1328.

The data on pressure drop given in Table XXIX were calculated by the rational-flow formula for oil of 20 deg Bé gravity and the

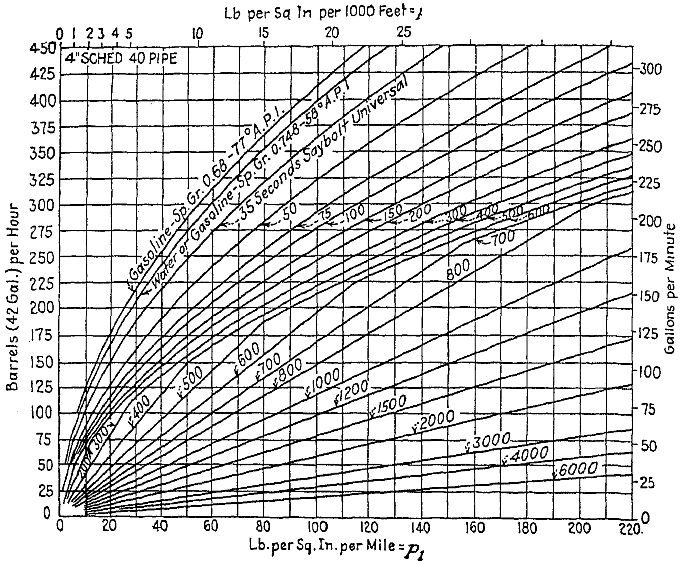


FIG. 33*c*.—Pipe friction based on average Saybolt Universal viscosity and specific gravity of fluid in pipe line. Pressure drop $p\lambda = p_1 \times$ sp. gr.

viscosities indicated. This table is reproduced from "Lubrication," by permission of The Texas Company. See also Table II, page 1330.

The charts shown in Figs. 33 *a*, *b*, *c*, *d*, *e*, *f*, and *g* furnish a simplified method of solving the rational formula for *oil* or *water* for pipe sizes to which they apply. These charts are reproduced from "Standards of The Hydraulic Society," 4th ed., by permission of The Hydraulic Society and the Worthington Pump and Machinery Corporation. Straight lines on the charts cover viscous or stream-line flow, and curved lines cover turbulent flow. It should be noted that the upper lines on each chart have their

humps smoothed out and appear as unbroken curves. As a result, the portions of such curves lying to the left of where the humps should be are somewhat inaccurate. When close results are required in this region, reference should be made to the complete rational solution previously mentioned and shown on pages 107 to 137. Vertical section lines on the charts of Fig. 33 give friction loss p_1 in psi per 1,000 ft and per mile of pipe for an oil having

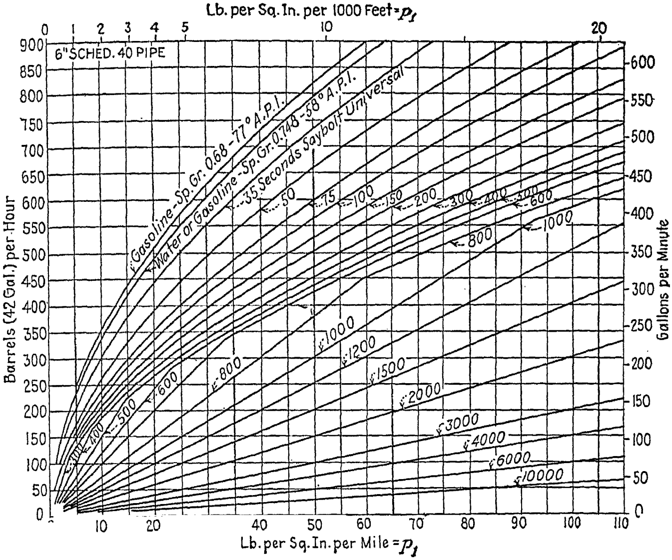


FIG. 33d.—Pipe friction based on average Saybolt Universal viscosity and specific gravity of fluid in pipe line. Pressure drop $p_\lambda = p_1 \times$ sp. gr.

an assumed specific gravity of unity. This value p_1 when multiplied by the actual specific gravity of the oil under the conditions of flow, gives the actual pressure drop per mile of pipe. Horizontal section lines represent the quantity of oil to be pumped in barrels per hour and in gallons per minute. The Saybolt Universal seconds and specific gravities used on the charts are those obtaining at the average temperatures actually existing during flow. See page 204 for a summary of where to find useful data on specific gravities of oil in this handbook. The following example serves to illustrate the use of these charts:

Example.—Six hundred barrel per hr of oil are to be pumped through an 8-in. pipe line 10 miles long. At 60 F the viscosity is 2,500 Saybolt Universal sec and the gravity is 20 degrees API. What is the pressure drop with the flow temperature at 60 F and what will be the drop if the oil is heated so as to maintain an average temperature of 100 F?

Solution for Flow Temperature of 60 *F.*—From Table XVII the specific gravity corresponding to 20 deg API at 60/60 F is 0.9340. Referring to Fig. 33*e*, the value of p_1 for 600 bbl per hr and 2,500 Saybolt Universal sec is 82.5 psi per mile. The pressure drop is $p_\lambda = p_1 \times$ specific gravity $\times$ length in miles $= 82.5 \times 0.9340 \times 10 = 771$ psi.

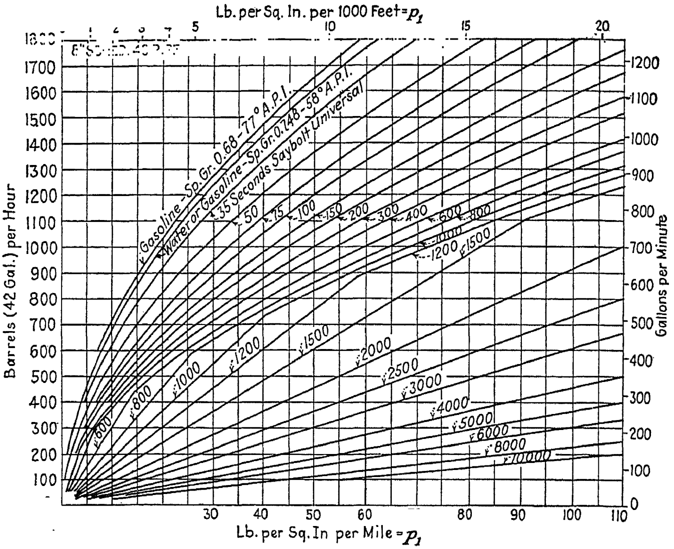


FIG. 33*e*.—Pipe friction based on average Saybolt Universal viscosity and specific gravity of fluid in pipe line. Pressure drop $p_\lambda = p_1 \times$ sp. gr.

NOTE.—A pressure drop of 800 psi was obtained by a check solution of the rational formula, using Fig. 15*a* and the long method.

Solution for Flow Temperature of 100 *F.*—From Fig. 32 the specific gravity of this oil at 100 F is 0.920. From Fig. 31 the viscosity at 100 F is approximately 500 Saybolt Universal sec. Referring to Fig. 33*e* the value of p_1 for 600 bbl per hr and 500 Saybolt Universal sec is 23.75 psi per mile. The pressure drop is $p_\lambda = p_1 \times$ specific gravity $\times$ length in miles $= 23.75 \times 0.920 \times 10 = 218$ psi.

NOTE.—The solution for a flow temperature of 100 F comes in the smoothed-over region of the chart as explained above, and for this reason is not particularly accurate. A pressure drop of 150 psi was obtained by a check solution of the rational formula, using Fig. 15*a* and the long method.

An *empirical formula* for the flow of oil through pipes may be employed for preliminary calculations where accurate results are not essential. If accurate results are desired it is recommended that the final calculations be made by the rational method previously described. The following empirical formula is taken from Marks's "Mechanical Engineers' Handbook":

The resistance to the flow of oils lighter than about 30 deg Bé (specific gravity 0.875) is not much different from that of water. A convenient formula used in

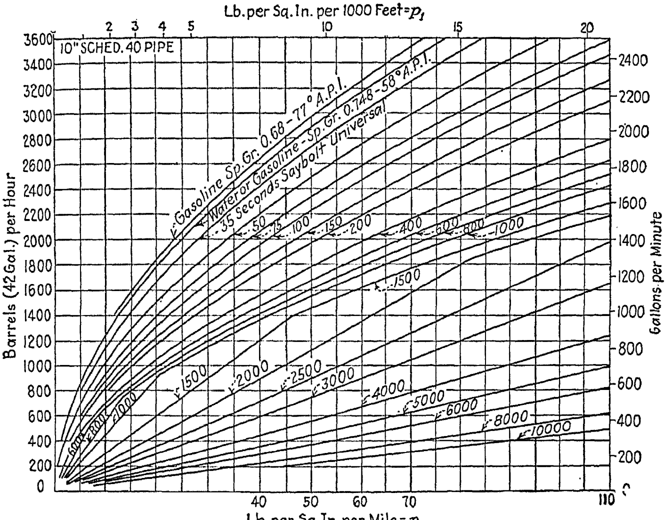


FIG. 33*f*.—Pipe friction based on average Saybolt Universal viscosity and specific gravity of fluid in pipe line. Pressure drop $p_\lambda = p_1 \times$ sp. gr.

the oil fields in designing pipe lines for crude oil (42 to 43 deg. Bé, specific gravity 0.814 to 0.819) is given below. The quantity discharged increases about 1 per cent for each 3 deg. Bé, *i.e.*, lighter oil flows more easily. $B = 1.125\, d^{2.5} \sqrt{p_\lambda / l}$ where B is expressed in barrels (of 42 U.S. gal) per hour; d is actual inside diameter in inches; p_λ is the total friction loss in pounds per square inch; and l is the length of the line in miles. Friction increases with cold oils in winter and decreases in summer. Deposits of paraffin in the pipes conveying crude oil reduce the effective diameter and increase the friction. Scrapers are, therefore, driven through the pipes periodically to clean away the paraffin. Exceedingly viscous asphalt-base oils having a specific gravity of about 0.97 (14 deg Bé) may be pumped through a long line by using a "rifled" pipe and mixing 10 per cent of water with the oil.

The foregoing empirical formula and that which follows in the succeeding paragraph are approximate in that they do not take into account the variation in friction factor with pipe diameter and velocity and that viscosity is considered only indirectly in connection with density. As was pointed out in the discussion of the rational formula on page 107, all of these have a bearing on the numerical value of the friction factor or flow constant.

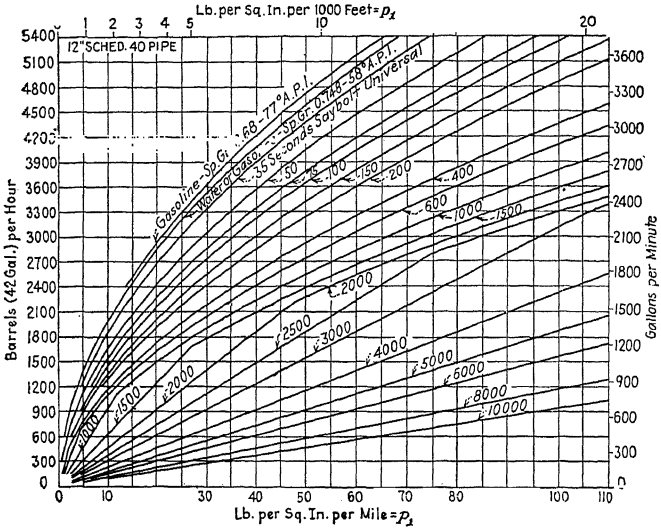


Fig. 33g.—Pipe friction based on average Saybolt Universal viscosity and specific gravity of fluid in pipe line. Pressure drop $p_\lambda = p_1 \times$ sp. gr.

The following empirical equation for oil flow in pipes is taken from the "Union Engineering Handbook" of the Union Steam Pump Company. It is similar to that quoted above from Marks but is accompanied by a more extensive schedule of constants and by Table XXX for the flow of 38 deg Bé oil.

The friction loss in oil pipe lines may be found by the following formula:

$$p_\lambda = \frac{\quad}{10 \times d^5}$$

where p_λ friction, psi per mile (5,280 ft).
C a constant from the table (on page 220) which depends on the character of the oil.
B = barrels of oil per hour (42 gal. per barrel).
d = inside diameter of pipe, in.

Using 9.00 as a *constant* for 38 deg Bé oil, subtract 0.06 for each degree above 38 deg, and add 0.06 for each degree below 38 deg.

For every 10 deg above 60 F, subtract 1 deg from the Baumé reading, and for every 10 deg below 60 F add 1 deg to the Baumé reading.

Example:

Raise 38 deg Bé oil from 60 F to 80 F; then 80 − 60 = 20 and 20 ÷ 10 = 2: and 38 − 2 = 36 or 36 deg Bé, the new Baumé reading for 38 deg Bé oil heated from 60 to 80 F.

Table XXX.—Friction of Oil[1]—38 Deg Bé
(Pounds per square inch in pipes 1 mile long)

Barrels per hour	Diameter of pipe						
	2 in.	3 in.	4 in.	5 in.	6 in.	8 in.	10 in.
10	2.8	0.37	0.088	0.0288	0.0116	0.00274	0.00091
15	6.2	0.83	0.198	0.063	0.026	0.0064	0.00203
20	11.2	1.48	0.352	0.115	0.0462	0.011	0.0036
25	17.6	2.3	0.55	0.18	0.0725	0.0172	0.0056
30	25.3	3.32	0.792	0.26	0.104	0.0247	0.0081
35	34.5	4.52	1.08	0.353	0.142	0.0337	0.0112
40	45.0	5.9	1.43	0.46	0.185	0.044	0.0144
45	57.0	7.49	1.78	0.58	0.235	0.0556	0.0181
50	70.0	9.24	2.2	0.72	0.29	0.0686	0.0225
55	85.0	11.2	2.26	0.87	0.35	0.084	0.0273
60	101.0	13.3	3.16	1.04	0.416	0.099	0.0325
65	118.4	15.6	3.72	1.22	0.49	0.116	0.0381
70	138.0	18.1	4.3	1.41	0.568	0.136	0.0442
75	158.0	20.8	4.95	1.62	0.651	0.155	0.0508
80	180.0	23.6	5.62	1.84	0.74	0.176	0.0578
85	203.0	26.7	6.35	2.08	0.836	0.198	0.0652
90	227.0	30.0	7.11	2.33	0.939	0.224	0.073
95	253.0	33.4	7.92	2.6	1.045	0.248	0.0814
100	280.0	37.0	8.8	2.88	1.16	0.275	0.0902
125	440.0	57.8	13.7	4.5	1.81	0.43	0.142
150	630.0	83.0	19.8	6.5	2.6	0.618	0.203
175	.....	113.0	27.0	8.8	3.54	0.84	0.275
200	.....	148.0	35.2	11.5	4.63	1.1	0.36
225	.....	187.0	42.5	14.6	5.85	1.39	0.455
250	.....	230.0	55.0	18.0	7.22	1.72	0.56
275	.....	280.0	66.5	21.7	8.75	2.08	0.68
300	.....	332.0	79.2	26.0	10.4	2.47	0.81
325	.....	390.0	93.0	30.4	12.2	2.9	0.95
350	.....	453.0	10.2	35.3	14.2	3.37	1.12
375	.....	.....	124.0	40.5	16.3	3.85	1.26
400	.....	.....	141.0	46.0	18.5	4.4	1.44
425	.....	.....	159.0	52.0	20.9	4.96	1.63
450	.....	.....	178.0	58.2	23.5	5.56	1.83
475	.....	.....	198.0	65.0	26.1	6.25	2.04
500	.....	.....	220.0	72.0	28.9	6.88	2.25
525	.....	.....	242.0	79.5	31.9	7.58	2.48
550	.....	.....	266.0	87.0	35.0	8.3	2.72

[1] Based on empirical formula of Union Steam Pump Co. described above. When p_λ is given and B computed, add 1 per cent to B for every 3 deg above and subtract 1 per cent from B for every 3 deg below 38 deg Bé. When B is given and p_λ computed subtract 2 per cent from p_λ for every 3 deg above and add 2 per cent for every 3 deg below—or interpolate for 9.00.

TABLE XXX.—(*Concluded*)

Barrels per hour	Diameter of pipe				Barrels per hour	Diameter of pipe	
	5 in.	6 in.	8 in.	10 in.		8 in.	10 in.
550	87.0	35.0	8.3	2.72	3,600	356	117
575	95.0	38.4	9.1	2.98	3,800	397	130
600	104.0	41.7	9.9	3.24	4,000	440	144
625	112.0	45.4	10.7	3.5	4,200	...	158
650	122.0	49.0	11.6	3.8	4,400	...	174
675	131.0	53.0	12.5	4.1	4,600	...	192
700	141.0	56.9	13.5	4.42	4,800	...	207
725	152.0	61.0	14.4	4.73	5,000	...	225
750	162.0	65.3	15.5	5.06	5,200	...	243
775	173.0	69.8	16.5	5.41	5,400	...	262
600	184.0	74.3	17.6	5.76	5,600	...	282
825	196.0	79.0	18.7	6.12	5,800	...	303
850	208.0	83.9	19.8	6.5	6,000	...	324
875	221.0	88.8	21.1	6.89	6,200	...	346
900	233.0	94.0	22.3	7.3	6,400	...	368
925	247.0	99.2	23.5	7.7	6,600	...	392
950	260.0	104.5	24.8	8.12	6,800	...	416
975	274.0	111.0	26.2	8.55	7,000	...	441
1,000	288.0	115.8	27.5	9.0			
1,100	348.0	140.0	33.2	10.9			
1,200	415.0	166.5	39.5	13.0			
1,300	486.0	195.5	46.4	15.2			
1,400	.....	227.0	53.8	17.6			
1,500	.....	261.0	61.8	20.3			
1,600	.....	296.0	70.4	23.1			
1,700	.....	335.0	79.3	26.1			
1,800	.....	375.0	89.0	29.2			
1,900	.....	418.0	99.0	32.5			
2,000	.....	463.0	110.0	36.1			
2,200	.....	.....	133.0	43.6			
2,400	.....	.....	158.0	52.0			
2,600	.....	.....	186.0	61.0			
2,800	.....	.....	216.0	70.6			
3,000	.....	.....	247.5	81.1			
3,200	.....	.....	284.0	92.2			
3,400	.....	.....	318.0	104.0			
3,600	.....	.....	356.0	117.0			

CONSTANT C FOR DIFFERENT OILS

Degrees, Baumé A. P. I.	Constants	Degrees, Baumé A. P. I.	Constants	Degrees, Baumé A. P. I.	Constants	Degrees, Baumé A. P. I.	Constants
65	7.38	53	8.10	41	8.82	29	9.54
62	7.56	50	8.28	38	9.00	26	9.72
59	7.74	47	8.46	35	9.18	23	9.90
56	7.92	44	8.64	32	9.36	20	10.08

The *economics* of oil flow through pipes is of prime importance in cross-country transmission lines where the capital costs of different size lines must be balanced against their respective pumping costs. Another interesting problem arises in designing loading systems

for tank ships, cars, or trucks where a three-way balance should be struck between pipe cost, pumping cost, and cost of delay to the transport equipment. Owing to the relatively high cost of valves, it sometimes is advantageous to select them one or two sizes smaller than the pipe line.[1]

Flow of Gasoline through Pipes.—Gasoline has a relatively low viscosity compared to other liquids (see Fig. 16, page 119). Relative to water which by definition has a viscosity of 1 centipoise at 68 F, gasoline has a viscosity of 0.45 to 0.85 centipoise depending upon its density. This fact, together with the decreasing effect which viscosity has on fluid motion in the turbulent range, makes it possible to solve gasoline flow problems with reasonable accuracy by empirical formulas. A modification of the Williams-Hazen hydraulic formula (see pages 276 to 279) has been employed extensively for this purpose.[2] Modified to include the specific gravity of the gasoline and expressing the friction loss p_λ as pounds per square inch per 1,000 ft of line, B in barrels (42 gal) per hour, and with the other symbols as defined on page 82, the Williams-Hazen formula may be written:

$$p_\lambda = \frac{2{,}324B^{1.85}S}{C^{1.85}d^{4.87}}.$$

In this form the Williams-Hazen equation contains all the factors of the rational solution except the viscosity of the gasoline which is assumed to be constant with minor variations neglected.

Numerical values of the Williams-Hazen coefficient C usually obtained in clean gasoline lines range from 125 to 150. Good

[1] For a discussion of all these phases of the economic problem, reference may be made to "Selection of the Most Economical Pipe and Valve Size and Rate of Flow in Piping Systems," by S. P. Johnson and F. L. Maker, *Proc. API*, Division of Refining, Mid-year Meeting, May 29, 1940. Also published in *Refiner and Natural Gasoline Manufacturer*, June, 1940, Vol. 19, No. 6, pp. 169–180. *Author's Note:* Where the factor (3.6^3) appears in the formulas of this reference it seems to have been transposed between numerator and denominator and should be reversed, with corresponding corrections in subsequent expressions. As far as has been determined, these errata do not affect the diagrams which, though ostensibly plotted from these formulas, are understood to have been actually plotted from earlier and somewhat different expressions.

[2] See (*a*) "A Discussion of Flow Formulas Used for Design of Gasoline Pipe Lines," by W. E. Rea, *Oil Gas J.*, Jan. 23, 1941, pp. 38–47.

(*b*) "Application of Pipe-line Efficiency Concept to Gasoline Pipe-line Calculations," by Benjamin Miller, *Oil Gas J.*, Vol. 41 (35), pp. 122–123, Jan. 7, 1943.

(*c*) "Use of Scrapers Maintains High C-Factors on Sohio-operated Lines," by R. L. Harris and E. F. Morrill, *Oil Gas J.*, Vol. 42 (27), pp. 244–248, Nov. 11, 1943.

average values for 4, 6, and 8 in. nominal diameter lines are said to be 140, 135, and 130, respectively, which is contrary to most published data which show an increase in C with pipe diameter [see Note 1(b) on page 221].

The rational formula also is used extensively in designing gasoline lines, but its successful application depends on the selection of a suitable friction factor f. The factors given in Fig. 15 have been found too low for gasoline flow in commercial pipe lines, although values given by Drew, Koo, and McAdams in Vol. 28, pages 56 to 72, of the *Trans. Amer. Inst. Chem. Engrs.* are considered to produce satisfactory results.

Flow of Mixtures of Oil and Gas.—Two-phase flow involving the simultaneous passage of a gas or vapor and liquid in a pipe is

TABLE XXXI.—CHARACTERISTICS OF TESTED OILS[1]

Oils tested	Gravity		Pounds per gallon
	Degrees Baumé	Specific gravity	
California, Bakersfield	16.0	0.9595	7.994
Louisiana, Jennings	24.0	0.9105	7.585
Ohio, Lima	37.0	0.8395	6.994
Oklahoma residuum	28.0	0.8870	7.390
Oklahoma residuum	24.0	0.9105	7.585
Oklahoma crude, (G. P.)	32.2	0.8631	7.198
Oklahoma crude	36.2	0.8423	7.023
Oklahoma crude	35.4	0.8464	7.055
Pennsylvania	43.7	0.8059	6.722
Pennsylvania	38.2	0.8323	6.944
Russian, Baku	29.0	0.8815	7.344
Texas, Beaumont	22.0	0.9220	7.681
Texas, Sour Lake	20.0	0.9340	7.781
Texas, residuum	18.0	0.9465	7.885
Texas, gas oil	27.9	0.8866	7.394
West Virginia	40.0	0.8250	6.873

[1] Reproduced from the "Union Engineering Handbook," 5th ed., by permission of the Union Steam Pump Co.

encountered in tube stills, refrigerator systems, condensate return lines, etc. A method of computing the pressure drop in a pipe in which known quantities of vapor and oil or other liquid are flowing simultaneously has been presented by Martinelli[1] *et al.* In an example in which 0.50 lb of air and 0.30 lb of oil per sec were

[1] "Isothermal Pressure Drop for Two-phase Two-component Flow in a Horizontal Pipe," by R. C. Martinelli, L. M. K. Boelter, T. H. M. Taylor, E. G. Thomsen, and E. H. Morrin, *Trans. ASME*, Vol. 66, No. 2, 1944, pp. 139–151. Contains 4 references and a bibliography of 15 related articles.

TABLE XXXII.—APPROXIMATE HEATING VALUE *vs.* DENSITY OF AVERAGE FUEL OILS[1]

Gravity		Weight	Calorific value		Gravity		Weight	Calorific value	
Degrees Baumé[2]	Specific gravity at 60/60 F	Pounds per gallon at 60 F	B.t.u. per pound	B.t.u. per gallon at 60 F	Degrees Baumé[2]	Specific gravity at 60/60 F	Pounds per gallon at 60 F	B.t.u. per pound	B.t.u. per gallon at 60 F
10	ONE	8.331	18,650	158,090	51	0.7750	6.457	20,290	131,013
11	0.9930	8.273	18,690	154,622	52	0.7710	6.423	20,330	130,580
12	0.9860	8.214	18,730	153,848	53	0.7670	6.390	20,370	130,164
13	0.9790	8.156	18,770	153,088	54	0.7630	6.357	20,410	129,746
14	0.9720	8.098	18,810	152,323	55	0.7585	6.319	20,450	129,224
15	0.9655	8.044	18,850	151,629	56	0.7545	6.286	20,490	128,804
16	0.9595	7.994	18,890	151,006	77	0.7505	6.252	20,530	128,354
17	0.9530	7.939	18,930	150,285	58	0.7470	6.223	20,570	128,007
18	0.9465	7.885	18,970	149,578	59	0.7430	6.190	20,610	127,576
19	0.9400	7.831	19,010	148,867	60	0.7390	6.157	20,650	127,142
20	0.9340	7.781	19,050	148,228	61	0.7355	6.127	20,690	126,768
21	0.9230	7.731	19,090	147,585	62	0.7315	6.094	20,730	126,229
22	0.9220	7.681	19,130	146,928	63	0.7280	6.065	20,770	125,970
23	0.9165	7.635	19,170	146,363	64	0.7240	6.032	20,810	125,526
24	0.9105	7.585	19,210	145,708	65	0.7205	6.002	20,850	125,142
25	0.9045	7.536	19,250	145,068	66	0.7165	5.969	20,890	124,692
26	0.8990	7.490	19,290	144,482	67	0.7130	5.940	20,930	124,342
27	0.8930	7.440	19,330	143,815	68	0.7095	5.911	20,970	123,954
28	0.8870	7.390	19,370	143,144	69	0.7060	5.882	21,010	123,580
29	0.8815	7.344	19,410	142,547	70	0.7025	5.853	21,050	123,206
30	0.8755	7.294	19,450	141,868	71	0.6990	5.823	21,090	122,807
31	0.8700	7.248	19,490	141,263	72	0.6955	5.794	21,130	122,327
32	0.8650	7.206	19,530	140,733	73	0.6920	5.765	21,170	122,045
33	0.8595	7.160	19,570	140,121	74	0.6890	5.740	21,210	121,7[illegible]5
34	0.8545	7.119	19,610	139,604	75	0.6855	5.711	21,250	121,369
35	0.8490	7.073	19,650	138,984	76	0.6820	5.682	21,290	120,970
36	0.8440	7.031	19,690	138,440	77	0.6790	5.657	21,330	120,664
37	0.8395	6.994	19,730	137,992	78	0.6755	5.628	21,370	120,270
38	0.8345	6.952	19,770	137,441	79	0.6720	5.598	21,410	119,843
39	0.8295	6.911	19,810	136,907	80	0.6690	5.573	21,450	119,541
40	0.8250	6.873	19,850	136,429	81	0.6655	5.544	21,490	119,141
41	0.8205	6.836	19,890	135,960	82	0.6620	5.515	21,530	118,739
42	0.8155	6.794	19,930	135,405	83	0.6585	5.486	21,570	118,333
43	0.8110	6.756	19,970	134,917	84	0.6545	5.453	21,610	117,839
44	0.8065	6.719	20,010	134,447	85	0.6510	5.423	21,650	117,480
45	0.8015	6.677	20,050	133,874	86	0.6480	5.398	21,690	117,083
46	0.7970	6.640	20,090	133,398	87	0.6450	5.373	21,730	116,755
47	0.7925	6.602	20,130	132,398	88	0.6420	5.349	21,770	116.448
48	0.7885	6.569	20,170	132,497	89	0.6390	5.324	21,810	116,116
49	0.7840	6.632	20,210	132,012	90	0.6365	5.303	21,850	115,871
50	0.7795	6.494	20,250	131,504	91	0.6330	5.274	21,890	115,448

[1] Reproduced from the "Union Engineering Handbook," 5th ed., by permission of the Union Steam Pump Co. See also "Thermal Properties of Petroleum Products," *Bur. Standards, Misc. Pub.* 97.

[2] For relation of API and Baumé scales to specific gravities, see Table XXVIII, p. 203.

flowing in a 2-in. pipe, it was shown that the resulting pressure drop for the two-phase flow was about fifteen times the pressure drop of either the air or oil flowing alone. In this example the air flow was turbulent, while the oil flow was viscous. Similar results were found for an air and water combination. Articles on the flow of flashing mixtures of water and steam through pipes and orifices are referenced on page 915.

Useful Information Regarding Oil.—The following useful information regarding oil is taken from the "Union Engineering Handbook":

1 bbl = 42 U.S. gal.
1 bbl per hr = 0.7 U.S. gpm.
Barrels per hr × 0.7 = gpm.
Gallons per min divided by 0.7 = bbl per hr.
Barrels per hr × 24 = bbl per day.
1 bbl per day = 0.0292 gpm.
Barrels per day × 0.0292 = gpm.
Gallons per min divided by 0.0292 = bbl per day.
Number of bbl in pipe 1 mile long equals inside diameter of pipe in inches squared × 5⅛.
Velocity in ft per sec = 0.0119 × bbl per day divided by diameter of pipe in inches squared; or velocity equals 0.2856 × bbl per hr divided by the diameter of pipe in inches squared, or velocity equals 0.408 × gal per min divided by the diameter of pipe in inches squared.
Net horsepower = the theoretical horsepower necessary to do the work.
Net horsepower = barrels per day × pressure × 0.000017.
Net horsepower = barrels per hour × pressure × 0.000408.
Net horsepower = gallons per minute × pressure × 0.000583.

The characteristics of tested oils and the characteristics of averaged oils given in Tables XXXI and XXXII were also taken from the same source.

PROPERTIES OF STEAM

The Formation of Steam.—Many substances can exist in more than one state under the proper conditions of temperature and pressure. Water exists as ice at low temperatures and as steam at higher temperatures, the temperature depending upon the pressure. If we apply heat to a vessel partly filled with cold water,

the temperature of the water will rise until a certain temperature is reached at which small particles of water are changed into steam. The steam bubbles rise through the mass of water and escape from the surface. The water is then said to boil. The temperature at which the water boils depends upon the pressure in the vessel. If the pressure is raised, as by partly closing the outlet, the temperature of the water will rise to the point corresponding to the existing pressure.

Steam, when still in contact with the water from which it is produced, remains at the temperature corresponding to its pressure and under this condition the steam is said to be saturated. Saturated steam is a vapor as defined on page 27, while superheated steam is a gas which follows more or less closely the laws of "perfect gases." If saturated steam is removed from contact with the water and further heated, its temperature will rise and the steam will then be superheated.

Superheated Steam.—Superheated steam is steam at a temperature higher than the temperature of the boiling point corresponding to the pressure. If water were to be intimately mixed with superheated steam, some of the heat in the steam would be used in evaporating the water and the temperature of the steam would be lowered. If sufficient water were added, the superheat would be entirely used up in evaporating the water and the steam would then be saturated. Superheated steam can have any temperature higher than that of the boiling point.

Saturated Steam.—When steam is at the temperature of the boiling point corresponding to its pressure, it is said to be saturated. If this saturated steam contains no suspended moisture, it is said to be dry saturated steam, or, in other words, dry saturated steam is steam at the temperature of the boiling point and containing no water in suspension. If heat is added to dry saturated steam, not in the presence of water, it will become superheated. If heat is taken away from dry saturated steam it will become wet steam. Dry saturated steam is not a perfect gas and the relations of its pressure, volume, and temperature do not follow any simple law but have been determined by experiment. The properties of dry saturated steam were originally determined by Regnault and published in the year 1847. So carefully was his work done that no errors in his results were apparent until within very recent years, when the great difficulty of obtaining steam which is exactly dry and saturated became appreciated, and new experiments by various

scientists proved that Regnault's results were slightly high at some pressures and slightly low at others.

Properties of Steam.—The heat used in the formulation of 1 lb of superheated steam at any pressure from water at 32 F may be divided into three parts: (1) the heat of the liquid, which is the heat required to raise the temperature of the water from 32 F to the temperature of the boiling point; (2) the latent heat of vaporization, which is the amount required to change one pound of water at the temperature of the boiling point to dry saturated steam at the same temperature; and (3) the "heat of superheat" or, more simply, the superheat, which is the heat added to 1 lb of steam to raise it from the boiling-point temperature to the final temperature.

Heat of the Liquid.—The heat of the liquid may be determined for any boiling-point temperature by the expression

$$h = c(t - 32)$$

in which h = the heat of the liquid, Btu per lb.

t = the boiling-point temperature, degrees F.

c = the specific heat of water, Btu per lb per degree F.

For approximate results c may be taken as = 1.

The change in the volume of the water during the increase in temperature is extremely small, and the amount of external work done may be neglected and all the heat of the liquid may be considered as going to increase the heat energy of the water.

The heat of the liquid, together with the other properties of saturated steam, is given in Table XXXIV for various steam pressures. This table is condensed from Marks and Davis's complete tables.[1]

Latent Heat.—The latent heat of steam has been defined as the heat required to convert 1 lb of water at the temperature of the boiling point into dry saturated steam at the same temperature. Experiments show that the latent heat, usually designated by L, diminishes as the pressure increases.

When water is changed into steam, the volume is greatly increased, so that a considerable portion of the latent heat is used in doing external work. The remainder may be considered as being utilized in changing the physical state of the water. Let p

[1] For values based on more recent investigations, see "Thermodynamic Properties of Steam Including Data for the Liquid and Solid Phases," by J. H. Keenan and F. H. Keyes, John Wiley & Sons, Inc., New York, 1936.

be the pressure in pounds per square inch absolute at which the steam is generated, V the volume of 1 lb of steam, and v the volume of 1 lb of water both measured in cubic feet; then the external work done in foot-pounds is equal to

$$W = 144p(V - v).$$

At 212 F the external work done in producing 1 lb of steam is equivalent to 73 Btu or about one-thirteenth of the latent heat.

Experiments show that the latent heat of steam diminishes about 0.695 heat unit for each degree that the temperature of the boiling point is increased. If t be the temperature of the boiling point, then approximately,

$$L = 1072.6 - 0.695(t - 32).$$

When steam condenses, the same amount of heat is given up as was required to produce it.

Total Heat of Steam.—The total heat[1] of dry saturated steam is the heat required to change 1 lb of water at 32 F into dry saturated steam. This quantity will be designated by H, and

$$H = h + L.$$

The experimental results given in the table for the value of the total heat may be approximated very closely by means of the formula

$$H = 1,072.6 + 0.305(t - 32).$$

It is more accurate, however, to take the values of the total heat from the tables than it is to compute them from the formula. Approximate values can be read from the Mollier chart reproduced in Fig. 34. The total heat in 1 lb of steam under any condition of moisture or superheat is the amount of heat required to change it from water at 32 F to its existing condition.

When steam contains entrained water, the percentage by weight of dry steam in the mixture is termed the "quality" of the steam. If we let q represent the quality of the steam. then the latent heat in 1 lb of wet steam equals

$$\frac{qL}{100}$$

and the total heat in 1 lb of wet steam equals

$$h + \frac{qL}{100}.$$

[1] Total heat, latent heat, and heat of the liquid are referred to as "enthalpy" in most recent works on thermodynamics.

The total heat of *superheated steam* is the total heat of dry saturated steam at the same pressure plus the amount of heat required to produce the number of degrees of superheat present. If C_p represents the mean specific heat at constant pressure for

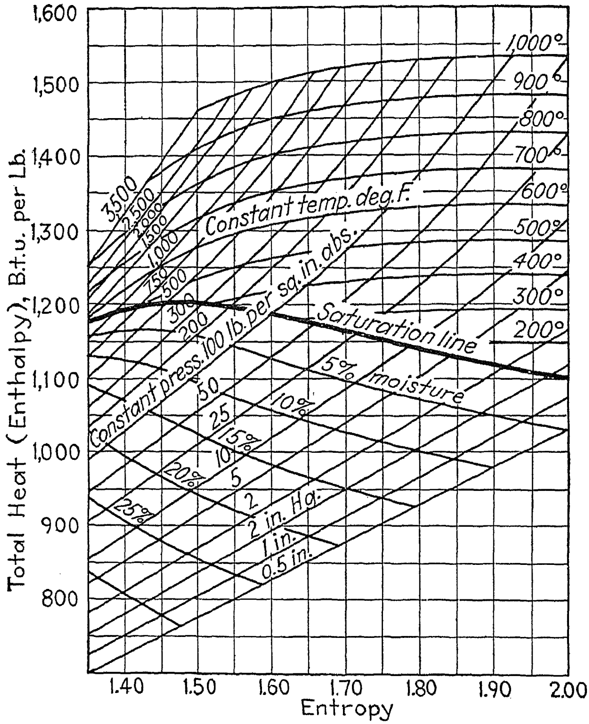


FIG. 34.—Total heat (enthalpy) *vs.* entropy.

superheated steam under the particular conditions of its formation, and t_s the number of degrees Fahrenheit superheat, then its total heat H is

$$H = h + L + C_p t_s.$$

Specific Heat of Superheated Steam.—The specific heat of superheated steam used in ordinary engineering calculations is the *mean specific heat of constant pressure.* Superheated steam,

in common with all gases, has two specific heats, one at constant volume and the other at constant pressure. The distinction between the specific heats at constant volume and constant pressure is explained in connection with "Air," on page 145. The specific heat at constant pressure varies with the temperature in addition to having different values at different pressures. The *mean specific heat at constant pressure* is the heat of superheat (in Btu) added above the saturation temperature under constant pressure conditions, divided by the number of degrees of superheat produced. This *mean specific heat* should be distinguished from the *instantaneous specific heat at constant pressure*, which is the amount of heat required to raise the temperature of 1 lb of steam 1 F above some particular temperature while at some particular pressure which is maintained constant. The mean specific heats given in Table XXXIII were computed from the ASME steam tables.

TABLE XXXIII.—MEAN SPECIFIC HEAT OF SUPERHEATED STEAM, BTU PER POUND PER DEGREE FAHRENHEIT

Absolute pressure, pounds per square inch	Superheat, degrees Fahrenheit				
	10	100	200	300	400
1	0.45	0.45	0.45	0.45	0.45
15	0.49	0.48	0.47	0.47	0.47
200	0.66	0.60	0.57	0.55	0.54
400	0.78	0.69	0.64	0.61	0.59
600	0.91	0.77	0.70	0.65	0.63
800	1.03	0.85	0.75	0.70	0.67
1000	1.17	0.94	0.81	0.75	0.70
1200	1.33	1.03	0.87	0.79	0.74
1400	1.52	1.11	0.93	0.84	0.78

Steam Tables.—The properties of *saturated* steam from 32 to 600 F are given in Table XXXIV. This is a so-called "temperature" table in which temperature is used as the reference by which the other properties of saturated steam are grouped.

Data for these *saturated* steam tables are taken in part from Marks and Davis's tables and diagrams of the "Thermal Properties of Saturated and Superheated Steam" by permission of the publishers, Longmans, Green and Co. Tabular values for temperatures below 400 F are substantially in agreement with most recent data. Values above 400 F do not differ more than 2 per cent from values given by Keenan and Keyes (see page 226). These tables

The total heat of *superheated steam* is the total heat of dry saturated steam at the same pressure plus the amount of heat required to produce the number of degrees of superheat present. If C_p represents the mean specific heat at constant pressure for

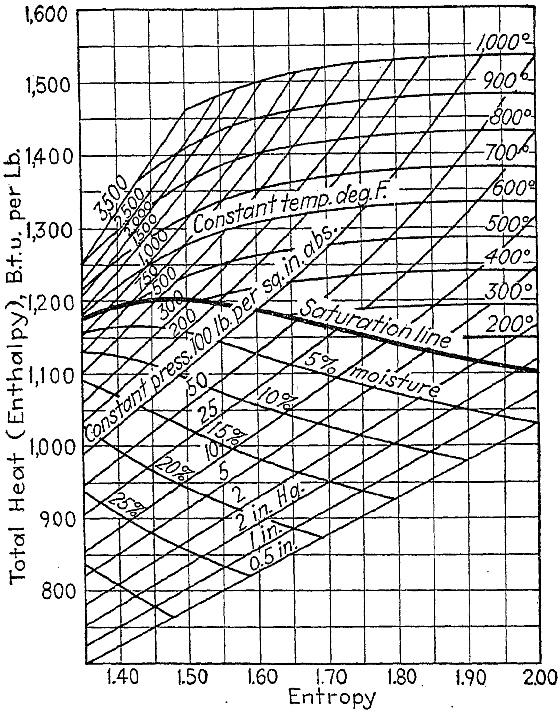


FIG. 34.—Total heat (enthalpy) *vs.* entropy.

superheated steam under the particular conditions of its formation, and t_s the number of degrees Fahrenheit superheat, then its total heat H is

$$H = h + L + C_p t_s.$$

Specific Heat of Superheated Steam.—The specific heat of superheated steam used in ordinary engineering calculations is the *mean specific heat of constant pressure*. Superheated steam,

in common with all gases, has two specific heats, one at constant volume and the other at constant pressure. The distinction between the specific heats at constant volume and constant pressure is explained in connection with "Air," on page 145. The specific heat at constant pressure varies with the temperature in addition to having different values at different pressures. The *mean specific heat at constant pressure* is the heat of superheat (in Btu) added above the saturation temperature under constant pressure conditions, divided by the number of degrees of superheat produced. This *mean specific heat* should be distinguished from the *instantaneous specific heat at constant pressure*, which is the amount of heat required to raise the temperature of 1 lb of steam 1 F above some particular temperature while at some particular pressure which is maintained constant. The mean specific heats given in Table XXXIII were computed from the ASME steam tables.

TABLE XXXIII.—MEAN SPECIFIC HEAT OF SUPERHEATED STEAM, BTU PER POUND PER DEGREE FAHRENHEIT

Absolute pressure, pounds per square inch	Superheat, degrees Fahrenheit				
	10	100	200	300	400
1	0.45	0.45	0.45	0.45	0.45
15	0.49	0.48	0.47	0.47	0.47
200	0.66	0.60	0.57	0.55	0.54
400	0.78	0.69	0.64	0.61	0.59
600	0.91	0.77	0.70	0.65	0.63
800	1.03	0.85	0.75	0.70	0.67
1000	1.17	0.94	0.81	0.75	0.70
1200	1.33	1.03	0.87	0.79	0.74
1400	1.52	1.11	0.93	0.84	0.78

Steam Tables.—The properties of *saturated* steam from 32 to 600 F are given in Table XXXIV. This is a so-called "temperature" table in which temperature is used as the reference by which the other properties of saturated steam are grouped.

Data for these *saturated* steam tables are taken in part from Marks and Davis's tables and diagrams of the "Thermal Properties of Saturated and Superheated Steam" by permission of the publishers, Longmans, Green and Co. Tabular values for temperatures below 400 F are substantially in agreement with most recent data. Values above 400 F do not differ more than 2 per cent from values given by Keenan and Keyes (see page 226). These tables

in their present form are reproduced from "Superheat Engineering Data," through the courtesy of the Superheater Company.

Vacuums in Table XXXIV are given in inches of mercury referred to a 30-in. barometer at a mercury temperature of 58.4 F, which is arrived at as follows: Each degree Fahrenheit change in the temperature of a column of mercury changes its height 0.000101 times. If the height is increased from 29.92 to 30 in., the mean height is 29.96 in., and the mean change for each degree rise in temperature is 29.96 × 0.000101 = 0.003026 in. The increase

TABLE XXXIV.—PROPERTIES OF SATURATED STEAM FROM 32 TO 600 F

Temperature, degrees Fahrenheit t	Vacuum in inches of mercury referred to 30-in. barometer (Mercury at 58.4° F.)	Absolute pressure, pounds per square inch p	Absolute pressure, inches of Hg. at 32° F. —	Specific volume, cubic feet per pound v or V	Density, pounds per cubic foot $1/v$	Heat of the liquid, B.t.u. per pound h or q	Latent heat of evaporation, B.t.u. per pound L or r	Total heat of steam, B.t.u. per pound H
32	29.819	0.0886	0.1804	3,294	0.000304	0.00	1073.4	1073.4
33	29.812	0.0922	0.1878	3,170	0.000316	1.01	1072.8	1073.8
34	29.804	0.0960	0.1955	3,052	0.000328	2.01	1072.2	1074.2
35	29.796	0.0999	0.2034	2,938	0.000340	3.02	1071.7	1074.7
36	29.788	0.1040	0.2117	2,829	0.000353	4.03	1071.1	1075.1
37	29.779	0.1081	0.2202	2,725	0.000367	5.04	1070.6	1075.6
38	29.771	0.1125	0.2290	2,626	0.000381	6.04	1070.0	1076.0
39	29.761	0.1170	0.2382	2,530	0.000395	7.05	1069.4	1076.5
40	29.752	0.1217	0.2477	2,438	0.000410	8.05	1068.9	1076.9
41	29.742	0.1265	0.2575	2,350	0.000425	9.05	1068.3	1077.4
42	29.732	0.1315	0.2677	2,266	0.000441	10.06	1067.8	1077.8
43	29.721	0.1366	0.2782	2,185	0.000458	11.06	1067.2	1078.3
44	29.710	0.1420	0.2890	2,107	0.000475	12.06	1066.7	1078.7
45	29.699	0.1475	0.3002	2,033	0.000492	13.07	1066.1	1079.2
46	29.687	0.1532	0.3118	1,961	0.000510	14.07	1065.6	1079.6
47	29.675	0.1591	0.3238	1,892	0.000529	15.07	1065.0	1080.1
48	29.663	0.1651	0.3363	1,826	0.000548	16.07	1064.5	1080.5
49	29.650	0.1715	0.3492	1,763	0.000567	17.08	1063.9	1081.0
50	29.637	0.1780	0.3625	1,702	0.000587	18.08	1063.3	1081.4
51	29.623	0.1848	0.3762	1,643	0.000608	19.08	1062.8	1081.9
52	29.609	0.1917	0.3903	1,586	0.000630	20.08	1062.2	1082.3
53	29.594	0.1989	0.4049	1,532	0.000653	21.08	1061.7	1082.7
54	29.579	0.2063	0.4201	1,480	0.000676	22.08	1061.1	1083.2
55	29.563	0.2140	0.4357	1,430	0.000700	23.08	1060.6	1083.6
56	29.547	0.2219	0.4518	1,381	0.000724	24.08	1060.0	1084.1
57	29.530	0.2301	0.4684	1,335	0.000749	25.08	1059.5	1084.5
58	29.513	0.2385	0.4856	1,291	0.000775	26.08	1058.9	1085.0
59	29.495	0.2472	0.5034	1,249	0.000801	27.08	1058.3	1085.4
60	29.477	0.2562	0.522	1,208	0.000828	28.08	1057.8	1085.9
61	29.458	0.2654	0.541	1,168	0.000856	29.08	1057.2	1086.3

TABLE XXXIV.—(*Continued*)

Temperature, degrees Fahrenheit t	Vacuum in inches of mercury referred to 30-in. barometer (Mercury at 58.4° F.)	Absolute pressure, pounds per square inch p	Absolute pressure, inches of Hg. at 32° F. —	Specific volume, cubic feet per pound v or V	Density, pounds per cubic foot $1/v$	Heat of the liquid, B.t.u. per pound h or q	Latent heat of evaporation, B.t.u. per pound L or r	Total heat of steam, B.t.u. per pound H
62	29.439	0.2749	0.560	1,130	0.000885	30.08	1056.7	1086.8
63	29.419	0.2847	0.580	1,093	0.000915	31.07	1056.1	1087.2
64	29.398	0.2949	0.601	1,058	0.000946	32.07	1055.6	1087.6
65	29.376	0.3054	0.622	1,024	0.000977	33.07	1055.0	1088.1
66	29.354	0.3161	0.644	991	0.001009	34.07	1054.5	1088.5
67	29.331	0.3272	0.667	959	0.001043	35.07	1053.9	1089.0
68	29.308	0.3386	0.690	928	0.001077	36.07	1053.4	1089.4
69	29.284	0.3504	0.714	899	0.001112	37.06	1052.8	1089.9
70	29.259	0.3626	0.739	871	0.001148	38.06	1052.3	1090.3
71	29.234	0.3751	0.764	843	0.001186	39.06	1051.7	1090.8
72	29.208	0.3880	0.790	817	0.001224	40.05	1051.2	1091.2
73	29.181	0.4012	0.817	792	0.001263	41.05	1050.6	1091.6
74	29.153	0.4148	0.845	767	0.001304	42.05	1050.0	1092.1
75	29.125	0.4288	0.873	743	0.001346	43.05	1049.5	1092.5
76	29.095	0.4432	0.903	720	0.001389	44.04	1048.9	1093.0
77	29.065	0.4581	0.933	698	0.001433	45.04	1048.4	1093.4
78	29.034	0.4735	0.964	677	0.001477	46.04	1047.8	1093.9
79	29.002	0.4893	0.996	657	0.001523	47.04	1047.3	1094.3
80	28.968	0.505	1.029	636.8	0.001570	48.03	1046.7	1094.8
81	28.934	0.522	1.063	617.5	0.001619	49.03	1046.2	1095.2
82	28.899	0.539	1.098	598.7	0.001670	50.03	1045.6	1095.6
83	28.863	0.557	1.134	580.5	0.001723	51.02	1045.1	1096.1
84	28.826	0.575	1.171	562.9	0.001777	52.02	1044.5	1096.5
85	28.788	0.594	1.209	545.9	0.001832	53.02	1044.0	1097.0
86	28.749	0.613	1.248	529.5	0.001889	54.01	1043.4	1097.4
87	28.708	0.633	1.289	513.7	0.001947	55.01	1042.8	1097.9
88	28.666	0.654	1.331	498.4	0.002007	56.01	1042.2	1098.3
89	28.624	0.675	1.373	483.6	0.002068	57.00	1041.7	1098.7
90	28.580	0.696	1.417	469.3	0.002131	58.00	1041.2	1099.2
91	28.535	0.718	1.462	455.5	0.002195	59.00	1040.6	1099.6
92	28.489	0.741	1.508	442.2	0.002261	60.00	1040.0	1100.1
93	28.441	0.765	1.556	429.4	0.002329	60.99	1039.5	1100.5
94	28.392	0.789	1.605	417.0	0.002398	61.99	1039.0	1101.0
95	28.341	0.813	1.655	405.0	0.002469	62.99	1038.4	1101.4
96	28.290	0.838	1.706	393.4	0.002542	63.98	1037.8	1101.8
97	28.237	0.864	1.759	382.2	0.002617	64.98	1037.3	1102.3
98	28.183	0.891	1.813	371.4	0.002693	65.98	1036.7	1102.8
99	28.127	0.918	1.869	360.9	0.002771	66.97	1036.2	1103.2
100	28.070	0.946	1.926	350.8	0.002851	67.97	1035.6	1103.6
101	28.011	0.975	1.985	341.0	0.002933	68.97	1035.1	1104.0
102	27.951	1.005	2.045	331.5	0.003017	69.96	1034.5	1104.5
103	27.889	1.035	2.107	322.2	0.003104	70.96	1034.0	1104.9
104	27.825	1.066	2.171	313.3	0.003192	71.96	1033.4	1105.3
105	27.759	1.098	2.236	304.7	0.003282	72.95	1032.8	1105.8
106	27.692	1.131	2.303	296.4	0.003374	73.95	1032.3	1106.2

TABLE XXXIV.—(*Continued*)

Temperature, degrees Fahrenheit t	Vacuum in inches of mercury referred to 30-in. barometer (Mercury at 58.4° F.)	Absolute pressure, pounds per square inch p	Absolute pressure, inches of Hg. at 32° F. —	Specific volume, cubic feet per pound v or V	Density, pounds per cubic foot $1/v$	Heat of the liquid, B.t.u. per pound h or q	Latent heat of evaporation, B.t.u. per pound L or r	Total heat of steam, B.t.u. per pound H
107	27.623	1.165	2.372	288.3	0.003469	74.95	1031.7	1106.7
108	27.550	1.199	2.443	280.5	0.003565	75.95	1031.2	1107.1
109	27.478	1.235	2.515	272.9	0.003664	76.94	1030.6	1107.5
110	27.404	1.271	2.589	265.5	0.003766	77.94	1030.0	1108.0
111	27.328	1.308	2.665	258.3	0.003871	78.94	1029.5	1108.4
112	27.250	1.346	2.740	251.4	0.003978	79.93	1028.9	1108.8
113	27.170	1.386	2.822	244.7	0.004087	80.93	1028.4	1109.3
114	27.088	1.426	2.904	238.2	0.004198	81.93	1027.8	1109.7
115	27.005	1.467	2.987	231.9	0.004312	82.92	1027.2	1110.2
116	26.919	1.509	3.073	225.8	0.004429	83.92	1026.7	1110.6
117	26.830	1.553	3.161	219.9	0.004548	84.92	1026.1	1111.0
118	26.739	1.597	3.252	214.1	0.004671	85.92	1025.5	1111.5
119	26.647	1.642	3.344	208.5	0.004796	86.91	1025.0	1111.9
120	26.553	1.689	3.438	203.1	0.004924	87.91	1024.4	1112.3
121	26.456	1.736	3.535	197.9	0.005054	88.91	1023.9	1112.8
122	26.355	1.785	3.635	192.8	0.005187	89.91	1023.3	1113.2
123	26.253	1.835	3.737	187.9	0.005323	90.90	1022.7	1113.6
124	26.149	1.886	3.841	183.1	0.005462	91.90	1022.2	1114.1
125	26.040	1.938	3.948	178.4	0.005605	92.90	1021.6	1114.5
126	25.931	1.992	4.057	173.9	0.005751	93.90	1021.1	1115.0
127	25.820	2.047	4.168	169.6	0.005900	94.89	1020.5	1115.4
128	25.706	2.103	4.282	165.3	0.006052	95.89	1019.9	1115.8
129	25.589	2.160	4.399	161.1	0.006207	96.89	1019.4	1116.2
130	25.48	2.219	4.52	157.1	0.00637	97.89	1018.8	1116.7
131	25.35	2.279	4.64	153.2	0.00653	98.89	1018.2	1117.1
132	25.23	2.340	4.76	149.4	0.00669	99.88	1017.7	1117.5
133	25.10	2.403	4.89	145.8	0.00686	100.88	1017.1	1118.0
134	24.67	2.467	5.02	142.2	0.00703	101.88	1016.5	1118.4
135	24.84	2.533	5.16	138.7	0.00721	102.88	1016.0	1118.8
136	24.70	2.600	5.29	135.4	0.00739	103.88	1015.4	1119.3
137	24.56	2.669	5.43	132.1	0.00757	104.87	1014.8	1119.7
138	24.41	2.740	5.58	128.9	0.00776	105.87	1014.3	1120.1
139	24.26	2.812	5.73	125.8	0.00795	106.87	1013.7	1120.6
140	24.11	2.885	5.88	122.8	0.00814	107.87	1013.1	1121.0
141	23.96	2.960	6.03	119.9	0.00834	108.87	1012.6	1121.4
142	23.81	3.037	6.18	117.1	0.00854	109.87	1012.0	1121.8
143	23.64	3.115	6.34	114.3	0.00875	110.87	1011.4	1122.3
144	23.47	3.195	6.51	111.6	0.00896	111.87	1010.8	1122.7
145	23.31	3.277	6.67	109.0	0.00918	112.86	1010.3	1123.1
146	23.14	3.361	6.84	106.5	0.00940	113.86	1009.7	1123.6

TABLE XXXIV.—(*Continued*)

Temperature, degrees Fahrenheit	Vacuum in inches of mercury referred to 30-in. barometer (Mercury at 58.4° F.)	Absolute pressure, pounds per square inch p	Absolute pressure, inches of Hg. at 32° F. —	Specific volume, cubic feet per pound v or V	Density, pounds per cubic foot $1/v$	Heat of the liquid, B.t.u. per pound h or q	Latent heat of evaporation, B.t.u. per pound L or r	Total heat of steam, B.t.u. per pound H
147	22.96	3.446	7.02	104.0	0.00962	114.86	1009.1	1124.0
148	22.78	3.533	7.20	101.6	0.00985	115.86	1008.6	1124.4
149	22.60	3.623	7.38	99.2	0.01008	116.86	1008.0	1124.8
150	22.42	3.714	7.57	96.9	0.01032	117.86	1007.4	1125.3
151	22.22	3.809	7.76	94.7	0.01056	118.86	1006.8	1125.7
152	22.03	3.902	7.95	92.6	0.01080	119.86	1006.2	1126.1
153	21.84	3.999	8.14	90.5	0.01105	120.86	1005.7	1126.5
154	21.64	4.098	8.34	88.4	0.01131	121.86	1005.1	1127.0
155	21.43	4.199	8.55	86.4	0.01157	122.86	1004.5	1127.4
156	21.22	4.303	8.76	84.5	0.01184	123.86	1003.9	1127.8
157	21.00	4.408	8.98	82.6	0.01211	124.86	1003.4	1128.2
158	20.78	4.515	9.20	80.7	0.01239	125.86	1002.8	1128.6
159	20.56	4.625	9.42	78.9	0.01267	126.86	1002.2	1129.1
160	20.32	4.737	9.65	77.2	0.01296	127.86	1001.6	1129.5
161	20.09	4.851	9.88	75.5	0.01325	128.86	1001.1	1129.9
162	19.85	4.967	10.12	73.8	0.01355	129.86	1000.5	1130.4
163	19.61	5.086	10.36	72.2	0.01386	130.86	999.9	1130.8
164	19.36	5.208	10.61	70.6	0.01417	131.86	999.3	1131.2
165	19.11	5.333	10.86	69.1	0.01448	132.86	998.7	1131.6
166	18.85	5.460	11.12	67.6	0.01480	133.86	998.2	1132.0
167	18.59	5.589	11.38	66.1	0.01513	134.86	997.6	1132.4
168	18.32	5.721	11.65	64.7	0.01546	135.86	997.0	1132.8
169	18.05	5.855	11.92	63.3	0.01580	136.86	996.4	1133.3
170	17.77	5.992	12.20	62.0	0.01614	137.87	995.8	1133.7
171	17.49	6.131	12.48	60.7	0.01649	138.87	995.2	1134.1
172	17.20	6.273	12.77	59.4	0.01685	139.87	994.6	1134.5
173	16.90	6.417	13.07	58.1	0.01721	140.87	994.0	1134.9
174	16.60	6.564	13.37	56.9	0.01758	141.87	993.5	1135.3
175	16.30	6.714	13.67	55.7	0.01796	142.87	992.9	1135.7
176	15.99	6.867	13.98	54.5	0.01834	143.87	992.3	1136.2
177	15.67	7.023	14.30	53.4	0.01873	144.88	991.7	1136.6
178	15.35	7.182	14.62	52.3	0.01912	145.88	991.1	1137.0
179	15.02	7.344	14.95	51.2	0.01953	146.88	990.5	1137.4
180	14.67	7.51	15.29	50.15	0.01994	147.88	989.9	1137.8
181	14.33	7.68	15.63	49.12	0.02036	148.88	989.3	1138.2
182	13.98	7.85	15.98	48.12	0.02078	149.89	988.7	1138.6
183	13.62	8.02	16.34	47.14	0.02121	150.89	988.1	1139.0
184	13.26	8.20	16.70	46.18	0.02165	151.89	987.5	1139.4
185	12.89	8.38	17.07	45.25	0.02210	152.89	986.9	1139.8
186	12.51	8.57	17.45	44.34	0.02255	153.89	986.3	1140.2

TABLE XXXIV.—(*Continued*)

Temperature, degrees Fahrenheit t	Vacuum in inches of mercury referred to 30-in. barometer (Mercury at 58.4 F.)	Absolute pressure, pounds per square inch p	Absolute pressure inches, of Hg. at 32 F. —	Specific volume, cubic feet per pound v or V	Density, pounds per cubic foot $1/v$	Heat of the liquid, B.t.u. per pound h or q	Latent heat of evaporation, B.t.u. per pound L or r	Total heat of steam, B.t.u. per pound H
187	12.13	8.76	17.83	43.45	0.02301	154.90	985.7	1140.6
188	11.74	8.95	18.22	42.59	0.02348	155.90	985.1	1141.0
189	11.35	9.14	18.61	41.74	0.02396	156.90	984.5	1141.4
190	10.93	9.34	19.02	40.91	0.02444	157.91	983.9	1141.8
191	10.52	9.54	19.43	40.10	0.02493	158.91	983.3	1142.2
192	10.10	9.74	19.85	39.31	0.02544	159.91	982.7	1142.6
193	9.68	9.95	20.27	38.54	0.02595	160.91	982.1	1143.0
194	9.24	10.17	20.71	37.78	0.02647	161.92	981.5	1143.4
195	8.80	10.39	21.15	37.04	0.02700	162.92	980.9	1143.8
196	8.35	10.61	21.60	36.32	0.02753	163.92	980.3	1144.2
197	7.90	10.83	22.05	35.62	0.02807	164.93	979.7	1144.6
198	7.43	11.06	22.52	34.93	0.02863	165.93	979.1	1145.0
199	6.96	11.29	22.99	34.26	0.02919	166.94	978.4	1145.4
200	6.47	11.52	23.47	33.60	0.02976	167.94	977.8	1145.8
201	5.99	11.76	23.95	32.96	0.03034	168.94	977.2	1146.2
202	5.49	12.01	24.45	32.33	0.03093	169.95	976.6	1146.6
203	4.98	12.26	24.96	31.72	0.03153	170.95	976.0	1146.9
204	4.46	12.51	25.48	31.12	0.03214	171.96	975.4	1147.3
205	3.93	12.77	26.00	30.53	0.03276	172.96	974.7	1147.7
206	3.40	13.03	26.53	29.95	0.03339	173.97	974.1	1148.1
207	2.85	13.30	27.08	29.39	0.03402	174.97	973.5	1148.5
208	2.30	13.57	27.63	28.85	0.03466	175.98	972.9	1148.9
209	1.74	13.85	28.19	28.32	0.03531	176.98	972.2	1149.2
210	1.16	14.13	28.76	27.80	0.03597	177.99	971.6	1149.6
211	0.59	14.41	29.33	27.29	0.03664	178.99	971.0	1150.0
212	0.00	14.70	29.92	26.79	0.03732	180.00	970.4	1150.4
220		17.19		23.15	0.04320	188.1	965.2	1153.3
230		20.77		19.39	0.0516	198.2	958.7	1156.9
240		24.97		16.32	0.0613	208.3	952.1	1160.4
250		29.82		13.82	0.0724	218.5	945.3	1163.8
260		35.42		11.76	0.0850	228.6	938.4	1167.0
270		41.85		10.06	0.0994	238.8	931.4	1170.2
280		49.18		8.64	0.1157	249.0	924.3	1173.3
290		57.55		7.46	0.1341	259.3	916.9	1176.2
300		67.00		6.46	0.1547	269.6	909.5	1179.1
310		77.67		5.62	0.1779	279.9	901.9	1181.8
320		89.63		4.91	0.2036	290.2	894.2	1184.4
330		103.0		4.306	0.2322	300.6	886.3	1186.9
340		118.0		3.787	0.2641	311.0	878.3	1189.3
350		134.6		3.342	0.2992	321.4	870.1	1191.5
360		153.0		2.957	0.3382	331.9	861.8	1193.7
370		173.3		2.627	0.3806	342.4	853.4	1195.8
380		195.6		2.340	0.427	352.9	844.8	1197.7
390		220.2		2.089	0.479	363.5	836.1	1199.6

TABLE XXXIV.—(*Concluded*)

Temperature, degrees Fahrenheit t	Absolute pressure, pounds per square inch p	Specific volume, cubic feet per pound v or V	Density, pounds per cubic foot $1/v$	Heat of the liquid, B.t.u. per pound h or q	Latent heat of evaporation, B.t.u. per pound L or r	Total heat of steam, B.t.u. per pound H
400	247.1	1.872	0.534	374.1	827.2	1201.3
410	276.4	1.679	0.596	384.7	818.2	1202.9
420	308.4	1.510	0.662	395.4	809.0	1204.4
430	343.2	1.361	0.735	406.2	799.6	1205.8
440	380.8	1.229	0.814	417	790.1	1207.1
450	421	1.11	0.90	428	780	1208
460	465	1.00	1.00	439	770	1209
470	513	0.90	1.11	451	759	1210
480	565	0.81	1.23	462	748	1210
490	622	0.73	1.37	473	737	1210
500	684	0.66	1.52	484	725	1209
510	751	0.60	1.67	496	712	1208
520	822	0.55	1.83	507	700	1207
530	897	0.50	2.00	519	686	1205
540	977	0.46	2.17	531	672	1203
550	1062	0.42	2.36	542	658	1200
560	1152	0.39	2.57	554	642	1196
570	1247	0.35	2.82	566	626	1192
580	1349	0.32	3.12	578	609	1187
590	1458	0.29	3.42	591	591	1182
600	1574	0.27	3.75	604	572	1176

from 29.92 is 0.08 in., which divided by 0.003026 equals 26.4 F. Adding this to 32 F, the temperature of the 29.92-in. barometer, 32 + 26.4 = 58.4 F is the temperature of a 30-in. barometer.

More complete data will be found in the steam tables of Keenan and Keyes in their book "Thermodynamic Properties of Steam Including Data for the Liquid and Solid Phases."[1] Although the values given in Table XXXV differ somewhat from those given in the latest tables of Keenan and Keyes, consistent use of these tables will give reasonably accurate results. In any case it is advisable to use the same reference in a given set of calculations to avoid minor discrepancies. For a comparison of the Keenan and Keyes steam tables with other tables formulated since 1910, see the review by R. C. H. Heck in *Mechanical Engineering*, February, 1937.

In Table XXXV are given, in condensed form, the properties of *saturated* and *superheated* steam from 1 to 1,200 lb abs. This table was made up from data in the paper entitled "Progress Report on the Development of Steam Charts and Tables from the Harvard Throttling Experiments," presented by J. H. Keenan

[1] John Wiley & Sons, Inc., New York, 1936.

TABLE XXXV.—PROPERTIES OF SATURATED AND SUPERHEATED STEAM

Data below 320 F from Marks and Davis's "Steam Tables," Longmans, Green and Co.

Data above 320 F from Davis-Kleinschmidt, Joule-Thompson tests as computed by J. H. Keenan from data obtained through courtesy of the ASME Committee on Properties of Steam and the Extension of the Steam Tables. (See *Mechanical Engineering*—February, 1926.)[1]

Absolute pressure, pounds per square inch		Liquid	Superheat, degrees Fahrenheit 0	25	50	75	100	125	150	175	200	250	300
1	t	101.8	101.8	126.8	151.8	176.8	201.8	226.8	251.8	276.8	301.8	351.8	401.8
	v	.....	333.0	348.6	363.9	378.9	393.9	408.8	423.8	438.7	453.	483.5	513.4
	h	69.8	1104.4	1115.7	1127.1	1138.5	1149.9	1161.3	1172.8	1184.2	1195 6	1218.4	1241.2
	n	0.1327	1.9754	1.9954	2.0144	2.0327	2.0503	2.0671	2.0835	2.0993	2.1145	2.143	2.166
5	t	162.3	162.3	187.3	212.3	237.3	262.3	287.3	312.3	337.3	362.3	412	462.3
	v	.....	73.3	76.5	79.7	82.6	85.7	.8	91.8	94.8	97.8	103.8	109.8
	h	130.2	1130.5	1142.0	1153.5	1164.9	1176.4	1187.9	1199.5	1211.0	1222	1245	1268.5
	n	0.2348	1.8433	1.8618	1.8794	1.8960	1.9125	1.9282	1.9428	1.957	1.972	2.000	2.028
10	t	193.2	193.2	218.2	243.2	268.2	293.2	318.2	343.2	368.2	393.	443.2	493.2
			38.38	40.0	41.6	43.1	44.6	46.1	47.6	49.1	50.6	53.6	56.6
		161.1	1143.1	1154.7	1166.3	1177.9	1189.5	1201.1	1212.7	1224.5	1236	1259.7	1283.1
	n	0.2832	1.7873	1.8050	1.8220	1.8371	1.8538	1.8689	1.884		1.912	.939	1.965
15	t	213.0	213.0	238.0	263.0	288.0	313.0	338.0	363.0	388.0	413.0	463.0	513.0
	v	.....	26.29	27.23	28.35	29.38	30.40	31.43	32.44	33.45	34.45	36.45	38.46
	h	180.9	1150.7	1162.5	1174.2	1185.9	1197.6	1209.4	1221.1	1232.9	1244.7	1268.2	1291.8
	n	0.3134	1.7547	1.7720	1.7886	1.8044	1.8199	1.835	1.850	1.864	1.878	1.905	1.930
20	t	228.0	228.0	253.0	278.0	303.0	328.0	353.0	378.0	403.0	428.0	478.0	528.0
		.....	20.09	20.88	21.66	22.44	23.21	23.98	24.75	25.50	26.26	27.78	29.23
		196.2	1156.2	1168.1	1179.9	1191.7	1203.6	1215.4	1227.3	1239.2	1251.1	1274.9	1298.6
		0.3355	1.7320	1.7489	1.7652	1.7810	1.796	1.811	1.826	1.840	1.854	1.880	1.905
25	t	240.1	240.1	265.1	290.1	315.1	340.1	365.1	390.1	415.1	440.1	490.1	540.1
	v		16.30	16.94	17.57	18.19	18.82	19.44	20.06	20.66	21.28	22.49	23.70
	h	208.3	1160.4	1172.4	1184.4	1196.3	1208.3	1220.3	1232.2	1244.2	256.3	1280.3	304.2
	n	0.3532	1.7136	1.7304	1.7466	1.7622	1.778	1.793	1.808	1.812	1.836	1.862	1.886
30	t	250.3	250.3	275.3	300.3	325.3	350.3	375.3	400.3	425.3	450.3	500.3	550.3
		......	13.74	14.28	14.81	15.33	15.85	16.37	16.88	17.39	17.90	18.93	19.93
		218.7	1163.9	1176.0	1188.1	1200.2	1212.2	1224.3	1236.4	1248.5	1260.6	1284.8	308.9
		0.3680	1.6991	1.7158	1.7319	1.748	1.763	1.778	1.794	1.806	1.820	1.846	1.871
35	t	259.3	259.3	284.3	309.3	334.3	339.3	384.3	409.3	434.3	459.3	509.3	559.3
	v	......	11.89	12.36	12.82	13.27	13.72	14.17	14.61	15.05	15.46	16.36	17.23
	h	227.8	1166.8	1179.1	1191.4	1203.7	1216.0	1228.2	1240.4	1252.5	1264.6	288.8	312.9
	n	0.3807	1.6868	1.7035	1.7196	1.735	1.750	1.766	1.780	1.794	1.806	1.832	1.857
40	t	267.3	267.3	292.3	317.3	342.3	367.3	392.3	417.3	442.3	467.3	517.3	567.3
	v	.....	10.49	10.90	11.30	11.70	12.10	12.50	12.89	13.29	13.66	14.42	15.18
	h	236.0	1169.4	1182.1	1194.6	1207.0	1219.3	1231.5	1243.7	1255.9	1268.0	292.2	316.4
	n	0.3919	1.6761	1.694	1.710	1.725	1.740	1.755	1.769	1.783	1.797	1.822	1.846

[1] Material as presented here has been rearranged and total heats as read from chart adjusted where necessary.

t = temperature in degrees Fahrenheit. h = total heat in Btu per pound.

v = specific volume in cubic feet per pound. n = entropy.

TABLE XXXV.—(*Continued*)

Absolute pressure,

Liquid	Superheat, degrees Fahrenheit 0	25	50	75	100	125	150	175	200	250	300
274.5	274.5	299.5	324.5	349.	374.5	399.5	424.5	449.5	474.5	524.5	574.5
....	9.39	9.77	10.13	10.48	10.84	11.19	11.54	11.88	12.21	12.91	13.57
243.2	1171 6	1184 3	1196 8	1209.2	1221.6	1234.0	1246.3	1258.6	1270.8	1295	1319.6
0.4019	1.666	1.684	1.700	1.716	1.731	1.746	1.760	1.773	1.786	1	1.836
281.0	281.0	306	331	356	381.0	406	431	456	481	531	581
....	8.51	8.83	9.16	9.49	9.80	10.12	10.44	10.75	11.06	11.67	12.29
249.9	1173.6	1186.9	1199.7	1212.3	1224.8	1237.2	1249	1261.7	1273.9	1298.2	1322.5
0.4111	1.658	1.675	1.692	1.708	.723	1.737	.751	1.765	1.778	1.803	1.827
287.1	287.1	312.1	337.1	362.1	387.1	412.1	437.1	462.1	487.1	537.1	587.1
....	7.78	8.08	8.38	8.68	8.98	9.27	9.54	9.83	10.11	10.68	11.24
256.1	1175.4	1189.0	1202.1	1214.8	1227.3	1239.7	1252.0	1264.3	1276.6	1300	1325.1
0.4194	1.650	1.668	1.685	1.700	1.715	1.729	1.743	1.757	1.770	1.796	1.820
292.7	292.7	317.7	342.7	367.7	392.7	417.7	442.7	467.7	492.7	542.7	592.7
....	7.16	7.44	7.73	8.01	8.27	8.54		9.07	9.33	9.84	10.36
261.9	1177.0	1190.5	1203.6	1216.4	1229.0	1241.5	1253.9	1266.2	1278.5	1302.8	1327.2
0.4269	1.643	1.661	1.678	1.693	1.708	1.722	1.736	1.750	1.763	1.788	1.812
298.0	298.0	323	348	373	398	423	448	473	498	548	598
....	6.64	6.90	7.17	7.42	7.67	7.92	8.17	8.41	8.65	9.12	9.59
267.3	1178.5	1192.0	1205.2	1218.1	1230.8	1243.3	1255.8	1268.2	1280.6	305.1	1329.5
0.4341	1.637	1.654	1.671	1.686	1.701	1.716	1.730	1.743	1.756	1.782	1.805
302.9	302.9	327.9	352.9	377.9	402.9	427.9	452.9	477.9	502.9	552.9	602.9
....	6.20	6.44	6.69	6.92	7.16	7.39	7.62	7.85	8.07	8.52	8.94
272.4	1179.8	1193.7	1207.1	1220.0	1232.7	1245.3	1257.8	1270.2	1282.6	307.2	1331.6
0.4405	1.631	1.648	1.665	1.680	1.695	1.709	1.723	1.737	1.750	1.776	1.800
307.6	307.6	332.6	357.6	382.6	407.6	432.6	457.6	482.6	507.6	557.6	607.6
....	5.80	6.04	6.27	6.49	6.71	6.93	7.15	7.36	7.57	7.98	8.40
277.2	1181.	1194.9	1208.3	1221.3	1234.2	1246.9	1259.5	1272.0	1284.4	309.1	1333.6
0.4470	1.625	1.643	1.660	1.675	1.690	1.704	1.718	1.732	1.745	1.770	1.794
312.0	312.0	337	362	387	412	437	462	487	512	562	612
....	5.46	5.69	5.90	6.11	6.32	6.52	6.72	6.92	7.11	7.50	7.89
281.8	1182.3	1196.4	1209.9	1223.0	1235.9	1248.6	1261.1	1273.6	1286.1	310.9	1335.5
0.4631	1.620	1.637	1.655	1.670	1.685	1.699	1.713	1.727	1.740	1.765	1.789
316.3	316.3	341.3	366.3	391.3	416.3	441.3	466.3	491.3	516.3	566.3	616.3
....	5.16	5.36	5.57	5.77		6.16	6.35	6.54	6.72	7.10	7.46
286.2	1183.4	1197.7	1211.3	1224.5	1237.4	1250.1	262.7	1275.3	1287.8	312.6	1337.3
0.4586	1.615	1.632	1.650	1.665	1.680	1.694	1.708	1.722	1.735	1.760	1.784
320.3	320.3	345.3	370.3	395.3	420.3	445.3	470.3	495.3	520.3	570.3	620.3
....	4.89	5.09	5.28	5.48	5.66	5.83	6.01	6.19	6.36	6.72	7.07
290.3	1184.4	1198.8	1212.5	1225.8	1238.7	1251.4	1264.1	1276.7	1289.2	1314.2	1339.0
0.4640	1.610	1.628	1.645	1.661	1.676	1.690	1.704	1.717	1.730	1.756	1.780
324.1	324.1	349.1	374.1	399.1	424.1	449.1	474.1	499.1	524.1	574.1	624.1
....	4.649	4.84	5.02	5.20	5.37	5.55	5.72	5.89	6.06	6.40	6.72
294.3	1185.4	1200.0	1213.8	1227.1	1240.1	1252.9	1265.6	1278.2	1290.7	1315	1340.7
0.4690	1.606	1.624	1.640	1.656	1.672	1.686	1.700	1.713	1.726	1.750	1.774
327.8	327.8	352.8	377.8	402.8	427.8	452.8	477.8	502.8	527.8	577.8	627.8
....	4.429	4.61	4.78	4.95	5.11	5.28	5.44	5.61	5.77	6.08	6.38
298.1	1186.3	1201.1	1215.1	1228.5	1241.5	1254.3	1267.0	1279.5	1292.1	1317.1	1342.1
0.4739	1.602	1.620	1.637	1.652	1.667	1.682	1.696	1.709	1.722	1.747	1.770

TABLE XXXV.—(*Continued*)

Absolute pressure, pounds per inch p		Liquid	Superheat, degrees Fahrenheit 0	25	50	75	100	125	150	175	200	250	300
105	t	331.4	331.4	356.4	381.4	406.	431.	456.4	481.4	506.	531.	581.	631.4
		.....	4.227	4.401	4.565	4.730	4.893	5.049	5.203	5.354	5.503	5.8	6.11
		301.8	1187.2	1202 1	1216.3	1229 8	1242.9	1255.8	1268.5	1281.1	1293.6	1318.6	1343.[illegible]
		0.4785	1.598	.616	.633	1.649	1.664	1.678	1.692	1.705	1.718	1.743	1.767
110	t	334.8	334.8	359.8	384.8	409.8	434.	459.8	484.8	509.8	534.8	584.8	634.8
	v	.....	4.045	4.208	4.370	4.525	4.684	4.833	4.978	5.128	5.27	5.56	5.85
	h	305.3	1188.0	1203.0	1217.2	1230.8	1244.0	1256.9	1269.7	1282.	1295.	320.0	1345.0
	n	0.4829	1.594	1.612	1.629	1.645	1.660	1.674	1.688	1.701	1.714	1.739	1.763
115	t	338.1	338.1	363.1	388.1	413.1	438.1	463.1	488.1	513.1	538.	588.	638.1
	v	.....	3.877	4.036	4.186	4.340	4.495	4.634	4.777	4.915	5.05	5.333	5.61
	h	308.8	1188.8	1203.9	1218.1	1231.8	1245.0	1258.0	1270.8	[illegible]	1296	1321.3	1346.3
	n	0.4872	1.590	1.608	1.626	1.642	1.657	1.671	1.685	1.698	1.71	1.736	1.760
120	t	341.3	341.3	366.3	391.3	416.3	441.3	466.3	491.3	516.3	541.	591.3	641.3
	v	.....	3.724	3.877	4.026	4.176	4.315	4.455	4.590	4.725	4.85	5.120	5.38
	h	312 1	1189.5	1204.6	1218.9	1232	1245.7	1258.8	1271.7	1284	1297	1322	1347.6
	n	0.4914	1.587	1.605	1.622	1.638	1.653	1.667	1.681	1.694	1.70	1.732	1.756
125	t	344.4	344.4	369.4	394.4	419.4	444.4	469.4	494.	519.4	544.	594.	644.4
	v	.....	3.582	3.735	3.875	4.015	4.152	4.282	4.41	4.54	4.673	4.930	5.180
	h	315.3	1190.2	1205.3	1219.7	1233.5	1246.9	1260.0	1272.9	1285.7	1298	1323.7	1348.9
	n	0.4953	1.583	1.602	1.619	1.635	1.650	1.665	1.679	1.692	1.705	1.729	1.753
130	t	347.4	347.4	372.4	397.	422.4	447.	472.	497.4	522.4	547.4	597.4	647.4
		.....	3.451	3.597	3.733	3.866	3.995	4.125	4.250	4.385	4.509	4.745	5.00
		318.4	1190.8	1206.1	1220.5	1234.4	1247.9	1261.0	1274.0	1286.9	1299.6	1324.9	1350.1
		0.4992	1.580	1.598	1.616	1.632	1.647	1.661	1.675	1.688	1.701	1.726	1.749
135	t	350.3	350.3	375.3	400.3	425.3	450.3	475.3	500.3	525.3	550.3	600.3	650.3
		.....	3.330	3.467	3.602	3.736	3.861	3.981	4.106	4.230	4.350	4.579	4.82
		321.4	1191.5	1206.8	221.3	1235.3	1248.7	1261.8	274.8	1287.7	1300.4	325.8	1351.2
		0.5029	1.577	1.596	1.613	1.629	1.644	1.659	1.672	1.685	1.698	1.723	1.746
140	t	353.1	353.1	378.1	403.1	428.1	453.1	478.1	503.1	528.1	553.1	603.1	653.1
	v	.....	3.216	3.351	3.486	3.611	3.731	3.851	3.971	4.083	4.205	4.431	4.65
	h	324.4	1192.1	1207.5	222.1	1236.2	1249.7	1263.0	276.0	1288.8	301.5	326.9	1352.3
	n	0.5065	1.574	1.593	1.610	1.626	1.641	1.655	1.669	1.682	1.695	1.719	1.743
145	t	355.8	355.8	380.8	405.8	430.8	455.8	480.8	505.8	530.8	555.8	605.8	655.8
	v	.....	3.111	3.242	3.369	3.491	3.605	3.724	3.838	3.950	4.069	4.284	4.510
	h	327.1	1192.6	1208.2	1222.9	1236.9	1250.4	1263.6	1276.7	1289.6	302.5	328.0	1353.4
	n	0.5100	1.571	1.590	1.607	1.623	1.638	1.653	1.667	1.680	1.693	1.717	1.740
150	t	358.5	358.5	383.5	408.5	433.5	458.5	483.5	508.5	533.5	558.5	608.5	658.5
	v	.....	3.011	3.138	3.263	3.382	3.493	3.608	3.714	3.824	3.936	4.144	4.370
	h	329.9	1193.2	1208.9	1223.7	1237.8	1251.5	1264.8	277.9	290.	303.6	329.1	354.5
	n	0.5135	1.569	1.587	1.604	1.620	1.636	1.650	1.664	1.677	1.690	1.714	1.738
160	t	363.6	363.6	388.6	413.6	438.6	463.6	.6	513.6	538.6	563.6	613.6	663.6
	v	.....	2.833	2.954	3.069	3.180	3.281	3.389	3.497	3.596	3.706	3.90	4.10
	h	335.3	1194.3	1210.0	224.8	1239.1	1253.0	1266.2	279.4	292.4	305.3	330.9	356.5
	n	0.5200	1.564	1.582	1.600	1.616	1.631	1.645	1.659	1.672	1.685	1.709	1.732
170	t	368.5	368.5	393.5	418.5	443.5	468.5	493.5	518.5	543.5	568.5	618.5	668.5
		.....	2.674	2.785	2.899	3.000	3.102	3.199	3.308	3.407	3.507	3.692	3.880
		340.4	1195.3	1211.2	226.2	1240.6	1254.4	1267.9	281.1	294.1	307.1	332.8	358.4
	n	0.5261	1.558	1.577	1.595	1.611	1.626	1.640	1.654	1.667	1.680	1.704	1.728

TABLE XXXV.—(*Continued*)

Absolute pressure, pounds per square inch		Liquid	Superheat, degrees Fahrenheit 0	25	50	75	100	125	150	175	200	250	300
180	*t*	373.1	373.	398.1	423.1	448.1	473.1	498.1	523.1	548.1	573.	623.1	673.2
	v	.....	2.531	2.636	2.744	2.841	2.940	3.038	3.126	3.226	3.313	3.492	3.63
	h	345.3	1196.1	212.0	1227.1	241.6	1255.7	1269.2	1282.5	1295.7	1308.8	1334.6	1360
	n	0.5319	1.554	1.572	1.590	1.606	1.621	1.635	1.649	1.663	1.676	1.700	1.727
190	*t*	377.6	377.6	402.6	427.6	452.6	477.6	502.6	527.6	552.6	577.6	627.6	677.9
	v	.....	2.403	2.503	2.603	2.701	2.792	2.887	2.975	3.065	3.150	3.324	3.49
	h	350.0	1196.9	1213.0	1228.2	1242.7	1256.9	270.6	1284.0	1297.3	1310.4	1336.3	1361.9
	n	0.5374	1.549	1.568	1.586	1.602	1.617	1.631	1.645	1.658	1.672	1.696	1.719
200	*t*	381.9	381.9	406.9	431.9	456.9	481.9	506.9	531.9	556.9	581.9	631.9	681.9
	v	.....	2.287	2.380	2.480	2.574	2.667	2.748	2.833	2.922	2.999	3.163	3.32
	h	354.5	1197.6	214.0	1229.4	1244.0	1258.1	1271.9	1285.4	1298.7	1311.9	1337.8	1363.6
	n	0.5426	1.545	1.564	1.581	1.598	1.612	1.626	1.640	1.654	1.667	1.691	1.714
210	*t*	386.0	386.0	411.0	436.0	461.0	486.0	511.0	536.0	561.0	586.0	636.0	686.0
	v		2.182	2.278	2.367	2.458	2.544	2.626	2.706	2.787	2.867	3.025	3.18
	h	358.8	1198	214.8	1230.4	1245.1	1259.2	1273.0	1286.6	1300.0	1313.3	1339.4	365.2
	n	0.5477	1.541	1.559	1.577	1.594	1.608	1.623	1.637	1.651	1.663	1.687	1.710
220	*t*	390.0	390.0	415.0	440.0	465.0	490.0	515.0	540.0	565.0	590.0	640.0	690.0
	v		2.086	2.179	2.263	2.347	2.430	2.513	2.587	2.669	2.744	2.890	3.040
	h	363.0	1198.8	215.5	1231.2	1246.0	1260.3	1274.2	1287.9	1301.4	1314.7	340.8	366.7
		0.5526	1.537	1.555	1.573	1.590	1.604	1.619	1.633	1.646	1.659	1.683	1.706
230	*t*	393.8	393.8	418.8	443.8	468.8	493.8	518.8	543.8	568.8	593.8	643.8	693.8
	v		1.998	2.087	2.169	2.249	2.332	2.405	2.487	2.561	2.633	2.773	2.910
	h	367.1	1199.4	1216.2	1231.9	1246.9	1261.3	1275.4	1289.1	1302.6	1 16.0	342.3	368.2
	n	0.557	1.533	1.552	1.570	1.586	1.601	1.616	.630	1.644	.656	1.680	1.703
240	*t*	397.5	397.5	422.5	447.5	472.5	497.5	522.5	547.5	572.5	597.5	647.5	697.5
	v	.....	.917	2.004	2.087	2.160	2.243	2.314	2.388	2.455	2.525	2.665	2.800
	h	371.0	1199.9	1216.8	1232.	1247.8	1262.3	1276.6	1290.4	1303.9	1317.2	1343.5	369.6
	n	0.5619	1.529	1.549	1.567	1.583	1.598	1.613	1.627	1.640	1.653	1.677	1.700
250	*t*	401.1	401.1	426.1	451.1	476.1	501.1	526.1	551.1	576.1	601.1	651.1	701.1
	v		1.843	1.925	2.003	2.0	2.154	2.224	2.294	2.363	2.430	2.562	2.690
	h	374.9	1200.	1217.3	1233.3	1248.6	1263.2	1277.5	1291.4	1305.0	1318	1344.9	371.0
	n	0.5663	1.525	1.546	1.563	1.580	1.595	1.610	1.624	1.636	1.650	1.674	1.697
260	*t*	404.5	404.5	429.5	454.5	479.5	504.5	529.5	554.5	579.5	604.5	654.5	704.5
	v		1.776	.853	.932	2.004	2.076	2.144	2.202	2.278	2.342	2.469	2.592
	h	378.6	1200.8	1218.0	1234.1	1249.	1264.1	1278.4	1292.4	1306.0	1319.5	1346.2	372.3
	n	0.5706	1.522	1.543	1.561	1.577	1.592	1.606	.620	1.634	1.647	1.672	1.694
270	*t*	407.9	407.	432.9	457.9	482.9	507.9	532.9	557.9	582.9	607.9	657.9	607.9
	v	....	1.71	1.788	1.863	1.934	2.003	2.070	2.135	2.198	2.261	2.383	2.502
	h	382.2	1201.	1218.5	1234.6	1250.1	1265.0	1279.4	1293.4	1307.1	1320.6	1347.3	1373.6
	n	0.5747	1.519	1.540	1.558	1.57		1.604	1.618	1.631	1.644	1.668	1.691
280	*t*	411.2	411.	436.2	461.2	486.	511.2	536.2	561.2	586.	611.2	661.2	711.2
	v		1.65	1.727	1.798	1.869	1.934	2.000	2.062	2.124	2.184	2.302	2.418
	h	385.7	1201.5	1219.0	1235.3	1250.8	1265.8	1280.4	1294.4	1308.1	1321.6	1348.3	1374.8
	n	0.5787	1.51	1.536	1.554	1.571	1.586	1.601	1.615	1.628	1.641	.665	1.688
290	*t*	414.4	414.4	439.4	464.	489.	514.4	539.4	564.4	589.4	614.4	664.4	714.4
			1.596	1.670	1.739	1.808	1.871	1.934	1.995	2.055	2.112	2.227	2.339
		389.1	1201.8	1219.7	1236.1	1251.6	1266.6	1281.1	1295.2	1309.0	1322.6	1349.5	1376.0
		0.5826	1.513	1.533	1.551	1.568	1.584	1.599	1.612	1.625	1.638	1.662	1.685

TABLE XXXV.—(*Continued*)

Absolute pressure, pounds per square inch		Liquid		Superheat, degrees Fahrenheit 25	50	75	100	125	150	175	200	250	300
300	*t*	417.5	417.5	442.5	467.5	492.5	517.5	542.5	567.5	592.5	617.5	667.5	717.6
	v	....	1.544	1.615	1.684	1.750	1.812	1.873	1.932	1.990	2.047	2.158	2.265
	h	392.4	202.1	1220.3	236 8	252.4	267.4	281.9	1296.0	1309.9	1323.5	1350.4	1377.
	n	0.5863	1.510	1.530	1.548	1.565	1.580	1.595	1.608	1.621	1.634	1.659	1.682
310	*t*	420.5	420.5	445.5	470.5	495.5	520.5	545.5	570.5	595.5	620.5	670.5	720.5
	v		1.495	1.564	1.632	1.696	1.756	1.815	1.873	1.929	1.985	2.093	2.197
	h	395.7	202.4	1220.5	237.2	253.0	268.1	282.7	1296.9	1310.8	1324.0	1351.4	1378.2
	n	0.5900	1.507	1.528	1.546	1.563	1.578	1.593	1.607	1.620	1.632	1.657	1.680
320	*t*	423.	423.4	448.4	473.4	498.4	523.4	548.4	573.4	598.4	623.4	673.4	723.4
	v	....	1.449	1.517	1.583	1.645	1.703	1.761	1.817	1.871	1.926	2.030	2.132
	h	398.9	1202.6	1220.8	237.8	253.6	268.8	283.4	1297.7	1311.6	1325.3	1352.4	1379.3
	n	0.5935	1.504	1.525	1.543	1.559	1.576	1.589	1.603	1.616	1.629	1.654	1.677
330	*t*	426.3	426.3	451.3	476.3	501.3	526.3	551.3	576.3	601.3	626.3	676.3	726.3
	v	....	1.406	.472	1.537	1.596	1.654	1.710	1.764	1.817	1.871	1.970	2.071
	h	402.0	1202.8	1221.2	238.2	254.2	269.5	284.2	1298.5	1312.4	1326.2	1353.4	1380.3
	n	0.5970	1.502	1.522	1.540	1.557	1.574	1.587	1.601	1.614	1.627	1.653	1.675
340	*t*	429.	429.1	454.1	479.1	504.1	529.1	554.1	579.1	604.1	629.1	679.1	729.1
	v	....	1.365	1.430	1.493	1.550	1.607	1.661	1.714	1.766	1.818	1.915	2.013
	h	405.0	1202.9	1221.6	238.7	254.6	270.1	284.9	1299.2	1313.2	1327.0	1354.3	1381.3
	n	0.6004	1.499	1.519	1.537	1.554	1.570	1.585	1.598	1.611	1.624	1.649	1.672
350	*t*	431.	431.9	456.9	481.9	506.9	531.9	556.9	581.9	606.9	631.9	681.9	731.9
	v		1.326	1.391	1.452	1.507	1.562	1.615	1.667	1.718	1.769	1.863	1.958
	h	408.0	1203.1	1222.0	239.2	255.3	270.7	285.6	1300.0	1314.0	1327.8	1355.1	1382.2
	n	0.6036	1.496	1.516	1.534	1.552	1.567	1.582	1.596	1.609	1.622	1.646	1.670
360	*t*	434.6	434.6	459.6	484.6	509.6	534.6	559.6	584.6	609.6	634.6	684.6	734.6
	v	...	1.291	1.352	1.413	1.466	1.519	1.572	1.622	1.672	1.722	1.814	1.906
	h	410	1203.2	1222.3	239.6	255.8	271.3	286.3	1300.8	1314.8	1328.6	1355.9	1383
	n	0.6068	1.494	1.513	1.533	1.550	1.566	1.581	1.594	1.608	1.620	1.645	1.668
370	*t*	437.2	437.2	462.	487.2	512.2	537.2	562.2	587.2	612.2	637.2	687.2	737.2
	v	.....	1.256	1.316	1.375	1.427	1.480	1.531	1.580	1.629	1.677	1.767	1.857
	h	413.7	1203.	1222.6	240.	256.3	271.9	286.9	1300.4	1315.5	1329.4	1356.9	1384.0
	n	0.6100	1.491	1.512	1.531	1.547	1.564	1.578	1.591	1.605	1.618	1.643	1.666
380	*t*	439.8	439.8	464.8	489.8	514.8	539.8	564.8	589.6	614.8	639.8	689.8	739.8
	v	....	1.223	1.282	1.339	1.391	1.443	1.493	1.540	1.588	1.635	1.722	1.810
	h	416.	1203.3	1222.8	1240.4	1256.8	1272.4	1287.5	1302.1	1316.5	1330.2	1357.7	1384.8
	n	0.613	1.489	1.510	1.529	1.546	1.561	1.576	1.590	1.603	1.616	1.640	1.664
390	*t*	442.	442.3	467.3	492.3	517.3	542.3	567.3	592.3	617.3	642.3	692.3	742.3
	v	.....	1.192	1.250	1.306	1.356	1.408	1.457	1.503	1.550	1.595	1.680	1.766
	h	419.	1203.	1223.1	1240.	1257.2	1272.9	1288.0	1302.6	1316.9	1330.9	1358.4	1385.6
	n	0.616	1.486	1.508	1.526	1.543	1.559	1.574	1.588	1.601	1.613	1.638	1.661
400	*t*	444.8	444.8	469.8	494.8	519.8	544.8	569.8	594.8	619.8	644.8	694.8	744.8
	v	......	1.162	1.220	1.275	1.324	1.374	1.423	1.468	1.514	1.558	1.641	1.725
	h	422.0	1203.5	1223.3	1241.0	1257.6	1273.4	1288.5	1303.2	1317.6	1331.6	1359.1	1386.4
	n	0.6190	1.483	1.506	.524	1.541	1.557	1.572	1.586	1.599	1.612	1.636	1.660
410	*t*	447.2	447.2	472.2	497.	522.2	547.	572.2	597.	622.2	647.2	697.2	747.2
	v	.....	1.134	1.191	1.245	1.293	1.342	1.391	1.435	1.479	1.522	1.603	1.685
	h	424.6	1203.5	1223.6	1241.4	1258.0	1273.9	1289.2	1303.9	1318.2	1332.2	1359.8	1387.1
	n	0.6219	.481	1.504	1.523	1.540	1.556	1.571	1.584	1.597	1.610	1.634	1.658

TABLE XXXV.—(*Continued*)

Absolute pressure, pounds per square inch p		Liquid	Superheat, degrees Fahrenheit 0	25	50	75	100	125	150	175	200	250	300
420	t	449.6	449.6	474.6	499.6	524.6	549.6	574.6	599.6	624.6	649.6	699.6	794.6
	v	......	1.107	1.163	1.216	1.264	1.311	1.360	1.403	1.446	1.488	1.567	1.647
	h	427.2	1203.5	1223.9	1241.7	1258.3	1274.3	1289.6	1304.3	1318.7	1332.8	1360.5	1387.8
	n	0.6247	1.479	1.501	1.520	1.538	1.553	1.569	1.583	1.595	1.608	1.633	1.656
440	t	454.2	454.2	479.2	504.2	529.2	554.2	579.2	604.2	629.2	654.2	704.2	754.2
	v	......	1.056	1.110	1.161	1.208	1.254	1.301	1.342	1.383	1.423	1.499	1.576
	h	432.3	1203.5	1224.1	1242.4	1259.4	1275.5	1290.8	1305.6	1320.0	1334.1	1361.9	1389.4
	n	0.6302	1.474	1.496	1.515	1.532	1.549	1.564	1.578	1.591	1.604	1.628	1.652
460	t	458.7	458.7	483.7	508.7	533.7	558.7	583.7	608.7	633.7	658.7	708.7	758.7
	v	......	1.010	1.062	1.111	1.157	1.201	1.246	1.285	1.325	1.362	1.436	1.511
	h	437.2	1203.3	1224.4	1243.0	1256.1	1276.3	1291.7	1306.6	1321.1	1335.3	1363.2	1390.9
	n	0.6355	1.470	1.492	1.512	1.526	1.545	1.560	1.574	1.588	1.601	1.625	1.648
480	t	463.1	463.1	488.1	513.1	538.1	563.1	588.1	613.1	638.1	663.1	713.1	763.1
	v	......	0.967	1.018	1.064	1.110	1.152	1.195	1.232	1.271	1.307	1.379	1.450
	h	422.0	1203.1	1224.7	1243 6	1260.8	1277.0	1292.5	1307.5	1322.1	1336.4	1364.5	1392.3
	n	0.6406	1.466	1.489	1.508	1.526	1.542	1.558	1.572	1.585	1.598	1.621	1.645
500	t	467.2	467.2	492.2	517.2	542.2	567.2	592.2	617.2	642.2	667.2	717.2	767.2
	v	......	0.928	0.976	1.021	1.066	1.107	1.149	1.185	1.221	1.256	1.326	1.394
	h	446.6	1202.9	1224.7	1243.9	1261.3	1277.7	1295.1	1308.4	1323.1	1337.5	1365.7	1393.6
	n	0.6455	1.462	1.485	1.504	1.522	1.538	1.554	1.569	1.582	1.595	1.619	1.642
520	t	471.3	471.3	496.3	521.3	546.3	571.3	596.3	621.3	646.3	671.3	721.3	771.3
	v	......	0.892	0.939	0.982	1.025	1.066	1.106	1.140	1.176	1.209	1.277	1.343
	h	451.1	1202.7	1224.9	1244.3	1261.9	1278.4	1294.1	1309.3	1324.1	1338.5	1366.8	1394.8
	n	0.6503	1.458	1.482	1.501	1.519	1.536	1.551	1.566	1.579	1.592	1.617	1.639
540	t	475.3	475.3	500.3	525.3	550.3	575.3	600.3	625.3	650.3	675.3	725.3	775.3
	v	......	0.858	0.904	0.945	0.988	1.027	1.065	1.099	1.133	1.166	1.232	1.295
	h	455.5	1202.4	1224.9	1244.6	1262.4	1279.0	1294.9	1310.2	1325.0	1339.5	1368.0	1396.0
	n	0.6549	1.454	1.479	1.498	1.517	1.532	1.548	1.562	1.576	1.589	1.614	1.636
560	t	479.1	4791	504.1	529.1	554.1	579.1	604.1	629.1	654.1	679.1	729.1	779.1
	v	......	0.827	0.872	0.912	0.953	0.992	1.028	1.060	1.094	1.126	1.190	1.250
	h	459.8	1202.0	1224.7	1244.7	1262.8	1279.6	1295.7	1311.1	1326.0	1340.5	1369.0	1397.1
	n	0.6594	1.450	1.476	1.495	1.512	1.529	1.544	1.559	1.573	1.586	1.610	1.634
580	t	482.8	482.8	507.8	532.8	557.8	582.8	607.8	632.8	657.8	682.8	732.8	782.8
	v	......	0.797	0.842	0.881	0.921	0.959	0.993	1.025	1.058	1.089	1.151	1.209
	h	463.9	1201.6	1225.1	1245.1	1263.2	1280.2	1296.5	1311.9	1326.9	1341.5	1367.0	1398.2
	n	0.6637	1.447	1.472	1.492	1.509	1.527	1.542	1.556	1.570	1.583	1.606	1.631
600	t	486.5	486.5	511.5	536.5	561.5	586.5	611.5	636.5	661.5	686.5	736.5	786.5
	v	......	0.769	0.813	0.851	0.891	0.927	0.961	0.992	1.024	1.055	1.115	1.171
	h	468.0	1201.2	1225.4	1245.4	1263.6	1280.8	1297.2	1312.8	1327.8	1342.4	1371.0	1399.2
	n	0.6679	1.443	1.469	1.489	1.507	1.523	1.539	1.554	1.567	1.580	1.604	1.628
620	t	490.1	490.1	515.1	540.1	565.1	590.1	615.1	640.1	665.1	690.1	740.1	790.1
	v	......	0.743	0.786	0.823	0.863	0.897	0.931	0.961	0.993	1.023	1.081	1.135
	h	472.0	1200.7	1225.4	1245.6	1264.0	1281.3	1297.8	1313.5	1328.6	1343.3	1372.0	1400.2
	n	0.6720	1.440	1.466	1.487	1.503	1.520	1.536	1.551	1.565	1.578	1.602	1.625
640	t	493.5	493.5	518.5	543.5	568.5	593.5	618.5	643.5	668.5	693.5	743.5	793.5
	v	......	0.719	0.760	0.798	0.836	0.870	0.903	0.932	0.963	0.993	1.049	1.102
	h	475.9	1200.2	1225.4	1245.8	1264.4	1281.8	1298.4	1314.2	1329.4	1344.2	1372.9	1401.2
	n	0.6760	1.436	1.463	1.484	1.501	1.516	1.533	1.547	1.562	1.575	1.560	1.622

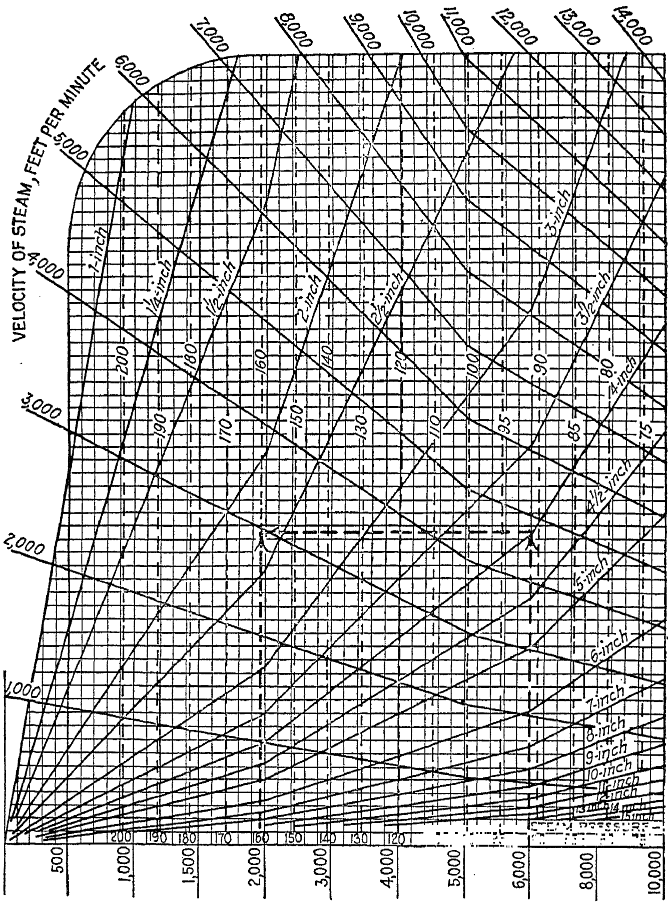
TABLE XXXV.—(*Continued*)

Absolute pressure, pounds per square inch p		Liquid	Superheat, degrees Fahrenheit 0	25	50	75	100	125	150	175	200	250	300
660	t	496.9	496.9	521.9	546.9	571.9	596.9	621.9	646.9	671.9	696.9	746.9	796.9
	v	......	0.696	0.736	0.774	0.810	0.844	0.875	0.904	0.935	0.964	1.019	1.071
	h	479.7	1199.6	1225.3	1245.9	1264.7	1282.3	1299.0	1314.9	1330.2	1345.0	1373.8	1402.1
	n	0.6799	1.433	1.460	1.481	1.499	1.514	1.531	1.546	1.559	1.573	1.597	1.620
680	t	500.2	500.2	525.2	550.2	575.2	600.2	625.2	650.2	675.2	700.2	750.2	800.2
	v	......	0.675	0.714	0.752	0.786	0.818	0.849	0.878	0.908	0.937	0.991	1.041
	h	483.5	1199.0	1225.2	1246.1	1265.1	1282.8	1299.6	1315.6	1330.9	1345.8	1374.6	1403.0
	n	0.6837	1.430	1.457	1.478	1.497	1.512	1.528	1.542	1.556	1.570	1.595	1.617
700	t	503.4	503.4	528.4	553.4	578.4	603.4	628.4	653.4	678.4	703.4	753.4	803.4
	v	......	0.655	0.693	0.730	0.764	0.794	0.825	0.854	0.883	0.911	0.964	1.013
	h	487.1	1198.3	1225.0	1246.2	1265.4	1283.3	1300.2	1316.3	1331.7	1346.6	1375.6	1403.9
	n	0.6874	1.426	1.454	1.475	1.494	1.510	1.526	1.541	1.555	1.568	1.593	1.615
725	t	507.3	507.3	532.3	557.3	582.3	607.3	632.3	657.3	682.3	707.3	757.3	807.3
	v	......	0.632	0.667	0.703	0.737	0.766	0.796	0.824	0.854	0.880	0.932	0.980
	h	491.6	1197.6	1224.9	1246.2	1265.6	1283.6	1300.6	1316.8	1332.3	1347.3	1376.5	
	n	0.6919	1.422	1.451	1.472	1.491	1.508	1.525	1.539	1.552	1.566	1.590	
750	t	511.1	511.1	536.1	561.1	586.1	611.1	636.1	661.1	686.1	711.1	761.1	811.1
	v	......	0.609	0.645	0.680	0.712	0.741	0.771	0.798	0.826	0.852	0.903	0.950
	h	495.9	1196.7	1224.8	1246.3	1265.8	1283.9	1301.0	1317.3	1332.9	1348.0	1377.4	
	n	0.6963	1.418	1.448	1.469	1.488	1.505	1.521	1.536	1.550	1.562	1.587	
775	t	514.9	514.9	539.9	564.9	589.9	614.9	639.9	664.9	689.9	714.9	764.9	814.9
	v	......	0.588	0.623	0.657	0.688	0.716	0.746	0.772	0.800	0.825	0.875	0.922
	h	500.1	1195.8	1224.6	1246.4	1265.9	1284.0	1301.3	1317.7	1333.4	1348.6	1378.2	
	n	0.7006	1.415	1.444	1.466	1.485	1.502	1.517	1.532	1.547	1.560	1.584	
800	t	518.5	518.5	543.5	568.5	593.5	618.5	643.5	668.5	693.5	718.5	768.5	818.5
	v	......	0.568	0.603	0.636	0.666	0.694	0.723	0.749	0.776	0.800	0.849	0.896
	h	504.3	1194.8	1224.4	1246.4	1266.0	1284.3	1301.6	1318.1	1333.9	1349.2	1379.1	
	n	0.7048	1.411	1.441	1.463	1.482	1.499	1.515	1.529	1.543	1.557	1.582	
825	t	522.0	522.0	547.0	572.0	597.0	622.0	647.0	672.0	697.0	722.0	772.0	
	v	......	0.549	0.583	0.616	0.646	0.673	0.701	0.727	0.753	0.776	0.824	
	h	508.4	1193.8	1224.0	1246.4	1266.0	1284.5	1301.9	1318.5	1334.4	1349.8	1379.9	
	n	0.7083	1.407	1.438	1.461	1.479	1.496	1.512	1.526	1.541	1.554	1.579	
850	t	525.5	525.5	550.5	575.5	600.5	625.5	650.5	675.5	700.5	725.5	775.5	
	v	......	0.532	0.565	0.597	0.626	0.653	0.681	0.706	0.732	0.754	0.801	
	h	512.4	1192.8	1223.8	1246.3	1266.0	1284.7	1302.2	1318.9	1334.9	1350.4	1380.7	
	n	0.7128	1.404	1.436	1.458	1.476	1.494	1.510	1.524	1.538	1.552	1.577	
875	t	528.9	528.9	553.9	578.9	603.9	628.9	653.9	678.9	703.9	728.9	778.9	
	v	......	0.516	0.547	0.579	0.608	0.634	0.662	0.686	0.711	0.733	0.779	
	h	516.4	1191.8	1223.5	1246.2	1266.0	1284.8	1302.4	1319.2	1335.3	1351.0	1381.5	
	n	0.7167	1.400	1.433	1.455	1.474	1.491	1.507	1.522	1.536	1.553	1.574	
900	t	532.3	532.3	557.3	582.3	607.3	632.3	657.3	682.3	707.3	732.3	782.3	
	v	......	0.500	0.531	0.562	0.590	0.617	0.644	0.668	0.692	0.713	0.758	
	h	520.3	1190.7	1223.0	1246.2	1266.0	1284.9	1302.6	1319.5	1335.8	1351.5	1382.2	
	n	0.7205	1.397	1.430	1.452	1.471	1.489	1.504	1.519	1.533	1.547	1.572	
925	t	535.6	535 6	560.6	585.6	610.6	635.6	660.6	685.6	710.6	735.6	785.6	
	v	......	0.485	0.516	0.546	0.574	0.601	0.626	0.651	0.673	0.694	0.738	
	h	524.1	1189.6	1222.5	1246.1	1266.0	1285.0	1302.8	1319.8	1336.2	1352.0	1382.8	
	n	0.7242	1.394	1.427	1.450	1.468	1.487	1.502	1.517	1.531	1.545	1.570	

TABLE XXXV.—(*Concluded*)

Absolute pressure, pounds per square inch p		Liquid	Superheat, degrees Fahrenheit 0	25	50	75	100	125	150	175	200	225	250
950	t	538.8	538.8	563.8	588.8	613.8	638.8	663.8	688.8	713.8	738.8	763.8	788.8
	v	......	0.471	0.502	0.531	0.558	0.585	0.610	0.634	0.656	0.676	0.701	0.720
	h	527.9	1188.5	1222.0	1246.0	1265.9	1285.0	1302.9	1320.1	1336.6	1352.5	1368.1	1383.5
	n	0.7278	1.390	1.424	1.448	1.466	1.484	1.500	1.515	1.529	1.542	1.555	1.568
975	t	541.9	541.9	566.9	591.9	616.9	641.9	666.9	691.9	716.9	741.9	766.9	791.9
	v	......	0.457	0.488	0.517	0.543	0.570	0.595	0.618	0.639	0.659	0.684	0.703
	h	531.6	1187.4	1221.5	1245.8	1265.8	1285.0	1303.1	1320.4	1337.0	1353.0	1368.7	1384.2
	n	0.7314	1.387	1.421	1.445	1.464	1.481	1.498	1.512	1.526	1.540	1.553	1.565
1000	t	544.9	544.9	569.9	594.9	619.9	644.9	669.9	694.9	719.9	744.9	769.9	794.9
	v	......	0.444	0.475	0.503	0.529	0.556	0.580	0.603	0.623	0.643	0.667	0.687
	h	535.2	1186.2	1220.8	1245.6	1265.7	1285.0	1303.2	1320.6	1337.4	1353.5	1369.3	1384.9
	n	0.7349	1.384	1.419	1.443	1.461	1.479	1.495	1.510	1.524	1.538	1.551	1.563
1025	t	547.9	547.9	572.9	597.9	622.9	647.9	672.9	697.9	722.9	747.9	772.9	
	v	......	0.432	0.462	0.490	0.516	0.542	0.566	0.589	0.608	0.628	0.651	
	h	538.7	1185.0	1220.3	1245.4	1265.6	1285.0	1303.3	1320.8	1337.7	1354.0	1369.9	
	n	0.7383	1.381	1.416	1.440	1.459	1.477	1.493	1.508	1.522	1.536	1.549	
1050	t	550.8	550.8	575.8	600.8	625.8	650.8	675.8	700.8	725.8	750.8	775.8	
	v	......	0.421	0.451	0.478	0.503	0.529	0.553	0.575	0.594	0.614	0.636	
	h	542.2	1183.8	1219.8	1245.2	1265.5	1285.0	1303.4	1321.0	1338.0	1354.5	1370.5	
	n	0.7417	1.378	1.414	1.438	1.458	1.475	1.491	1.506	1.520	1.534	1.547	
1075	t	553.7	553.7	578.7	603.7	628.7	653.7	678.7	703.7	728.7	753.7	778.7	
	v	......	0.410	0.440	0.466	0.491	0.516	0.540	0.562	0.581	0.600	0.622	
	h	545.7	1182.6	1219.3	1244.9	1265.4	1285.0	1303.5	1321.2	1338.3	1354.9	1371.0	
	n	0.7450	1.375	1.417	1.436	1.455	1.472	1.489	1.504	1.518	1.532	1.545	
1100	t	556.6	556.6	581.6	606.6	631.6	656.6	681.6	706.6	731.6	756.6	781.6	
	v	......	0.400	0.429	0.455	0.480	0.504	0.527	0.549	0.568	0.588	0.608	
	h	549.1	1181.4	1218.5	1244.6	1265.3	1235.0	1303.6	1321.4	1338.6	1335.3	1371.5	
	n	0.7482	1.372	1.409	1.434	1.452	1.470	1.486	1.502	1.516	1.530	1.543	
1125	t	559.4	559.4	584.4	609.4	634.4	659.4	684.4	709.4	734.4	759.4	784.4	
	v	......	0.390	0.418	0.444	0.469	0.492	0.515	0.537	0.556	0.576	0.595	
	h	552.5	1180.2	1217.8	1244.2	1265.2	1285.0	1303.7	1321.6	1338.9	1355.7	1372.0	
	n	0.7514	1.369	1.406	1.431	1.451	1.467	1.483	1.499	1.514	1.528	1.542	
1150	t	562.2	562.2	587.2	612.2	637.2	662.2	687.2	712.2	737.2	762.2	787.2	
	v	......	0.380	0.408	0.434	0.458	0.431	0.503	0.525	0.544	0.564	0.583	
	h	555.8	1179.0	1217.5	1243.8	1265.1	1285.0	1303.8	1321.8	1339.2	1356.1	1372.5	
	n	0.7545	1.366	1.404	1.428	1.448	1.466	1.482	1.498	1.512	1.526	1.540	
1175	t	565.0	565.0	590.0	615.0	640.0	665.0	690.0	715.0	740.0	765.0	790.0	
	v	......	0.370	0.398	0.424	0.443	0.470	0.493	0.513	0.533	0.552	0.571	
	h	559.1	1177.7	1216.8	1243.4	1265.0	1285.0	1303.9	1322.0	1339.5	1356.5	1373.0	
	n	0.7576	1.363	1.402	1.426	1.446	1.464	1.480	1.496	1.510	1.524	1.538	
1200	t	567.7	567.7	592.7	617.7	642.7	667.7	692.7	717.7	742.7	767.7	792.7	
	v	......	0.361	0.389	0.414	0.438	0.460	0.432	0.502	0.522	0.541	0.560	
	h	562.3	1176.4	1216.0	1243.0	1264.9	1284.9	1303.9	1322.1	1339.7	1356.8	1373.5	
	n	0.7607	1.360	1.399	1.424	1.444	1.462	1.478	1.494	1.508	1.522	1.536	

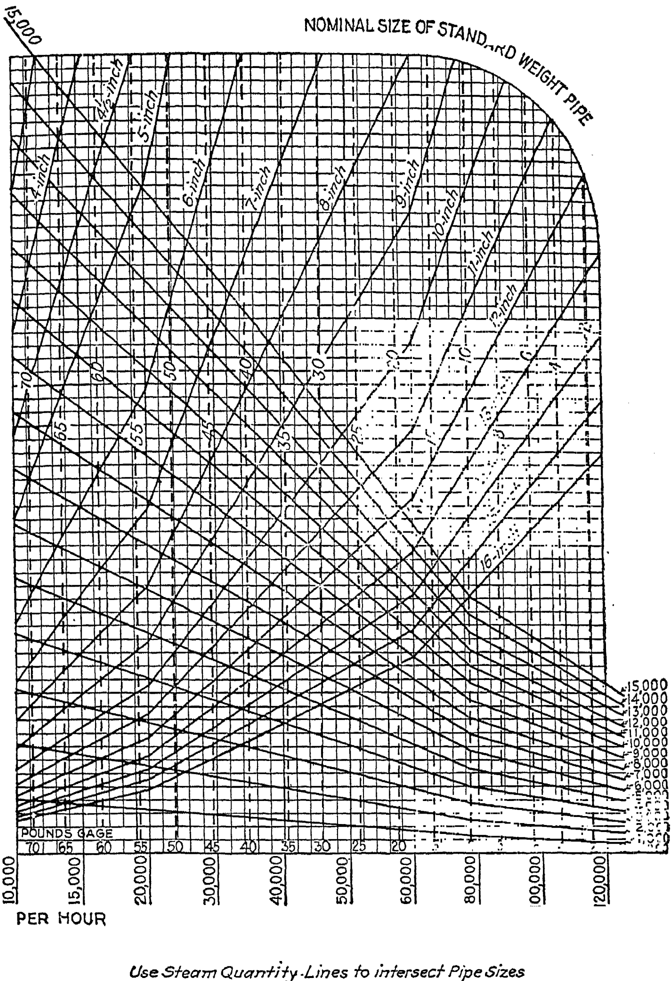
t = temperature in degrees Fahrenheit.
v = specific volume in cubic feet per pound.
h = total heat in Btu per pound.
n = entropy.



Typical Solution:
6000 Lb. Steam per Hour
4 Inch Pipe
160 Lb. Gage Pressure
3000 Ft. per Min. Velocity

When any three are known, the fourth can be determined

Fig. 35.—Saturated-steam flow chart for standard-weight (Schedule 40)
Prof. G. F.



pipe. *(Drawn by J. M. Spitzglass for "Steam Power Plant Engineering," by Gebhardt.)*

to the ASME Committee on the Properties of Steam and the Extension of the Steam Tables on Oct 20, 1925. This paper was published in *Mechanical Engineering* for February, 1926. The properties of steam for temperatures up to 320 F are in agreement with Marks and Davis's "Steam Tables." At higher temperatures, the properties are based on the above-mentioned ASME steam-table research.

Tables XXXIV and XXXV are based on the use of the following constants: 1 standard atm = 14.696 psi abs = 29.921 in. of mercury column at 32 F; absolute zero = 459.6 F; mechanical equivalent of work = 777.5 Btu per lb.

Flow of Steam in Pipes.—Where the quantity of flow and the pressure and temperature or quality of the steam are known, the velocity in a given size pipe readily can be calculated from steam-table data and the inside pipe areas given in Tables I to XII on pages 357 to 368. Similarly, where the velocity and steam properties are known, the size of pipe to give a certain discharge can be calculated in the same manner. In the case also where the quantity and velocity are given, the corresponding pipe area can be calculated and the proper size selected by reference to the above-mentioned tables giving pipe areas. A convenient chart in which these relations are worked out for standard weight pipe and dry saturated steam up to 200 lb gauge pressure is furnished in Fig. 35. The subject of allowable velocities and reasonable pressure drops is discussed in the section on "Determining Pipe Sizes," page 864.

Pressure Drop in Steam Pipes.—Methods of calculating pressure drop in pipes for fluids in general are discussed at some length on pages 81 to 137. The more common formulas in general use for figuring pressure drop in steam pipes are those of Babcock, Carpenter, Unwin, and Fritzsche. The first three named are practically identical, the only difference being a slight variation in the constant employed, which affects the result approximately 1 per cent. Fritzsche's formula is somewhat different in form from the others and is believed to give more accurate results for pipes 10 in. and larger carrying steam at high velocities, as stated on page 134. The graphical solution of Unwin's formula shown in Fig. 36 fits Babcock or Carpenter's formulas almost equally well. A graphical solution of Fritzsche's formula is given in Fig. 37. Attention is called also to the *rational method* of solution for flow problems described on pages 107 to 137.

In calculating pressure drop for compressible fluids such as steam, air, and gases, it is necessary to use average values for density or specific volume, if the drop exceeds 10 to 15 per cent of the initial absolute pressure or to employ the expression $(p_1^2 - p_2^2)$ in the flow formula. This is explained at greater length on pages 168 to 171.

Unwin's formula can be expressed in units common to engineering work as follows: The symbols used are identical with those given on page 82 in connection with the general friction-loss formulas but are repeated here for convenience. W = pounds of steam per hour; L = equivalent length of pipe in feet; d = inside diameter of pipe in inches; y = density of steam in pounds per cubic foot $= 1/V$ where V is the specific volume; p_λ = pressure drop in pounds per square inch; K is a coefficient combining the flow factor f with the formula constants of the general flow formula (see pages 175 to 177); and v = velocity in feet per second. Unwin's coefficient of friction for air and steam as expressed for use in the general flow formula is $f = 0.002705\left(\frac{1 + 3.6}{d}\right)$. Substituting this value in equation (56a) (page 87),

$$p_\lambda = \frac{fLW^2}{74{,}500yd^5} = \frac{0.002705\left(1 + \frac{3.6}{d}\right)LW^2}{74{,}500yd^5}$$

$$= \frac{3{,}625\left(1 + \frac{3.6}{d}\right)LW^2}{10^{11} \times yd^5} = \frac{KLW^2}{yd^5} = \frac{KVLW^2}{d^5}$$

where

$$K = \frac{3{,}625}{10^{11}}\left(1 + \frac{3.6}{d}\right).$$

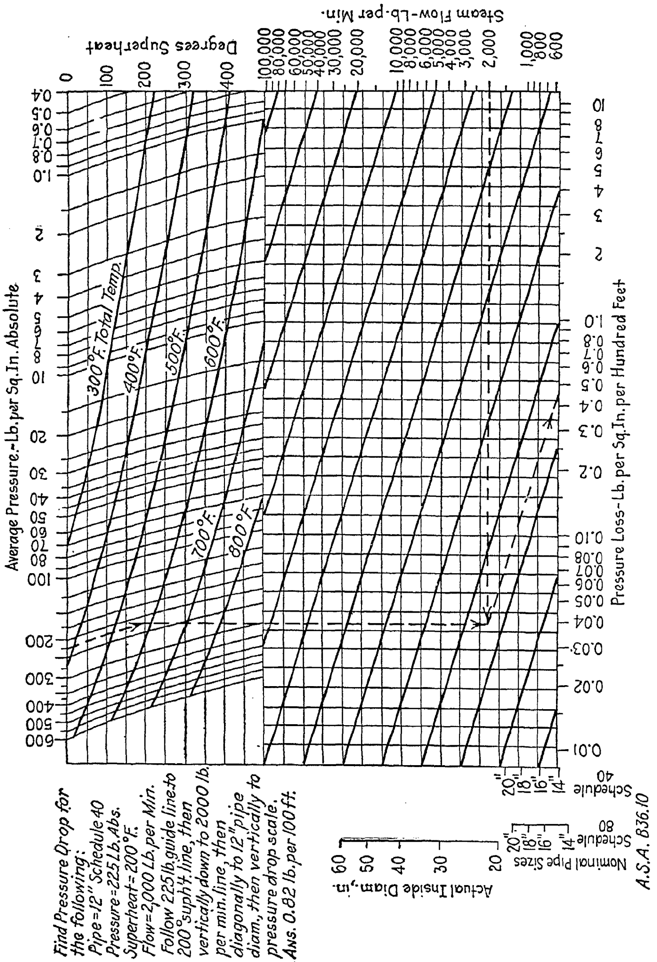
Where the weight of flow is desired in pounds per *minute* instead of per *hour*, $W = 60M$ can be substituted in the above equation with the following results:

$$p_\lambda = \frac{3{,}600KLM^2}{yd^5} = \frac{3{,}600KVLM^2}{d^5}.$$

The above equation can be converted readily to other forms convenient for certain types of problems by the general methods outlined on pages 175 to 177. Transposing and solving for W and M, respectively,

$$W = \sqrt{\frac{yp_\lambda d^5}{KL}} = \sqrt{\frac{p_\lambda d^5}{KVL}},$$

$$M = \frac{1}{60}\sqrt{\frac{yp_\lambda d^5}{KL}} = \frac{1}{60}\sqrt{\frac{p_\lambda d^5}{KVL}}$$


Steam Flow-Lb. per Min.
Degrees Superheat
Average Pressure,-Lb. per Sq. In. Absolute
Pressure Loss-Lb. per Sq. In. per Hundred Feet
300°F. Total Temp.
400°F.
500°F.
600°F.
700°F.
800°F.
Actual Inside Diam., in.
Nominal Pipe Sizes
Schedule 40
Schedule 80
A.S.A. B36.10
Find Pressure Drop for the following:
Pipe=12" Schedule 40
Pressure=225 Lb. Abs.
Superheat=200°F.
Flow=2,000 Lb. per Min.
Follow 225 lb. guide line to 200° supht. line, then vertically down to 2000 lb. per min. line, then diagonally to 12" pipe diam., then vertically to pressure drop scale.
Ans. 0.82 lb. per 100 ft.

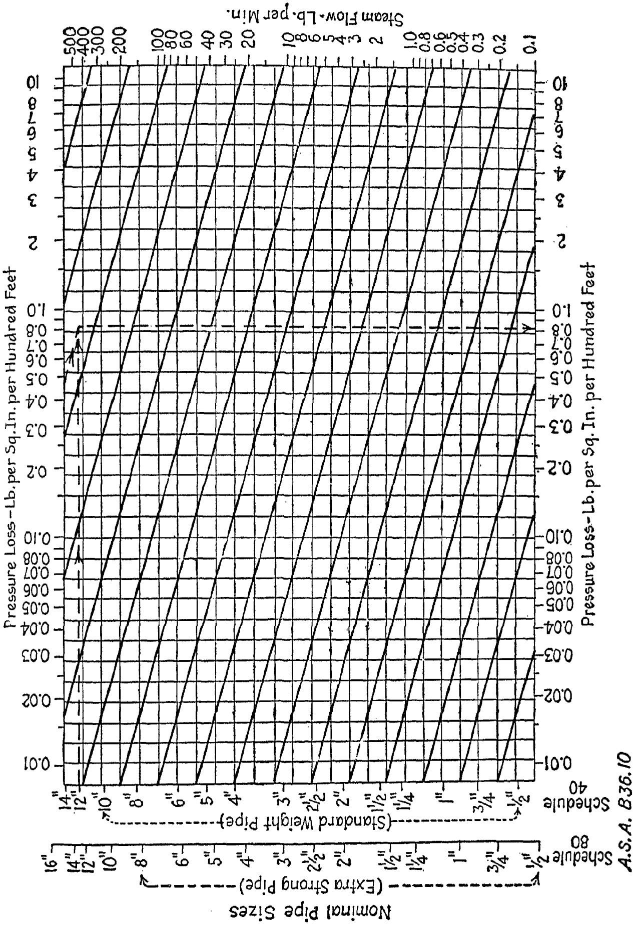
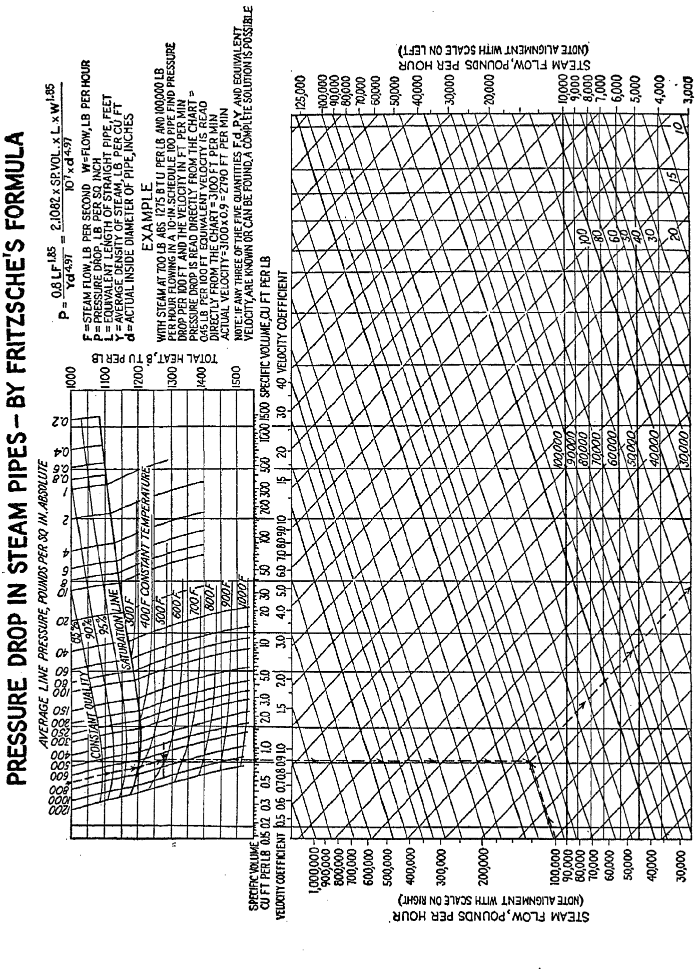


FIG. 33.—Graphical solution of Unwin's[1] formula for flow of steam in pipes.

[1] All formulas for flow of steam in pipes are of the form $p_\lambda = KLW^2 \div yd^5$. Values of K ascribed by Unwin, Babcock, Carpenter, and Martin give practically the same result.


PRESSURE DROP IN STEAM PIPES — BY FRITZSCHE'S FORMULA
P = 0.8 L F^1.85 / Y d^4.97 = 2.1082 x SP.VOL. x L x W^1.85 / 10^7 x d^4.97
F = STEAM FLOW, LB PER SECOND W = FLOW, LB PER HOUR
P = PRESSURE DROP, LB PER SQ INCH
L = EQUIVALENT LENGTH OF STRAIGHT PIPE, FEET
Y = AVERAGE DENSITY OF STEAM, LB PER CU FT
d = ACTUAL INSIDE DIAMETER OF PIPE, INCHES
EXAMPLE
WITH STEAM AT 700 LB ABS 1275 BTU PER LB AND 100,000 LB PER HOUR FLOWING IN A 10-IN. SCHEDULE 100 PIPE, FIND PRESSURE DROP PER 100 FT AND THE VELOCITY IN FT PER MIN
PRESSURE DROP IS READ DIRECTLY FROM THE CHART = 0.45 LB PER 100 FT EQUIVALENT VELOCITY IS READ DIRECTLY FROM THE CHART = 3100 FT PER MIN
ACTUAL VELOCITY = 3100 x 0.9 = 2790 FT PER MIN
NOTE: IF ANY THREE OF THE FIVE QUANTITIES F, d, P, Y AND EQUIVALENT VELOCITY, ARE KNOWN OR CAN BE FOUND, A COMPLETE SOLUTION IS POSSIBLE
AVERAGE LINE PRESSURE, POUNDS PER SQ IN. ABSOLUTE
TOTAL HEAT, B T U PER LB
CONSTANT QUALITY
85%
90%
95%
SATURATION LINE
300 F
400 F CONSTANT TEMPERATURE
500 F
600 F
700 F
800 F
900 F
1000 F
SPECIFIC VOLUME, CU FT PER LB
VELOCITY COEFFICIENT
STEAM FLOW, POUNDS PER HOUR
(NOTE ALIGNMENT WITH SCALE ON LEFT)
STEAM FLOW, POUNDS PER HOUR
(NOTE ALIGNMENT WITH SCALE ON RIGHT)

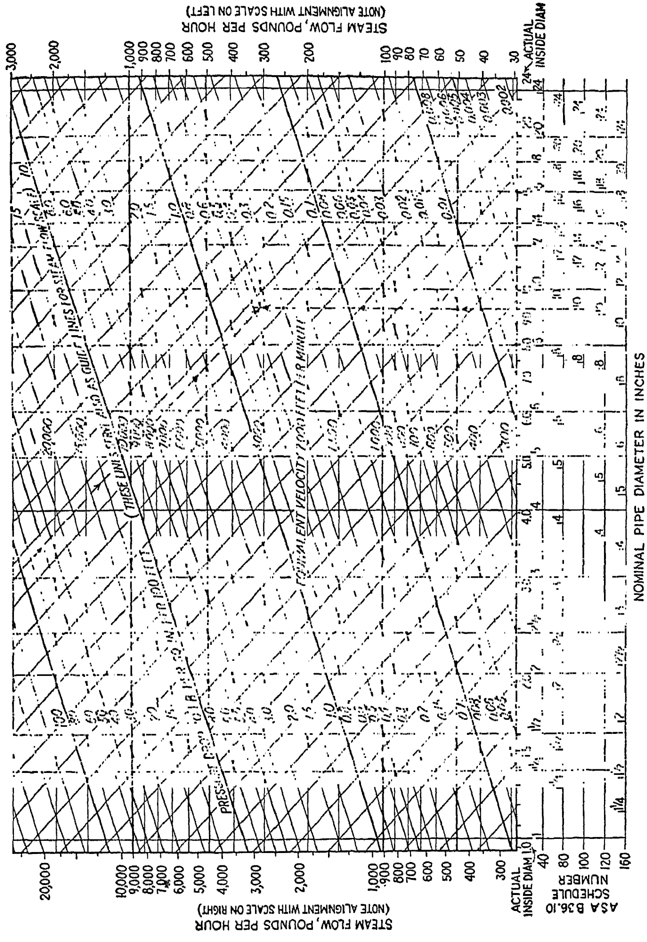


FIG. 37.—Pressure drop in steam pipes by Fritzsche's formula.

The velocity in feet per second can be obtained by substituting for W, $W = 3{,}600Avy$ and solving for v,

$$W = \frac{3{,}600\pi d^2vy}{4 \times 144} = \sqrt{\frac{yp_\lambda d^5}{KL}}$$

$$v = 0.05095\sqrt{\frac{p_\lambda d^5}{yKL}} = 0.05095\sqrt{\frac{p_\lambda V d^5}{KL}}.$$

Likewise $M = 60Avy$ and

$$v = 3.057\sqrt{\frac{p_\lambda d^5}{yKL}} = 3.057\sqrt{\frac{p_\lambda V d^5}{KL}}.$$

Those wishing to solve directly for d in Unwin's formula can refer to a tabular method devised by Charles A. Pohlig which was published in *Heating, Piping and Air Conditioning*, pp. 570 and 571, October, 1944.

Fritzsche's formula can be expressed in convenient engineering units, using the same symbols.

$$p_\lambda = \frac{52y^{0.85}Lv^{1.85}}{10^6 \times d^{1.27}},$$

$$p_\lambda = \frac{2.1082LW^{1.85}}{10^7 \times yd^{4.97}} = \frac{2.1082VLW^{1.85}}{10^7 \times d^{4.97}}.$$

Similar transpositions to those given above for Unwin's formula can be made to solve for W or v. These solutions involve the taking of fractional roots, which can be accomplished either by the use of logarithms or by a log-log slide rule. It is interesting to note the close similarity between Fritzsche's formula for steam and that of Saph and Schoder for water given on page 270, the only practical difference being in the numerical constant which adjusts conditions to fit the particular fluid in question.

Use of Pressure-drop Charts.—The graphical solutions for the Unwin and Fritzsche formulas given in Figs. 36 and 37, respectively, cover the entire range of steam conditions encountered in ordinary engineering practice. The form of the Fritzsche equation is such that it lends itself to the graphic presentation of more steam-flow data than is possible with the Unwin-Carpenter-Babcock type of equation. The sample problems on the charts are self-explanatory, but it is desirable to emphasize here that it is not necessary to start with the same quantities known and unknown as are used in the illustrations. Any combination will work out provided there are not too many unknowns. The follow-

ing explanation regarding the sample problem shown on the Fritzsche chart will aid in tracing it through on the chart.

NOTES ON THE USE OF THE FRITZSCHE-FORMULA PRESSURE-DROP CHART FOR STEAM

Example.—Steam pressure = 700 psi abs;
Total heat = 1,275 Btu per lb;
Weight of steam flowing = 100,000 lb per hr;
Pipe diameter = 10 in. ASA Schedule 100.

Solution.—Entering the chart at the correct point of pressure (700 lb) shown by the scale at the top, proceed along a constant pressure line to the point where it intersects the total heat, total temperature, or quality line given for the particular problem at hand. (1,275 Btu total heat.) From there proceed vertically downward to the flow desired (100,000 lb per hr) noting the velocity coefficient at the point of crossing the velocity coefficient scale (0.90). Thence, proceed to the right or left (right in this case) along the diagonal guide lines which slope from upper left to lower right to the pipe-diameter line (10 in. ASA Schedule 100). At that point read pressure drop in pounds per square inch per 100 ft of pipe (0.45) and the equivalent velocity in feet per minute (3,100 ft). On applying the velocity coefficient the actual velocity in the pipe is found to be $3{,}100 \times 0.9 = 2{,}790$ ft per min. With a little practice in the use of the chart, it may be worked in any direction with ease and entered at the points for which data are available.

It may be observed that the representation of steam conditions is nothing but a total heat *vs.* volume chart for steam and that the only scale involved which essentially belongs to the pressure-drop chart proper is the specific-volume scale. In this connection, attention is called to the fact that any steam condition not covered by the chart may still be worked through the chart by obtaining the specific volume from a steam table and starting from that point on the specific-volume scale.

It should be noted that the equivalent-velocity scale is not laid out for a given pipe size, as is the case with a different type of steam-pressure drop chart in common use. The equivalent velocity here given is the velocity which would exist at a given pressure drop per 100 ft if the specific volume of the steam were unity. The velocity coefficient is then a correction for specific volume of the steam flowing in the pipe and is independent of the pipe diameter.

Pressure Drop in Fittings.—The resistance of fittings and valves expressed in equivalent length of straight pipe is given in Table XIV on page 100. The equivalent resistance of 90-deg elbows can also be determined by reference to Fig. 15*a* on page 108. Suggestions regarding friction factors to use with creased bends and corrugated pipe will be found in item 5 on page 133.

Flow of Steam through Nozzles and Orifices.—See pages 68 to 80.

Calorimeters for Saturated Steam.—In the case of saturated or wet steam where pressure and temperature alone are not sufficient for defining the heat content, it is necessary to determine the

amount of moisture or entrained water present. For this purpose an appropriate design of steam calorimeter should be selected from the numerous types available and described in the ASME Power Test Codes, Information on Instruments and Apparatus, Part 11, Determination of Quality of Steam, PTC 19.11.[1]

FLOW OF GASES AND VAPORS IN PIPES LIMITED BY ACOUSTIC VELOCITY

The concept that with the flow of expansive fluids through pipes there is a limiting velocity which is identical with the velocity of sound in the same medium can be explained through associating accepted pipe flow principles with critical pressure as it governs the weight of discharge through nozzles and orifices.[2] According to the flow equations given in the section on Friction Loss in Pipes starting on page 81, the carrying capacity of a given size of pipe operating with a fixed amount of pressure loss varies inversely with the first power of its length. Hence, in approaching zero length a pipe would have infinite carrying capacity and the corresponding velocity of flow would be infinite. But this limiting case of zero length is represented by an orifice or nozzle, which experience and mathematical derivations both prove to have a discharge capacity for compressible fluids that is limited by the acoustic velocity, or velocity of sound in the fluid at its particular condition of pressure and specific volume in the nozzle throat (see page 73).

A pipe of uniform diameter is rarely thought of as a nozzle, but there are applications of such piping operating at high capacity where the acoustic velocity phenomenon occurs. In these cases the expansion of the fluid occurs entirely within the pipe and the velocity of flow at the downstream end cannot exceed the value computed from the equation[3]

$$v_t = 68.3\sqrt{kp_tV_t}$$

[1] Published by the American Society of Mechanical Engineers, 29 West 39th St., New York 18, N.Y.

[2] See "How to Design Steam Piping for Maximum Capacity and High Pressure Drop," by Max W. Benjamin, *Heating, Piping and Air Conditioning*, September, 1936, pp. 475–478.

[3] This relation is derived as follows:

The kinetic energy in ft-lb per lb of fluid at the throat velocity is

$$K.E. = \frac{v_t^2}{2g}. \qquad (1)$$

(For definition of additional symbols see pages 69 and 82.) Since no external work is done as the fluid flows through a pipe or nozzle, the increase in kinetic

where v_t = acoustic velocity, ft per sec.
k = adiabatic exponent for the particular fluid (see Table XXIII, page 166).
p_t = pressure at the downstream end of the pipe, psi abs.
V_t = specific volume, cu ft per lb, of fluid in the downstream end of the pipe at pressure p_t.

For steam, the acoustic velocity usually falls within the range of 1,300 to 2,100 ft per sec, depending upon the pressure and specific volume conditions.

---

energy (assumed at zero initially) is equal to the change of enthalpy, *viz.*, $K.E. = H_1 - H_t$. Also, from thermodynamics,

$$H_1 - H_t = c_p(T_1 - T_t) \text{ Btu per lb,}$$
$$= 778c_p(T_1 - T_t) \text{ ft-lb per lb.} \qquad (2)$$

Combining equations (1) and (2), $\frac{v_t^2}{2g} = 778c_p(T_1 - T_t)$. (3)

From the equation of a perfect gas, (p. 144),

$$T = \frac{PV}{R} = \frac{144pV}{R} \qquad (4)$$

and by substitution in equation (3)

$$\frac{v_t^2}{} = \frac{778c_p \times 144(p_1V_1 - p_tV_t)}{} \qquad (5)$$

However, from thermodynamics, $R = 778(c_p - c_v)$, hence $c_p = \frac{R}{778} + c_v$ and, by definition, $k = c_p/c_v$.

Combining the foregoing expressions, $c_p = \frac{R}{778} + \frac{c_p}{k}$.

Transposing and rearranging, $\frac{c_p(k-1)}{k} = \frac{R}{778}$, and $\frac{c_p}{R} = \frac{k}{778(k-1)}$. (6)

Substituting in (5), $\frac{v_t^2}{2g} = \frac{144k}{(k-1)}(p_1V_1 - p_tV_t)$. (7)

Therefore, $v_t = \sqrt{\frac{144k(p_1V_1 - p_tV_t)}{k-1}}$. (8)

Also from the equation of the adiabatic in thermodynamics $\frac{p_t}{p_1} = \left(\frac{V_1}{V_t}\right)^k$. (9)

And from equation (32) (p. 73), $\frac{p_t}{p_1}$ (10)

Equating (9) and (10) $\frac{V_1}{V_t}$ (11)

and dividing (11) by (10) p_tV_t (12)

Substituting (12) in (8) $v_t = \sqrt{2g \times 144kp_tV_t \left(\frac{1}{} - 1\right)}$ (13)

from which $v_t = 12\sqrt{gkp_tV_t} = 68.3$ (14)
= the same value given in physics texts for the velocity of sound.

In a pipe of uniform diameter carrying a compressible fluid the maximum velocity must occur at its downstream end if the expansion of the fluid occurs entirely within the pipe because friction losses persist to the end of the pipe. As the pressure falls, the specific volume and, therefore, the velocity increase. The pressure in the end of a pipe will not be the same as the critical pressure in a nozzle (see pages 73 to 81), although it will approach it if the pipe is short. A pipe, with friction loss, operating at maximum carrying capacity is analogous to an orifice or convergent nozzle preceded by a throttling device. With a given initial pressure a longer length of pipe corresponds to additional throttling, reducing the rate of flow and also the pressure at the pipe outlet just as additional throttling ahead of a nozzle would reduce the weight of flow through it and also its throat pressure. Figure 38 qualitatively shows the effect of additional pipe length on the capacity and outlet pressure corresponding to the acoustic velocity. The mathematical limits of the effect of pipe length are those corresponding to (1) an extremely short pipe, or nozzle, for which the acoustic velocity is reached with an outlet pressure equal to 54.4 per cent (for superheated steam) of the inlet pressure and (2) a pipe length approaching infinity for which the acoustic velocity would be reached with an outlet pressure approaching 0 psi abs and weight of flow approaching 0.

It is possible for a pipe to carry a compressible fluid at a velocity *higher* than that of sound only if the acoustic velocity is attained through a nozzle or orifice preceding the pipe and further acceleration takes place subsequent to the nozzle, either in a diverging tube or in the natural expansion of the jet to discharge-pipe diameter.[1] If supersonic velocity is attained, however, it exists only through some intermediate section of the pipe and the acoustic velocity cannot be exceeded at the outlet of the pipe (except with a very short length of pipe) where it discharges to the atmosphere or into a receiver. Hence the principles of acoustic velocity set forth herein can be used as a basis of design for pipes operating at maximum capacity such as those illustrated in the examples.

[1] See (*a*) "Steam and Gas Turbines," by Stodola-Loewenstein, McGraw-Hill Book Company, Inc., New York, 1927, pp. 82–105.

(*b*) "Flow of Gases at a Rate Exceeding the Acoustic Velocity," by O. G. Tietjens, *Trans. ASME*, 1931, paper APM-53-4, pp. 49–58.

(*c*) "Flow in Smooth Straight Pipes at Velocities above and below Sound Velocity," by W. Frossel, National Advisory Committee on Aeronautics, *Techn. Memorandum* 844, January, 1938.

Reference to Fig. 38 shows that for relatively short pipes of uniform diameter the possibility of flow at the acoustic velocity exists whenever the conditions are such that expansion of the fluid within the pipe could take place through a pressure drop amounting to about 50 per cent of the inlet pressure. With longer pipes the acoustic velocity will be achieved only with larger pressure drops.

In the case of pipes carrying steam, the equation and coefficients of Table XXXVI give a simple check on whether the acoustic velocity will be reached. In the case of pipes carrying air or gas a corresponding check on whether the acoustic velocity will be

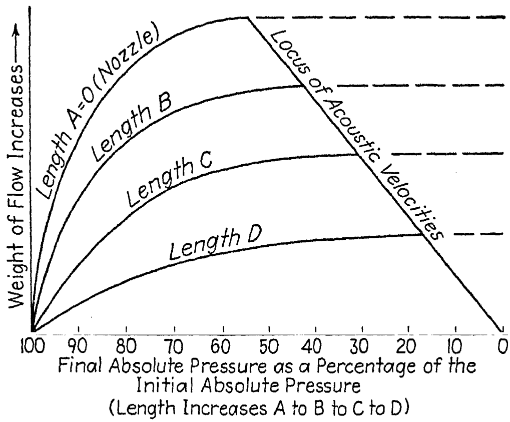


FIG. 38.—Relationship between initial pressure, final pressure, weight of flow, and pipe length for pipes carrying compressible fluids at maximum capacity.

reached is made through the equation and coefficients given in Table XXXVII. The acoustic velocity will not be the limiting factor if p_t as computed from Table XXXVI or XXXVII is less than the outlet pressure needed for actual operation, in which event the capacity can be determined by the usual empirical flow formulas such as those of Unwin, Spitzglass, Fritzsche, and Weymouth, or by the rational method. On the other hand, should p_t be computed to be greater than the outlet pressure needed for actual operation, the acoustic velocity will limit the capacity. The actual capacity of such a line will be less than that determined by the usual flow formulas except as they are applied with the proper value of p_t at the pipe outlet rather than with the lower

TABLE XXXVI.—EQUATION AND COEFFICIENTS FOR CRITICAL PRESSURE p_t CORRESPONDING TO THE ACOUSTIC VELOCITY AT THE OUTLET OF PIPES CARRYING STEAM

Equation for critical pressure, $p = \frac{W}{Cd^2}$

Enthalpy of Steam at Beginning of Line, Btu per Lb	Value of Coefficient C
1,150 (saturated)	87.7
1,200 (saturated)	80.9
1,150 (superheated)	82.0
1,200 (superheated)	75.6
1,250 (superheated)	71.4
1,300 (superheated)	67.7
1,350 (superheated)	64.4
1,400 (superheated)	61.6
1,450 (superheated)	59.0
1,500 (superheated)	56.7

Derivation of Formulas and Constants Shown in Table XXXVI.

Acoustic velocity $= v_t = 68.3\sqrt{kp_tV_t}$

$223.7\sqrt{H_1 - H_t}$ [see equation (33), p. 74]

and from F v [equation (19), p. 52], v $\quad \frac{W \times V_t \times 144}{} = \frac{WV_t}{25d^2} = \quad WV_t$

$$\frac{W}{d^2} = \frac{19.67v_t}{V_t} = \frac{4{,}400\sqrt{H_1 - H_t}}{V_t}$$

where W = weight of steam, lb per hr.
d = diameter of pipe, in.
H_1 = enthalpy of steam at pipe inlet, Btu per lb.
H_t = enthalpy of steam at pipe outlet, Btu per lb.

The other terms are as defined on pages 69 and 82.

In the foregoing equations the value of $H_1 - H_t$ is practically constant, irrespective of the actual pressures involved, provided H_1 is constant and provided the ratio of initial pressure to final pressure is constant. Furthermore, if H_1 is constant, the value of V_t is practically in inverse ratio to p_t.

Therefore
$$\frac{W}{d^2} = \frac{C'}{V_t} = \frac{C'}{\frac{C''}{p_t}} = Cp_t$$

or
$$p_t = \frac{W}{Cd^2}$$

The straight-line form of this equation may be verified and checked analytically as follows:

Assume $\qquad H_1 = 1{,}400 \quad \text{and} \quad k = 1.3,$

$$p_t = p_1\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} = p_1\left(\frac{2}{2.3}\right)^{\frac{1.3}{0.3}} = p_1 \times 0.869^{4.33} = 0.544p_1.$$

Assume $p_1 = 100$ psi abs; then $p_t = 0.544 \times 100 = 54.4$ psi abs. Isentropic expansion to 54.4 psi abs from $p_1 = 100$ psi abs and $H_1 = 1{,}400$ gives $H_t = 1{,}326$.

$V_t = 11.3$ cu ft per lb $\quad$ when $\quad H_t = 1{,}326 \quad$ and $\quad p_t = 54.4$ psi abs.

$$\frac{W}{d^2} = \frac{4{,}400\sqrt{1{,}400 - 1{,}326}}{11.3} = 3{,}350.$$

Other values of W/d^2 are computed similarly for additional values of p_t. These values of W/d^2 when plotted against p_t lie on an essentially straight line from which the value of C may be determined.

TABLE XXXVII.—EQUATION AND COEFFICIENTS FOR CRITICAL PRESSURE p_t CORRESPONDING TO THE ACOUSTIC VELOCITY AT THE OUTLET OF PIPES CARRYING AIR OR GAS

$$\text{Equation for critical pressure, } p_t = \frac{W}{d^2} \times \frac{\sqrt{T_1}}{C}$$

Gas	Value of C	Gas	Value of C
Acetylene.	2,520	Methane.	2,030
Air.	2,870	Methyl chloride.	3,330
Ammonia.	2,080	Natural gas (see Table XXI, p. 16[illegible])	
Argon	3,830	87% CH_4 : [illegible]	2,070
Carbon dioxide . .	3,330	7[illegible] CH_4 : [illegible]	2,330
Carbon monoxide.	2,820	79.4[illegible] CH_4 : 20% [illegible]	2,670
Ethylene.	2,480	Nitric oxide.	2,900
Helium.	1,210	Nitrogen.	2,800
Hydrochloric acid . .	3,200	Nitrous oxide.	3,230
Hydrogen.	750	[illegible]	2,990
		[illegible]r dioxide.	3,870

Derivation of Formula and Constants Shown in Table XXXVII (for definition of symbols see pages 69 and 82).—From the equation of a perfect gas, p. 144, and noting that P is in pounds per square foot,

$$PV = WRT, \quad \text{and for 1 lb,} \quad PV = RT. \tag{1}$$

Acoustic velocity (see footnote, page 255) $\quad v_t = \sqrt{gkP_tV_t}. \quad$ (2)

Combining (1) and (2) $\quad v_t = \sqrt{gkRT_t}. \quad$ (3)

$$\text{Kinetic energy of the gas in Btu} = \frac{v_t^2}{2gJ} = (H_1 - H_t) = c_p(T_1 - T_t). \tag{4}$$

Combining (3) and (4) $\quad \frac{gkRT_t}{2gJ} = c_p(T_1 - T_t), \quad$ (5)

from which $\quad (T_1 - T_t) = \frac{kRT_t}{2Jc_p} \quad \text{and} \quad T_1 = T_t\left(1 + \frac{kR}{2Jc_p}\right). \quad$ (6)

But $\quad R = J(c_p - c_v) \quad \text{and} \quad \frac{c_p}{c_v} = k; \quad \text{also} \quad c_v = \frac{c_p}{k},$

from which $\quad R = J\left(c_p - \frac{c_p}{k}\right) = Jc_p\left(1 - \frac{1}{k}\right) = Jc_p\left(\frac{k-1}{k}\right) \quad$ (7)

Substituting (7) in (6), $\quad T_1 = T_t\left(1 + \frac{kJc_p(k-1)}{2Jc_pk}\right)$

$$= T_t\left(1 + \frac{k-1}{2}\right) = T_t\left(\frac{k+1}{2}\right), \tag{8}$$

$$T_t = \frac{2T_1}{k+1}. \tag{9}$$

Substituting (9) in (3), $\quad v_t = \sqrt{\frac{2gkRT_1}{k+1}}. \quad$ (10)

From equation (19), p. 52, $\quad F = Av \quad \text{or} \quad v = \frac{F}{A}.$

Therefore, $\quad v_t = \frac{F_t}{A_t} = \frac{WV_t \times 144}{3{,}600 \times \frac{\pi d^2}{4}} = \frac{WV_t}{19.67d^2}$

and
$$\frac{W}{d^2} = \frac{19.67v_t}{V_t}. \tag{11}$$

From the equation of a perfect gas, p. 144. $\frac{P_tV_t}{T_t} \quad P_1V_1$

and
$$V_t = \frac{T_tP_1V_1}{P_tT_1}. \tag{12}$$

Combining (12) with (9), $V_t = \dfrac{2T_1P_1V_1}{P_tT_1(k+1)} = \dfrac{2P_1V_1}{P_t(k+1)}$, (13)

and, substituting from (1), $V_t = \dfrac{2RT_1}{P_t(k+1)}$ (14)

Combining (11), (10), and (14).

$$\frac{W}{d^2} = \frac{19.67\sqrt{\dfrac{2gkRT_1}{k+1}}}{\dfrac{2RT_1}{P_t(k+1)}} = \frac{9.88P_t(k+1)\sqrt{\dfrac{2gkRT_1}{k+1}}}{RT_1}$$
$$= 79.2P_t\sqrt{\frac{k(k+1)}{RT_1}}. \tag{15}$$

Changing to psi and solving for p_t,

$$p_t = \frac{P_t}{144} = \frac{\dfrac{W}{d^2}}{144 \times 79.2\sqrt{\dfrac{k(k+1)}{RT_1}}} = \frac{W}{d^2} \times \frac{\sqrt{\dfrac{RT_1}{k(k+1)}}}{11{,}400}. \tag{16}$$

For any particular gas, R and k have definite values so that in general

$$p_t = \frac{W}{d^2} \times \frac{\sqrt{T_1}}{C}. \tag{17}$$

Values of C for different gases are given in Table XXXVII.

value of outlet pressure originally expected. A few trials may be needed to obtain an outlet pressure for flow calculation that will agree with the value of p_t corresponding to that flow.

The foregoing limitations are encountered in numerous applications where two regions of different pressure are connected by a pipe. In some cases the phenomenon occurs from physical necessity for keeping pipe sizes within reasonable limits when handling fluids at low absolute pressures. In other cases it may be desired to operate a section of pipe at high velocity and pressure drop, perhaps in lieu of a pressure-reducing valve. As an illustration consider the following example:

Example.—A steam turbine driving an emergency boiler-feed pump exhausts 40,000 lb of steam per hour through an equivalent length of 150 ft of 16-in. Schedule 20 pipe to a main unit condenser where the pressure is 0.49 psi abs. Enthalpy of the exhaust steam based on the manufacturer's data is 1220 Btu per lb. What is the pressure at the turbine exhaust?

Solution.—Since the pipe exhausts to a region of 0.49 psi abs pressure, it follows that an initial pressure of only 1.0 psi abs would be high enough to indicate the possibility of an acoustic velocity condition at the pipe outlet. Therefore

the outlet should be checked from Table XXXVI to find out what its critical pressure would be. Interpolating from Table XXXVI to obtain the critical pressure corresponding to initial enthalpy of 1220 Btu gives $p_t = W/73.9d^2$; W is given as 40,000, and d^2 for a 16-in. Schedule 20 pipe is obtained from Table IX, page 365, as 236.42.

$$\text{Hence } \frac{W}{d^2} = 40{,}000 \div 236.42 = 169 \quad \text{and} \quad p_t = 169 \div 73.9 = 2.29 \text{ psi abs.}$$

This shows that the pressure in the pipe at the outlet end will be considerably higher than the condenser pressure into which the pipe discharges. The difference of $2.29 - 0.49 = 1.80$ psi will be lost in turbulence of free expansion.

After estimating the average specific volume of the steam in the pipe, the pressure drop can be calculated by any of the methods that seems appropriate. With an estimated specific volume of 149 cu ft per lb the pressure drop by Fritzsche's formula (see page 252) is

$$p_\lambda = \frac{2.1082 \times 150 \times 40{,}000^{1.852} \times 149}{10^7 \times 15.375^{4.973}} \quad 1.92 \text{ psi.}$$

The turbine exhaust pressure is the sum of this pressure drop and the pipe outlet pressure, *viz.*, $2.29 + 1.92 = 4.21$ psi abs.

Steam Flow through Safety Valves at or near the Acoustic Velocity.—The design of steam safety-valve vent lines constitutes a special application of the principles just discussed since a proper relationship must be established between the safety valve and its vent pipe if an umbrella fitting is used[1] (see Fig. 11, page 891). Operation of this type of vent system depends upon the kinetic energy of the steam leaving the safety valve to overcome all resistance offered by the vent pipe, thus avoiding blowback through the open umbrella connection. In many cases steam leaves the valve at or above the acoustic velocity and, in all cases, with considerable kinetic energy or velocity head. The vent pipe also may operate with steam flowing at the acoustic velocity at the outlet end and, when this is true, the static pressure near the vent-pipe outlet may exceed atmospheric. In no case will the static pressure there be less than atmospheric. Friction loss from inlet to outlet results in a static pressure greater than atmospheric near the vent-pipe inlet in all cases.

The criterion of successful operation of an umbrella-fitted vent pipe is that the velocity head of the steam leaving the safety valve shall at least equal the static gauge pressure existing in the vent pipe at its inlet end. Because there may be uncertain turbulence

[1] See "Sizing Vent Piping for Safety Valves," by Max W. Benjamin, *Heating, Piping and Air Conditioning*, October, 1941, pp. 615–619; also "How to Design Safety-valve Vent Piping of the Umbrella Type," by Max W. Benjamin, *Heating, Piping and Air Conditioning*, December, 1943, pp. 655 to 659.

effects at the vent-pipe entrance, it is recommended that some definite margin of velocity head be provided in excess of the static gauge pressure in the vent pipe inlet. The design should be checked as follows:

1. Using the appropriate equation in Table XXXVI, determine the pressure in the vent pipe near its outlet end. The computed value of the outlet pressure should be used in subsequent steps if it exceeds atmospheric pressure. Atmospheric pressure should be used in subsequent steps if the computed value of outlet pressure is less than atmospheric.
2. Compute the friction loss in the vent pipe, working upstream, from outlet to inlet.
3. Add the friction loss to the pressure found in step (1) (atmospheric, or greater than atmospheric, as the case may be) to get the pressure at the vent inlet.
4. For a safety-valve outlet pressure equal to the vent-pipe inlet pressure found in step (3) determine from Table XXXVIII the minimum value of W/d^2 for which the safety valve would produce the required kinetic energy in the jet at the valve outlet.
5. Compare the minimum necessary value of W/d^2 with the actual W/d^2 for the valve. If the actual value is the larger of the two, the design is satisfactory. If the actual value is smaller than the tabulated minimum, it is likely that blowback will occur. The best solution for the latter case is to use a larger diameter vent pipe.

The W/d^2 values given in Table XXXVIII for steam safety-valve outlets were taken from calculated curves. Those values given in plain type are for cases in which the velocity of steam leaving the valve is determined by the pressure of the region into which the valve discharges. In such cases the valve outlet pressure is equal to that of the discharge region and the over-all efficiency of the valve considered as a nozzle is dependent upon the conditions of operation. The boldface figures correspond to values of W/d^2 large enough to produce outlet pressures greater than that of the region into which the valve discharges. In these cases it is assumed that the valve acts as a nozzle of only 30 per cent efficiency, *viz.*, that of all the energy theoretically available for producing steam velocity, only 30 per cent is effective, the remainder going into resuperheating the steam. Under these circumstances the pressure at the valve outlet does not remain constant but increases with increasing values of W/d^2. A higher

value of efficiency would result in larger velocity heads and permit the use of a smaller vent pipe. In the absence of over-all "nozzle efficiency" data for safety valves, it is believed that a conservatively low value such as 30 per cent should be used. In some designs the safety valve is a highly efficient nozzle from inlet to valve seat, but in all designs the effect of the seat and disk is to create turbulence and reheat the steam rather than to produce velocity.

TABLE XXXVIII.—MINIMUM VALUES OF W/d^2 AT SAFETY-VALVE OUTLETS REQUIRED FOR SATISFACTORY OPERATION WITH UMBRELLA-FITTED VENTS[1]

Initial steam temperature, F	Valve outlet pressure, psi gauge	Initial steam pressure, psi gauge								
		20	50	100	200	400	600	800	1,200	1,500
Saturation	0	710	960	1,140	1,340	1,450	1,550	1,600	1,830	1,780
	5	...	1,140	1,340	1,570	1,810	1,880	2,040	2,360	2,310
	10	...	.....	1,610	1,850	2,160	2,320	2,440	2,830	2,790
	20	...	.....	.....	**2,630**	2,870	3,120	3,260	3,740	3,680
	30	...	.....	.....	.....	**3,500**	3,730	3,930	4,650	4,550
	40	...	.....	.....	.....	**5,350**	**4,480**	4,700	5,460	5,350
	50	...	.....	.....	.....	.....	**5,970**	**5,460**	6,330	6,210
600	0	...	.....	890	1,000	1,160				
	5	...	.....	1,120	1,240	1,450				
	10	...	.....	1,280	1,490	1,790				
	20	...	.....	.....	**2,530**	2,360				
	30	...	.....	.....	**4,580**	3,260				
	40	...	.....	.....	.....	4,780				
700	0	...	.....	850	1,020	1,140	1,240			
	5	...	.....	1,060	1,260	1,410	1,530			
	10	...	.....	1,260	1,530	1,730	1,870			
	20	...	.....	.....	**2,380**	2,320	2,520			
	30	...	.....	.....	.....	**3,160**	3,020			
	40	...	.....	.....	.....	**4,650**	**4,100**			
800	0	...	.....	.....	1,010	1,080	1,200	1,240	1,300	
	5	...	.....	.....	1,280	1,410	1,540	1,590	1,680	
	10	...	.....	.....	1,490	1,690	1,830	1,930	2,000	
	20	...	.....	.....	**2,280**	2,230	2,420	2,540	2,700	
	30	...	.....	.....	.....	**3,030**	2,990	3,120	3,320	
	40	...	.....	.....	.....	**4,450**	**3,930**	**3,720**	3,870	
	50	...	.....	.....	.....	.....	.....	**5,000**	**4,520**	
900	0	...	.....	.....	.....	1,080	1,120	1,170	1,230	1,230
	5	...	.....	.....	.....	1,380	1,440	1,520	1,630	1,650
	10	...	.....	.....	.....	1,610	1,710	1,790	1,930	1,970
	20	...	.....	.....	.....	2,140	2,300	2,420	2,560	2,620
	30	...	.....	.....	.....	**2,910**	2,830	2,960	3,140	3,200
	40	...	.....	.....	.....	**4,320**	**3,740**	**3,560**	3,720	3,850
	50	...	.....	.....	.....	.....	**5,050**	**4,760**	**4,380**	4,380

[1] Values of W/d^2 in plain type indicate conditions in which the limiting factor is the valve outlet pressure. Boldface type indicates conditions in which the limiting factor is the assumed maximum nozzle efficiency of 30 per cent.

The following example illustrates the application of this method to the design of safety valve vents:

Example.—A safety valve with 4-in. outlet relieves steam at 400 psi gauge, 700 F, and is equipped with an umbrella-fitted vent pipe 5 in. in diameter, Schedule 40, and of 100-ft equivalent length. The valve is designed to pass 18,000 lb per hr. Is the vent line large enough to prevent blowing steam out the umbrella fitting?

Solution.—Enthalpy of steam at 400 psi gauge, 700 F = 1361.9 Btu. W is given as 18,000; Table X, page 366 gives $d^2 = 25.47$. W/d^2 for vent pipe $= \dfrac{18{,}000}{25.47} = 706$.

Interpolating from Table XXXVI to obtain the critical pressure corresponding to 1361.9 Btu initial enthalpy gives

$$p_t = \frac{W}{63.7d^2} = 706 \div 63.7 = 11.09 \text{ psi abs.}$$

Therefore the vent pipe does not operate at the acoustic velocity and the actual outlet pressure will be the same as atmospheric, *i.e.*, 14.7 psi abs or 0 psi gauge.

After estimating the specific volume of steam in the pipe, the pressure loss to the inlet of the vent pipe as calculated by Fritzsche's formula (see page 252)

$$p_\lambda = \frac{2.1082 \times 100 \times 18{,}000^{1.852} \times 26}{10^7 \times 5.047^{4.973}} = 13.2 \text{ psi.}$$

The pressure at the inlet of the pipe is, therefore,

$$13.2 + 14.7 - 14.7 = 13.2 \text{ psi gauge}$$

Table X, page 366 gives d^2 for the 4-in. valve outlet as 16.21 sq in. W/d^2 for the valve $= \dfrac{18{,}000}{16.21} = 1{,}110$.

From Table XXXVIII the minimum value of W/d^2 for 400 psi gauge, 700 F, initial conditions with 13.2 psi gauge outlet pressure is interpolated to be 1,920. Since the actual value of W/d^2 for the valve is less than that required to balance the static gauge pressure at the vent-pipe inlet, it appears that steam would blow out through the umbrella fitting. A 6-in. vent would be found satisfactory in this case.

WATER

PROPERTIES OF WATER

Composition.—Pure water is a liquid which is composed of one part by weight of hydrogen combined with eight parts of oxygen. In the formation of water 2 atoms of hydrogen are chemically combined with 1 atom of oxygen as follows:

	Molecular weight	by weight
Hydrogen, 2 atoms.	2.00	11.11
Oxygen, 1 atom...	16.00	
Molecular weight..	18.00	100.00

Water is never found pure in nature owing to the readiness with which it absorbs impurities from the air and soil. The usual impurities absorbed by water are organic matter, salts, and soluble gases. The presence of dissolved salts causes water to become "hard." A distinction is made between two kinds of hardness, *i.e.*, temporary and permanent. Temporary hardness is due to the presence of the bicarbonates of lime and magnesium which may be precipitated by boiling at 212 F, and water containing no other scale-forming elements becomes "soft" under such treatment. Permanent hardness is due mainly to the presence of sulphate of lime, which is precipitated only at temperatures above 300 F, but may be removed by chemical treatment.

The compositions of salt water, which vary in the different oceans, the Dead Sea, and Great Salt Lake, are given in Van Nostrand's "Scientific Encyclopedia."[1] These compositions may be used as bases for computing the density or specific gravity of the respective sea water. In general, the total dissolved solids in sea water are of the order of 2½ to 5 per cent, and the specific gravity usually is given as 1.025 to 1.030. In the Great Salt Lake and the Dead Sea, however, the total dissolved solids are in the range of 15 to 25 per cent by weight.

Dissolved gases can be removed by boiling, which is frequently done under vacuum, as in deaerators, etc. Organic matter can be removed by filtration, coagulation, etc. An evaporator and condenser are frequently employed to obtain pure water through the elimination of any or all of the above mentioned impurities. Where dissolved gases are among the impurities to be removed by evaporation, some method of deaeration is essential in connection with the condensing apparatus.

Boiling Temperature of Water at Various Altitudes.—The boiling point of water varies with the barometric pressure which, in turn, varies with the altitude above sea level. Assuming standard atmospheric conditions at sea level, the boiling points at various altitudes are as given in Table XXXIX.

Boiling Temperature of Water at Various Pressures.—The boiling temperature of water at various pressures is given in the steam tables on pages 230 to 243. Thus water at a nominal atmospheric pressure of 14.7 psi abs boils at 212 F.

[1] Published by D. Van Nostrand Company, Inc., 250 Fourth Ave., New York, N.Y.

Density.—Water has its greatest density at 39.1 F (4 C) when in the pure state it weighs 62.428 lb per cu ft. Above this temperature the density decreases as the temperature increases. For ordinary engineering calculations with fresh water at atmospheric temperature, the density is taken as 62.4 lb per cu ft. Table XL gives the densities, relative volumes, and pounds per gallon of water from 32 to 706 F. The relative volumes serve to indicate the volumetric expansion with change in temperature.

TABLE XXXIX.—BOILING POINT OF WATER AT VARIOUS ALTITUDES

Boiling point, degrees Fahrenheit	Altitude above sea level, feet	Atmospheric pressure, pounds per square inch	Barometer reduced to 32°F., inches	Boiling point, degrees Fahrenheit	Altitude above sea level, feet	Atmospheric pressure, pounds per square inch	Barometer reduced to 32°F., inches
184	15,221	8.20	16.70	199	6,843	11.29	22.90
185	14,649	8.38	17.96	200	6,304	11.52	23.47
186	14,075	8.57	17.45	201	5,764	11.76	23.95
187	13,498	8.76	17.83	202	5,225	12.01	24.45
188	12,934	8.95	18.22	203	4,697	12.26	24.96
189	12,367	9.14	18.61	204	4,169	12.51	25.48
190	11,799	9.34	19.02	205	3,642	12.77	26.00
191	11,243	9.54	19.43	206	3,115	13.03	26.53
192	10,685	9.74	19.85	207	2,589	13.30	27.08
193	10,127	9.95	20.27	208	2,063	13.57	27.63
194	9,579	10.17	20.71	209	1,539	13.85	28.19
195	9,031	10.39	21.15	210	1,025	14.13	28.76
196	8,481	10.61	21.60	211	512	14.41	29.33
197	7,932	10.83	22.05	212	Sea level	14.70	29.92
198	7,381	11.06	22.52				

Specific Heat.—The specific heat of water varies at different temperatures. Table XLI gives the instantaneous specific heats from 20 to 600 F. The average specific heat for the range between 32 and 212 F is unity. For rough calculations at temperatures below 250 F it is customary to consider the specific heat as one.

Compressibility.—Water at its maximum density (39.1 F) has a coefficient of compressibility, according to the Smithsonian Physical Tables, of about 0.00005 for each atmosphere of added pressure; and a modulus of elasticity in compression of about 300,000. The coefficient of compressibility decreases and the modulus increases with a rise in temperature; at 212 F they are about 0.00004 and 360,000, respectively. Thus 1 cu ft of water, which weighs 62.425 lb under 1 atm, under a pressure of 11 atm weighs 62.425 ×

(1 + 0.00005 × 10) or 62.456 lb, a very slight increase. Except where very accurate work is required or high pressures are involved, water may be considered as incompressible.

TABLE XL.—RELATIVE VOLUME AND WEIGHT PER CUBIC FOOT AND PER GALLON OF WATER AT VARIOUS TEMPERATURES AND CORRESPONDING SATURATION PRESSURES

(Based on Marks and Davis' Steam Tables)

Temperature degrees Fahrenheit	Relative volume	Weight in pounds per cubic foot	Weight in pounds per gallon	Temperature degrees Fahrenheit	Relative volume	Weight in pounds per cubic foot	Weight in pounds per gallon	Temperature degrees Fahrenheit	Relative volume	Weight in pounds per cubic foot	Weight in pounds per gallon
32	.00013	62.42	8.34	250	1.061	58.83	7.86	480	1.256	49.7	6.64
39.1	.00000	62.428	8.345	260	1.066	58.55	7.83	490	1.269	49.2	6.58
40	.00001	62.42	8.34	270	1.072	58.26	7.79	500	1.283	48.7	6.51
50	.00027	62.42	8.34	280	1.077	57.96	7.75	510	1.297	48.1	6.43
60	.00096	62.37	8.34	290	1.083	57.65	7.71	520	1.312	47.6	6.36
70	.00201	62.30	8.33	300	1.089	57.33	7.66	530	1.329	47.0	6.28
80	.00338	62.22	8.32	310	1.095	57.00	7.62	540	1.35	46.3	6.19
90	.00504	62.11	8.30	320	1.102	56.66	7.57	550	1.37	45.6	6.10
100		62.00	8.29	330	1.109	56.30	7.53	560	1.39	44.9	6.00
110	.0092	61.86	8.27	340	1.116	55.94	7.48	570	1.42	44.1	5.90
120	.0116	61.71	8.25	350	1.124	55.57	7.43	580	1.44	43.3	5.79
130	.0142	61.55	8.23	360	1.131	55.18	7.38	590	1.46	42.6	5.69
140	.0171	61.38	8.21	370	1.140	54.78	7.32	600	1.49	41.8	5.59
150	.020	61.20	8.18	380	1.148	54.36	7.27	610	1.52	41.0	5.48
160	.023	61.00	8.15	390	1.157	53.94	7.21	620	1.55	40.2	5.37
170	.027	60.80	8.13	400	1.167	53.5	7.15	630	1.59	39.4	5.27
180	.030	60.58	8.10	410	1.177	53.0	7.09	640	1.63	38.5	5.15
190	.034	60.36	8.07	420	1.187	52.6	7.03	650	1.67	37.5	5.01
200	.038	60.12	8.04	430	1.197	52.2	6.98	660	1.72	36.4	4.87
210	.043	59.88	8.00	440	1.208	51.7	6.91	670	1.78	35.2	4.71
212	.044	59.83	8.00	450	1.220	51.2	6.84	680	1.86	33.8	4.52
220	.047	59.63	7.97	460	1.232	50.7	6.78	690	1.95	32.1	4.29
230	.052	59.37	7.94	470	1.244	50.2	6.71	706.1	3.11	20.1	2.69
240	.056	59.11	7.90								

Freezing Point.—Water freezes at 32 F under ordinary atmospheric pressure, and ice melts at the same temperature. The *latent heat of fusion of ice* is 144 Btu per lb. The latent heat of fusion of a solid may be defined as the heat required in Btu to convert 1 lb of the substance from the solid to the liquid state or vice versa, without change in temperature. The *specific heat of ice* is 0.504 Btu per lb.

In freezing solid in a confined space water expands about 10 per cent in volume with an irresistible pressure which no pipe material can support without being permanently stretched or burst. The effect of increased pressure is to lower the freezing point to the

extent of 0.0126 deg F per atm. According to Bridgman,[1] ice in the temperature range 0 to +32 F has a coefficient of compressibility of about 0.000023 for each atmosphere of added pressure. This value is about one-half that given for water. The modulus of elasticity for ice in compression is approximately 400,000 psi as determined by static methods. Values three times as great have been found by dynamic methods and are regarded by physicists as being the true values.[2] However, from engineering considerations the static values seem more useful.

Absorption of Gases.—Many gases are readily absorbed by water. Other liquids also possess this power to a greater or less degree. Water will, for example, absorb its own volume of car-

TABLE XLI.—INSTANTANEOUS SPECIFIC HEAT OF WATER
(From Marks and Davis' Steam Tables)

Degrees Fahrenheit	Specific heat	Degrees Fahrenheit	Specific heat	Degrees Fahrenheit	Specific heat	Degrees Fahrenheit	Specific heat	Degrees Fahrenheit	Specific heat	Degrees Fahrenheit	Specific heat
20	1.0168	120	0.9974	220	.007		1.035		1.072		.123
30	1.0098	130	0.9979	230	.009		1.038		1.077		.128
40	1.0045	140	0.9986	240	.012		1.041		1.082		.134
50	1.0012	150	0.9994	250	.015		1.045		1.086		.140
60	0.9990	160	.0002	260	.018		1.048		.091		.146
70	0.9977	170	.0010	270	.021		1.052		.096		.152
80	0.9970	180	.0019	280	.023		1.056		.101		.158
90	0.9967	190	.0029	290	.026		1.060		.106		.165
100	0.9967	200	.0039	300	.029		1.064		.112		1.172
110	0.9970	210	.0050	310	.032		1.068		.117		

bonic acid gas, 430 times its volume of ammonia, 2⅓ times its volume of chlorine, and only about one-twentieth its volume of oxygen.

The weight of gas that is absorbed by a given volume of liquid is proportional to the pressure. But as the volume of a mass of gas is less as the pressure is greater, the volume which a given amount of liquid can absorb at a certain temperature will be constant, whatever the pressure. Water, for example, can absorb its own volume of carbonic acid gas at atmospheric pressure; it will also dissolve its own volume if the pressure is twice as great, and consequently twice the weight of gas is dissolved.

[1] P. W. Bridgman, *Proc. Am. Acad. Arts Sci.*, Vol. 47, pp. 439–558, 1912.

[2] See "Properties of Ordinary Water Substance," by N. E. Dorsey, Reinhold Publishing Corporation, 330 West 42d St., New York 18, N.Y., 1940, p. 444.

The gas-absorbing properties of water vary a great deal with the temperature. Practical use of this fact is made in the absorption ice machine. Tables showing this variation can be found in Marks's handbook.

FLOW OF WATER IN PIPES

General Principles of Flow.—The flow of fluids in general is discussed at some length on pages 81 to 137 and much of the work there is directly applicable to water. In practically all cases water may be considered as incompressible and, for calculations at approximately atmospheric temperature, it is customary to assume that it has a uniform density of 62.4 lb per cu ft which holds very nearly constant throughout the temperature range from 32 to 60 F (see Table XL, page 267). In calculations for hot-water heating systems, boiler-feed pump discharge heads, etc., it is necessary to take into account the change in density and viscosity with temperature. Application of the common empirical equations for water flow is limited to water at usual atmospheric temperatures of, say, 32 to 100 F. At higher temperatures the changes in density and viscosity have a considerable bearing on flow relations and, where close results are desired, the use of the *rational solution* (see pages 107 to 137) is recommended in preference to any of the commonly used empirical formulas.

In *ordinary hydraulic* work, y is commonly regarded as constant since the density of fresh water is approximately 62.4 lb per cu ft from 32 to 60 F. Under these conditions 62.4 can be substituted for y in equations (48), (51), and (54) on pages 85, 86 and the formulas become:

$$p_\lambda = \frac{fLv^2}{3.1d}, \qquad v = 1.76\sqrt{\frac{p_\lambda d}{fL}}.$$

$$p_\lambda = \frac{10{,}860fLF^2}{d^5}, \qquad F = 0.00958\sqrt{\frac{p_\lambda d^5}{fL}}.$$

$$p_\lambda = \frac{fLG^2}{18.6d^5}, \qquad G = 4.31\sqrt{\frac{p_\lambda d^2}{fL}}.$$

(See pages 82 to 84 for definition of symbols.)

These basic equations for hydraulic work are equivalent to what are variously known as the "common formula," the "Chézy formula," the "Darcy formula," the "Fanning formula," etc.[1]

[1] For equivalent values of f and the formula constants for use in these formulas see, "Handbook of Hydraulics," by H. W. King, McGraw-Hill Book Company, Inc., New York, 3d ed., 1939.

At the same time the factor f, as expressed above, is identical with the f used in the rational method of solution and numerical values can be determined from Figs. 15a and 15b on pages 108 and 112. For flow charts, see pages 212 to 218 and 1314 to 1316.

Before the rational method of solution was developed numerous investigators undertook modifications of the basic hydraulic formulas with a view to securing better agreement with test results throughout the usual range of operating conditions. Several of the empirical formulas commonly used in hydraulic work are given in succeeding paragraphs.

Saph and Schoder Formula.—One of the most carefully worked out empirical formulas for water flow is that of Saph and Schoder,[1] which is used extensively by mechanical engineers. The experimental work on which they based their formulas was carried out with unusual care and frequently has been quoted by other investigators and writers. Saph and Schoder gave the following values based on an analysis of their experimental data for velocities above the critical:

$$h_\lambda = \frac{0.296 \text{ to } 0.469}{d'^{1.25}} v^{1.74 \text{ to } 2.00}$$

where h_λ = head lost, ft of water per 1,000 ft of pipe.
v = velocity, ft per sec.
d' = inside diameter of pipe, ft.

The low values of the constant and exponent given above apply to smooth pipes and the high values to rough pipes. Their criterion for critical velocity was $v = 0.37/d^{0.85}$, d in this case being the inside diameter of the pipe in inches.

In a later article[2] Schoder gave the following formulas as representing average conditions for clean cast-iron and wrought-iron or wrought-steel pipe:

$$h_\lambda = \frac{0.38 v^{1.86}}{d'^{1.25}}$$

This formula is the basis for the equations and charts shown in Figs. 39a and b from which, in turn, the flow data given in Table XLII were tabulated. These charts and tables, which correspond closely to the Williams and Hazen values for "smooth new iron pipe" and for C = 130 to 140, are recommended by the authors

[1] "An Experimental Study of the Flow of Water in Pipes," *Trans. ASCE*, *Paper* 964, Vol. 51, presented at meeting of Sept. 2, 1903, by Augustus V. Saph and Ernest H. Schoder.

[2] *Eng. Record*, Sept. 3, 1904.

of "Piping Handbook" for designing aboveground service-water piping in power and industrial plants.

Where the interior surface is *decidedly rough* and for *spiral riveted* pipe, Schoder proposed $h_\lambda = 0.5v^{1.95}/d'^{1.25}$. In the case of ordinary spiral-riveted pipe, the friction loss is much greater than for cast-iron or steel pipes of the same size and diameter, because the overlapping plates and projecting rivet heads usually offer more resistance than the joints in cast-iron or steel pipe. With new spiral-*welded* pipe or new spiral-*riveted* pipe having taper joints with beveled edges, a friction loss comparable with "smooth new iron pipe" is to be expected (see values given under Scobey formulas).

Change of Interior Condition with Age.—The extent to which the interior wall surface of a pipe changes with age has been the subject of much discussion. The following statement is quoted from the "Handbook of Cast Iron Pipe," published by the Cast Iron Pipe Research Association:

> In attempting to compute the capacity of a pipe line or to figure the probable loss in head after the pipe has reached a certain age, it is absolutely essential to know something about the water to be conveyed. In most cases the quality of the water is such that the carrying capacity is affected very little by the age of the pipe. In other cases, the water may be so soft as to cause tuberculation and consequent loss in carrying capacity, or so turbid as to cause deposits of sand or mud with the same effect. Waters that cause tuberculation are the rare exception, and outside of a few raw water conduits, muddy water is also unusual.
>
> In spite of this fact, many of the books and articles on hydraulics and water supply make the bold statement that a definite correction factor must be applied to flow formulas as the pipe increases in age. It is evident that this is incorrect, since, first of all, a large number of experiments have been made that show quite definitely that in many places there is no change whatever in carrying capacity with age. Secondly, assume that a layer of tubercles 2 in. thick is produced as a result of many years' use of a pipe, it is evident that the carrying capacity of a 12-in. pipe would be considerably more reduced than would a 48-in. pipe with the same thickness of tubercles, a fact that is not taken into account in the formulas in common use.

The formulas, charts, and tables shown in Figs. 39*a* and *b* and Table XLII give conservative values for clean iron or steel pipe having a smooth interior unobstructed with rivet heads or the like. These charts furnish safe design data for such pipe under ordinary conditions of service, irrespective of the age of the pipe. Where unusual conditions are expected, other constants and exponents may be substituted in the Saph and Schoder formula as previously indicated, or one of the hydraulic formulas discussed in succeeding paragraphs may be used instead. Approximately

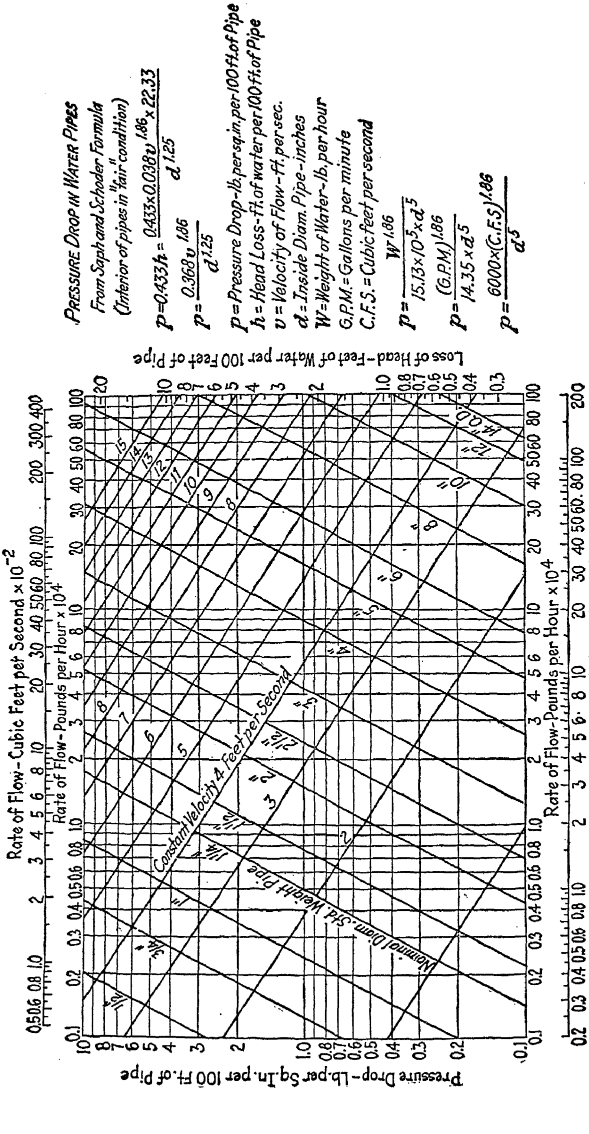


FIG. 39a.—Flow of water in pipes, ½ to 14 in. diameter.

PRESSURE DROP IN WATER PIPES

From Saph and Schoder Formula

(Interior of pipes in "fair" condition)

$$p = 0.433h = \frac{0.433 \times 0.038\, v^{1.86} \times 22.33}{d^{1.25}}$$

$$p = \frac{0.368\, v^{1.86}}{d^{1.25}}$$

p = Pressure Drop - lb. per sq. in. per 100 ft. of Pipe

h = Head Loss - ft. of water per 100 ft. of Pipe

v = Velocity of Flow - ft. per sec.

d = Inside Diam. Pipe - inches

W = Weight of Water - lb. per hour

G.P.M. = Gallons per minute

C.F.S. = Cubic feet per second

$$p = \frac{W^{1.86}}{15.13 \times 10^5 \times d^5}$$

$$p = \frac{(G.P.M.)^{1.86}}{14.35 \times d^5}$$

$$p = \frac{6000 \times (C.F.S.)^{1.86}}{d^5}$$

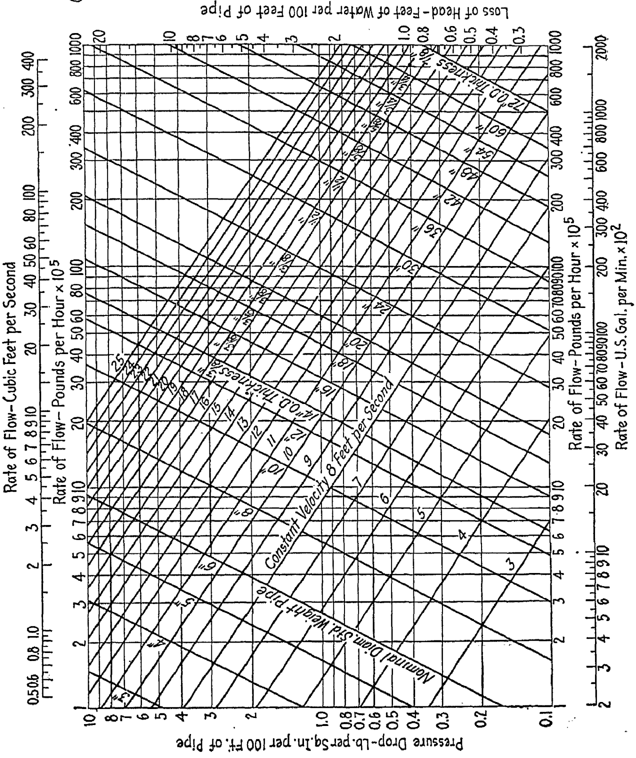


FIG. 39b.—Flow of water in pipes, 3 to 72 in. diameter.

TABLE XLII.—CARRYING CAPACITY AND FRICTION LOSS FOR STANDARD-WEIGHT WROUGHT-IRON AND STEEL WATER PIPES, ALSO APPROXIMATE FOR TYPE K COPPER WATER TUBES OF SAME NOMINAL SIZE

(Independent variables: gallons per minute and pipe size. Dependent variables: velocity, friction head, and pressure drop. Friction head and pressure drop per 100 ft of pipe, interior "fair" condition)

Gallons per minute	Velocity, feet per second	Friction head, feet	Friction loss, pounds per square inch	Velocity, feet per second	Friction head, feet	Friction loss, pounds per square inch	Velocity, feet per second	Friction head, feet	Friction loss, pounds per square inch	Velocity, feet per second	Friction head, feet	Friction loss, pounds per square inch	Velocity, feet per second	Friction head, feet	Friction loss, pounds per square inch	Velocity, feet per second	Friction head, feet	Friction loss, pounds per square inch	Velocity, feet per second	Friction head, feet	Friction loss, pounds per square inch
		¾ in.			1 in.			1¼ in.			1½ in.			2 in.			2½ in.			3 in.	
5	3.00	8.30	3.60	1.86	2.54	1.10	1.07	0.65	0.28	0.79	0.30	0.13	0.48	0.09	0.04	0.33	0.04	0.02			
10	6.00	23.7	10.3	3.71	9.00	3.90	2.14	2.31	1.00	1.57	1.06	0.46	0.96	0.30	0.13	0.67	0.12	0.05	0.43	0.04	0.02
15	9.00	62.3	27.0	5.56	18.9	8.20	3.20	4.90	2.12	2.36	2.24	0.97	1.43	0.65	0.28	1.00	0.27	0.12	0.65	0.09	0.04
20	12.0	113.0	49.0	7.42	33.2	14.4	4.27	8.44	3.65	3.14	3.81	1.65	1.91	1.08	0.47	1.34	0.46	0.20	0.87	0.16	0.07
25	15.0	166.0	72.0	9.27	49.8	21.6	5.35	12.7	5.50	3.93	5.77	2.50	2.39	1.66	0.72	1.67	0.69	0.30	1.08	0.23	0.10
30				11.1	71.5	31.0	6.42	18.0	7.80	4.72	8.08	3.50	2.87	2.31	1.00	2.01	0.97	0.42	1.30	0.32	0.14
35				13.0	95.0	41.2	7.48	24.9	10.8	5.50	10.85	4.70	3.34	3.05	1.32	2.34	1.29	0.56	1.51	0.44	0.19
		4 in.																			
40	1.01	0.14	0.06	14.9	121.0	52.6	8.56	31.7	13.7	6.30	13.85	6.00	3.82	3.92	1.70	2.68	1.66	0.72	1.73	0.55	0.24
45	1.13	0.18	0.08				9.62	39.0	16.9	7.07	17.10	7.40	4.29	4.88	2.12	3.01	2.08	0.90	1.95	0.69	0.30
50	1.26	0.21	0.09				10.7	47.4	20.5	7.86	21.0	9.10	4.77	5.95	2.58	3.35	2.49	1.08	2.16	0.85	0.37
					5 in.																
60	1.51	0.30	0.13	0.96	0.09	0.04	12.8	66.6	28.8	9.44	30.0	13.0	5.73	8.30	3.60	4.02	3.51	1.52	2.59	1.20	0.52
70	1.76	0.42	0.18	1.12	0.14	0.06	14.9	88.4	38.2	11.0	39.7	17.2	6.68	11.1	4.80	4.68	4.66	2.02	3.02	1.57	0.68
75	1.89	0.46	0.20	1.20	0.16	0.07	16.0	100.0	43.2	11.8	44.8	19.4	7.16	12.6	5.45	5.02	5.31	2.30	3.24	1.80	0.78
80	2.01	0.53	0.23	1.28	0.18	0.08				12.6	51.8	22.4	7.64	14.1	6.10	5.37	6.00	2.60	3.47	2.03	0.88
90	2.27	0.65	0.28	1.44	0.21	0.09				14.1	63.2	27.4	8.59	17.5	7.60	6.03	7.38	3.20	3.89	2.54	1.10
								6 in.													
100	2.52	0.78	0.34	1.60	0.26	0.11	1.11	0.09	0.04	15.7	76.2	33.1	9.55	21.5	9.30	6.71	9.00	3.90	4.34	3.05	1.32
125	3.14	1.20	0.52	2.00	0.39	0.17	1.39	0.16	0.07	19.6	116.2	50.4	11.9	33.2	14.4	8.37	13.8	6.00	5.42	4.62	2.00
150	3.77	1.68	0.73	2.40	0.55	0.24	1.67	0.21	0.09				14.3	46.4	20.1	10.1	19.4	8.40	6.50	6.47	2.80
175	4.42	2.26	0.98	2.80	0.74	0.32	1.94	0.30	0.13				16.7	61.5	26.7	11.7	25.6	11.1	7.58	8.76	3.80
											8 in.										
200	5.04	2.88	1.25	3.20	0.92	0.40	2.22	0.37	0.16	1.28	0.09	0.04	19.1	78.3	33.9	13.4	33.2	14.4	8.67	11.2	4.85
250	6.30	4.38	1.90	4.00	1.41	0.61	2.78	0.58	0.25	1.60	0.14	0.06	23.9	119.5	51.8	16.7	49.8	21.6	10.8	17.1	7.40
300	7.55	6.07	2.63	4.80	2.01	0.87	3.33	0.81	0.35	1.92	0.18	0.08	28.6	166.0	72.0	20.1	69.7	30.2	13.0	23.5	10.2

350	8.80	8.07	3.50	5.60	2.65	1.15	3.90	1.06	0.46	2.24	0.27	0.12				23.5	93.0	40.3	15.2	31.7	13.7
400	10.1	10.4	4.50	6.40	3.39	1.47	4.45	1.36	0.59	2.56	0.35	0.15		10 in.					17.3	40.2	17.4
450	11.3	12.9	5.60	7.20	4.27	1.85	5.00	1.71	0.74	2.88	0.42	0.18	1.83	0.14	0.06		12 in.		19.5	49.8	21.6
500	12.6	15.7	6.80	8.00	5.07	2.20	5.55	2.05	0.89	3.20	0.51	0.22	2.03	0.16	0.07	1.42	0.07	0.03	21.7	61.5	26.6
750	18.9	34.0	14.7	12.0	10.9	4.75	8.34	4.34	1.88	4.80	1.09	0.47	3.05	0.35	0.15	2.12	0.14	0.06			
1,000				16.0	18.7	8.10	11.1	7.38	3.20	6.40	1.87	0.81	4.07	0.60	0.26	2.83	0.25	0.11		14 in.	⅜ in.
		16 in.	⅜ in.																	O.D.	thick
1,250		O.D.	thick		18 in.	⅜ in.	13.9	11.1	4.80	8.00	2.81	1.22	5.07	0.90	0.39	3.53	0.37	0.16	2.90	0.23	0.10
1,500	2.63	0.16	0.07		O.D.	thick	16.7	15.7	6.80	9.60	3.92	1.70	6.10	1.25	0.54	4.25	0.51	0.22	3.50	0.32	0.14
2,000	3.51	0.28	0.12	2.75	0.14	0.06				12.8	6.69	2.90	8.13	2.17	0.94	5.65	0.90	0.39	4.65	0.55	0.24
2,500	4.39	0.42	0.18	3.43	0.23	0.10		20 in.	⅜ in.	16.0	10.1	4.40	10.2	3.23	1.40	7.05	1.34	0.58	5.82	0.89	0.36
3,000	5.27	0.58	0.25	4.12	0.30	0.13		O.D.	thick	19.2	14.1	6.10	12.2	4.62	2.00	8.50	1.89	0.82	7.00	1.15	0.50
3,500	6.15	0.76	0.33	4.81	0.42	0.18	3.86	0.23	0.10				14.2	6.24	2.70	9.93	2.52	1.09	8.15	1.55	0.67
4,000	7.03	0.97	0.42	5.49	0.53	0.23	4.41	0.30	0.13		24 in.	⅜ in.	16.3	7.99	3.46	11.3	3.23	1.40	9.31	1.99	0.86
4,500	7.91	1.22	0.53	6.18	0.65	0.28	4.96	0.37	0.16		O.D.	thick	18.3	9.93	4.30	12.8	4.04	1.75	10.5	2.47	1.07
5,000	8.78	1.48	0.64	6.86	0.81	0.35	5.51	0.46	0.20	3.78	0.18	0.08				14.2	4.92	2.13	11.6	3.00	1.30
5,500	9.66	1.78	0.77	7.55	0.95	0.41	6.06	0.55	0.24	4.16	0.21	0.09		30 in.	½ in.	15.6	5.87	2.54	12.8	3.58	1.55
6,000	10.5	2.08	0.90	8.24	1.13	0.49	6.62	0.65	0.28	4.53	0.25	0.11		O.D.	thick	17.0	6.88	2.98	14.0	4.20	1.82
6,500	11.4	2.40	1.04	8.92	1.29	0.56	7.16	0.74	0.32	4.91	0.30	0.13	3.16	0.09	0.04				15.1	4.85	2.10
7,000	12.3	2.77	1.20	9.61	1.50	0.65	7.72	0.85	0.37	5.29	0.35	0.15	3.40	0.12	0.05		36 in.	¾ in.	16.3	5.59	2.42
7,500	13.2	3.14	1.36	10.3	1.71	0.74	8.27	0.99	0.43	5.67	0.39	0.17	3.64	0.12	0.05		O.D.	thick	17.5	6.35	2.75
8,000	14.1	3.56	1.54	11.0	1.92	0.83	8.82	1.11	0.48	6.05	0.44	0.19	3.89	0.14	0.06	2.75	0.07	0.03	18.6	7.16	3.10
9,000	15.8	4.41	1.91	12.4	2.38	1.03	9.92	1.39	0.60	6.80	0.53	0.23	4.37	0.18	0.08	3.09	0.07	0.03			
10,000				13.7	2.91	1.26	11.0	1.69	0.73	7.56	0.65	0.28	4.86	0.21	0.09	3.43	0.09	0.04		42 in.	¾ in.
11,000				15.1	3.47	1.50	12.1	2.01	0.87	8.31	0.79	0.34	5.34	0.25	0.11	3.78	0.12	0.05		O.D.	thick
12,000				16.5	4.07	1.76	13.2	2.36	1.02	9.07	0.92	0.40	5.83	0.30	0.13	4.12	0.14	0.06	2.99	0.05	0.02
13,000							14.3	2.73	1.18	9.83	1.06	0.46	6.32	0.35	0.15	4.46	0.14	0.06	3.24	0.07	0.03
14,000							15.4	3.14	1.36	10.6	1.22	0.53	6.80	0.39	0.17	4.80	0.16	0.07	3.49	0.07	0.03
15,000							16.5	3.56	1.54	11.3	1.39	0.60	7.29	0.46	0.20	5.15	0.18	0.08	3.74	0.09	0.04
20,000										15.1	2.38	1.03	9.71	0.79	0.34	6.86	0.32	0.14	4.98	0.14	0.06
25,000													12.1	1.20	0.52	8.58	0.51	0.22	6.23	0.23	0.10
30,000													14.6	1.66	0.72	10.3	0.69	0.30	7.47	0.32	0.14
35,000													17.0	2.22	0.96	12.0	0.92	0.40	8.72	0.42	0.18
40,000																13.7	1.20	0.52	9.96	0.53	0.23

Friction head and pressure drop from curves based on Saph and Schoder formula: $p_\lambda = \frac{(\text{G.P.M.})^{1.86}}{14.35 \times d^5}$ in which p_λ = pressure drop, psi per 100-ft pipe and d = inside diameter of pipe, inches. Applies to pipe with interior in "fair" condition. Based on 62.428 lb per cu ft. To convert gallons per minute to cubic feet per second, divide by 448.8; to convert gallons per minute to pounds per hour, multiply by 500.7.; to convert gallons per minute to millions of gallons per day, divide by 694.4. See also Table XLVIII for conversions.

equivalent values of the roughness coefficients used in these formulas will be found in Table XLVI, page 284. Conservative design for underground water-supply mains where long life is the controlling factor is discussed on page 1026 under Loss in Capacity with Age.

Williams and Hazen Formula.—Among water-supply and hydraulic engineers the empirical formula developed by Williams and Hazen[1] in the early 1900's is extensively used for pipes 4 in. in diameter and larger. This formula is written:

$$v = 1.318CR^{0.63}s^{0.54} \quad \text{and} \quad s = \frac{3.0236v^{1.852}}{C^{1.852}d'^{1.167}},$$

from which

$$h_\lambda = \frac{3023.6v^{1.852}}{C^{1.852}d'^{1.167}} = K\frac{v^{1.852}}{d'^{1.167}}$$

Values of K corresponding to different values of C are as follows:

C	60	65	70	75	80	85	90	95	100
K	1.540	1.329	1.154	1.021	0.905	0.828	0.727	0.657	0.598

C	105	110	115	120	125	130	135	140
K	0.546	0.501	0.461	0.426	0.395	0.367	0.343	0.321

where v = mean velocity of flow, ft per sec.
C = a coefficient representing the roughness of the interior surface of the pipe.
$K = 3023.6/C^{1.852}$.
d' = inside diameter of pipe, ft.
R = mean hydraulic radius in feet, which for a circular pipe is ¼ the inside diameter d' expressed in feet.
s = hydraulic grade or slope, ft per ft of length of a pipe of uniform size.
h_λ = loss of head, ft per 1,000 ft of pipe.

Values of C recommended for use in the Williams-Hazen formula will be found in Table XLIII. The direct solution of formulas having fractional exponents requires the use of logarithms

[1] See "Hydraulic Tables," by G. S. Williams and Allen Hazen, 3d ed., John Wiley & Sons, Inc., New York, 1920. For the sake of uniformity with the Chézy formula, the expression for velocity frequently is written in the following equivalent form:

$$v = CR^{0.63}s^{0.54}0.001^{-0.04}.$$

TABLE XLIIIA.—VALUES OF C RECOMMENDED FOR USE IN THE WILLIAMS AND HAZEN FORMULA

C = 140 for "extremely smooth and straight pipes" with "continuous interior" and welded or coupled joints, such as
- New brass, copper, lead, tin.
- New cast iron.
- New welded or seamless steel.
- Smooth concrete (see Scobey concrete formula for full details on various degrees of roughness).
- Smooth cement-lined cast-iron or steel pipe.
- Cement asbestos.

C = 130 for "very smooth" pipes, such as
- Welded or seamless steel with "continuous interior" in "fair" condition.
- New welded-steel pipe with riveted girth joints.
- New cast iron, usual value.
- Old brass, copper, lead, tin.

C = 120 for "smooth" pipes, such as
- Smooth wooden pipes or wood-stave pipes.
- Ordinary concrete.

C = 110–130 for "new full-riveted" steel or wrought-iron pipe, depending on thickness of plate and extent to which rivets are countersunk (see also Scobey formula).

C = 110 for old cement-lined pipe, or vitrified-crock sewers in good condition.

C = 100 for old cast-iron or "old continuous interior" steel pipes where the carrying capacity over a long period of years is somewhat impaired through tuberculation or sedimentation. For sizes below 6 in., somewhat lower values should be used (see page 278). Velocities in feet per second and loss of head in feet per 1,000 ft of pipe for C = 100 are given in Table XLIV.

C = 95 for "old full-riveted" steel under the same conditions.

C = 90 for brick sewers.

C = 60 for "corrugated"[1] pipe or "badly tuberculated" iron or steel pipes.

[1] For flow through corrugated pipe see "Flow of Water through Culverts," by D. L. Yarnell, F. A. Nagler, and S. M. Woodward, Univ. of Iowa, Studies in Engineering, 1926, which gives results of experiments on concrete, vitrified clay, and corrugated-metal pipe culverts 12, 18, 24, and 30 in. in diameter, and on box culverts.

or a log-log slide rule. Various aids to solution are available for use with the respective formulas which are of material assistance where repeated solutions are required. In the case of the Williams-Hazen formula their own hydraulic tables can be used, or numerous charts and tables are available in texts on hydraulics and in pipe manufacturers' catalogues.[1]

TABLE XLIIIB.—APPROXIMATE VALUES OF C RECOMMENDED BY NATIONAL BOARD OF FIRE UNDERWRITERS FOR USE IN THE WILLIAMS AND HAZEN FORMULA[1]

Kind of Pipe or Hose	Value of C
Cast-iron or other iron or steel pipe with smooth interior surface:	
New pipe	120
10 years old[2]	110
15 years old	100
20 years old	90
30 years old	80
50 years old	70
75 years old	60
Riveted steel pipe, new	110
Smooth brass, copper, or lead pipe	140
Cement-lined pipe	140
Cement-asbestos pipe	140
Good quality cotton rubber-lined hose	140
Unlined linen hose	90

[1] See "Crosby-Fiske-Forster Handbook of Fire Protection," published by National Fire Protection Association, 60 Batterymarch St., Boston, Mass.

[2] The corrosive effect of various kinds of water may have much greater or less effect than indicated by these factors; the choice of factor should be guided by experience with the particular water.

The carrying capacities for different pipe sizes given in Table XLIV were compiled for a C coefficient of 100 which Williams and Hazen considered a fair value to use for general computation with cast-iron and steel pipes after some years in service. The greater degree of obstruction with pipes smaller than 6 in. in diameter, or with all sizes of pipes where the water is particularly

[1] See, for instance,

(a) "Hydro-Electric Handbook," by W. P. Creager and J. D. Justin, John Wiley & Sons, Inc., New York, 1927.

(b) "American Sewerage Practice," Vol. I, "Design of Sewers," by L. Metcalf and H. P. Eddy, McGraw-Hill Book Company, Inc., New York, 1928.

(c) Pipe Section of "General Catalog," Taylor Forge and Pipe Works, P.O. Box 485, Chicago, Ill.

conducive to corrosion, sometimes calls for using C coefficients considerably less than 100 (see pages 1024 to 1029). Under these circumstances, or where the use of a higher C coefficient is warranted, adjustment of the values of Table XLIV up or down to the desired basis is readily made through applying the factors given below the table.

Scobey Formulas.—Another formula widely used for computing flow in large *iron and steel* water conduits is that of Fred C. Scobey[1] which is written in the following forms:

$$h_\lambda = K_s \frac{v^{1.9}}{d'^{1.1}} \qquad v = \frac{h_\lambda{}^{0.53} d'^{0.58}}{K_s{}^{0.53}} \qquad F = \frac{0.78 h_\lambda{}^{0.53} d'^{2.53}}{K_s{}^{0.53}}$$

where h = feet of head lost per 1,000 ft of pipe.

K_s = a coefficient representing the roughness of the interior surface of the pipe (see pages 279, 282).

v = mean velocity of flow, ft per sec.

d' = inside diameter of pipe, ft.

F = flow, cu ft per sec.

In order to define K_s for this formula, Scobey divided iron and steel pipe into three classes in accordance with the smoothness of its interior surface as follows:

Class 1.—Full-riveted pipe, having both longitudinal and girth seams held by one or more lines of rivets with projecting heads. From a capacity standpoint, pipe with countersunk rivet heads on the interior belongs in Class 3.

Class 2.—Girth-riveted pipe, having no retarding rivet heads in the longitudinal seams but having the same girth seams as full-riveted pipe.

Class 3.—Continuous-interior pipe, having the interior unmarred by plate offsets or by projecting rivet heads in either longitudinal or girth seams. Not necessarily described as "smooth." Pipe having longitudinal-welded seams, girth-welded seams, or spiral-welded seams belongs in Class 3.

For the three foregoing classes of pipe, Scobey gave the following coefficients:

Class 1*a*.—K_s = 0.38 for new sheet-metal full-riveted pipes up to $\frac{3}{16}$ in. thickness.

[1] See "The Flow of Water in Riveted Steel and Analogous Pipes," by Fred C. Scobey, *Tech. Bull.* 150, U. S. Dept. of Agriculture. For sale by the Supt. of Documents, Washington, D.C. Contains a bibliography of 184 references on the flow of water in pipes. At the time of these investigations Mr. Scobey was principal irrigation engineer, U. S. Dept. of Agriculture.

TABLE XLIV.—CARRYING CAPACITY AND FRICTION LOSS PER 1,000 FT OF PIPE HAVING ACTUAL INSIDE DIAMETERS SHOWN, WILLIAMS AND HAZEN FORMULA FOR $C = 100$[1]

(See Table XLVII for converting gallons per minute to millions of gallons per day, cubic feet per second and thousands of pounds per hour[2])

Gpm	Velocity, ft per sec	Friction head, ft	Friction loss, psi	Velocity, ft per sec	Friction head, ft	Friction loss, psi	Velocity, ft per sec	Friction head, ft	Friction loss, psi	Velocity, ft per sec	Friction head, ft	Friction loss, psi	Velocity, ft per sec	Friction head, ft	Friction loss, psi	Velocity, ft per sec	Friction head, ft	Friction loss, psi	Velocity, ft per sec	Friction head, ft	Friction loss, psi
		3 in.			4 in.			6 in.			8 in.			10 in.			2 in.			2½ in.	
100	4.54	50	22	2.56	12	5.20	1.13	1.65	0.72	0.64	0.42	0.18	0.41	0.14	0.06	10.2	358	155	6.55	120	52
120	5.45	70	30	3.07	17	7.36	1.36	2.30	1.00	0.77	0.58	0.25	0.49	0.20	0.09	12.2	500	217	7.86	168	73
140	6.35	92	40	3.58	23	9.97	1.58	3.12	1.35	0.89	0.78	0.34	0.57	0.26	0.11	14.3	670	290	9.17	223	97
160	7.26	118	51	4.10	29	12.6	1.81	3.95	1.71	1.02	0.98	0.43	0.65	0.33	0.14	16.3	860	373	10.5	290	126
180	8.16	148	64	4.60	36	15.6	2.03	4.96	2.15	1.15	1.20	0.52	0.74	0.42	0.18	18.4	1070	464	11.7	357	155
200	9.08	178	78	5.12	44	19.1	2.26	6.00	2.60	1.28	1.50	0.65	0.82	0.51	0.22	20.4	1290	560	13.1	431	187
250	11.3	271	120	6.40	67	29	2.82	9.15	3.97	1.60	2.20	0.95	1.04	0.77	0.33	25.5	1965	840	16.4	655	284
300	13.6	380	160	7.68	93	40	3.39	12.8	5.55	1.92	3.15	1.37	1.22	1.09	0.47		12 in.		19.6	920	399
350	15.9	505	220	8.96	124	54	3.96	17.0	7.36	2.24	4.42	1.92	1.43	1.45	0.63	0.85	0.59	0.26	22.9	1220	530
400	18.2	650	280	10.2	160	69	4.52	22	9.53	2.56	5.40	2.34	1.63	1.94	0.84	1.14	0.75	0.33		14 in.	
450	20.4	805	350	11.5	198	86	5.08	28	12.1	2.88	6.70	2.90	1.84	2.30	1.00	1.28	0.93	0.40	0.94	0.45	0.20
500		16 in.		12.8	240	104	5.65	36	15.6	3.20	8.20	3.56	2.04	2.80	1.21	1.42	1.13	0.49	1.05	0.57	0.25
600	0.96	0.39	0.17	15.4	337	146	6.78	47	20	3.84	11.8	5.11	2.45	3.95	1.71	1.70	1.59	0.69	1.25	0.75	0.33
700	1.12	0.52	0.23	17.9	449	194	7.91	62	27	4.48	15.4	6.68	2.86	5.30	2.30	1.99	2.10	0.91	1.46	1.00	0.43
800	1.28	0.68	0.29		18 in.		9.04	80	35	5.12	20	8.67	3.27	6.80	2.95	2.27	2.70	1.17	1.67	1.27	0.55
900	1.44	0.84	0.36	1.13	0.48	0.21	10.2	100	43	5.75	25	10.8	3.67	8.40	3.64	2.56	3.45	1.50	1.88	1.57	0.68
1,000	1.60	1.02	0.44	1.26	0.55	0.24	11.3	118	51	6.40	30	13.0	4.08	10.2	4.42	2.84	4.05	1.76	2.09	1.90	0.82
1,200	1.92	1.45	0.63	1.51	0.79	0.34	13.5	169	73	7.68	42	18.2	4.90	14.5	6.28	3.41	5.85	2.54	2.51	2.73	1.18
1,400	2.24	1.90	0.82	1.76	1.05	0.46	15.8	222	96	8.95	55	24	5.70	19.0	8.23	3.98	7.80	3.38	2.92	3.60	1.56
1,600	2.56	2.35	1.04	2.02	1.33	0.58		20 in.		10.2	72	31	6.50	24	10.4	4.55	9.80	4.25	3.34	4.60	2.00
1,800	2.88	2.98	1.29	2.27	1.67	0.72	1.84	1.01	0.44	11.5	90	39	7.40	31	13.4	5.11	12.4	5.37	3.76	5.80	2.51
2,000	3.20	3.60	1.56	2.52	2.03	0.88	2.04	1.22	0.53	12.8	110	48	8.20	37	16.0	5.68	15.0	6.50	4.18	7.00	3.03
2,500	4.00	5.48	2.37	3.15	3.10	1.34	2.55	1.83	0.79		24 in.		10.4	57	25	7.10	23	10.0	5.23	10.4	4.51
3,000	4.80	7.70	3.34	3.78	4.30	1.86	3.06	2.57	1.11	2.13	1.07	0.46	12.3	77	33	8.52	32	14.0	6.27	15.0	6.50
3,500	5.60	10.4	4.50	4.41	5.75	2.49	3.57	3.45	1.50	2.48	1.45	0.63		30 in.		9.94	42	18.0	7.32	19.5	8.45

4,000	6.40	13.3	5.76	5.04	7.40	3.21	4.08	4.40	1.91	2.84	1.83	0.79	1.82	0.62	0.27	11.4	53	23	8.37	26	11.3
4,500	7.20	16.5	7.15	5.67	9.20	3.99	4.59	5.50	2.38	3.19	2.30	1.00	2.04	0.78	0.34		36 in.		9.40	31	13.4
5,000	8.00	20	8.66	6.30	11.3	4.90	5.10	6.70	2.91	3.55	2.76	1.20	2.27	0.93	0.40	1.58	0.38	0.16	10.5	38	16.5
6,000	9.60	28	12.1	7.56	15.5	6.71	6.12	9.40	4.07	4.26	3.90	1.69	2.72	1.30	0.56	1.90	0.53	0.23	12.5	54	23.4
7,000	11.2	37	16.0	8.82	22	9.53	7.14	11.2	4.85	4.97	5.20	2.25	3.18	1.75	0.76	2.22	0.72	0.31		42 in.	
8,000		48 in.		10.1	27	11.7	8.16	16.0	6.93	5.68	6.70	2.90	3.63	2.23	0.97	2.54	0.92	0.40	1.86	0.43	0.19
9,000	1.60	0.27	0.12	11.3	33	14.3	9.18	19.5	8.45	6.38	8.30	3.60	4.08	2.80	1.21	2.85	1.13	0.49	2.09	0.54	0.23
10,000	1.78	0.33	0.14		54 in.		10.2	24	10.4	7.10	10.0	4.33	4.54	3.50	1.52	3.17	1.38	0.60	2.32	0.64	0.28
12,000	2.14	0.47	0.20	1.68	0.27	0.12	12.2	33	14.3	8.52	14.5	6.28	5.45	4.75	2.06	3.80	2.00	0.87	2.78	0.90	0.39
14,000	2.49	0.63	0.27	1.96	0.36	0.16		60 in.		9.94	19.0	8.23	6.35	6.25	2.71	4.44	2.60	1.13	3.25	1.20	0.52
16,100	2.85	0.80	0.35	2.24	0.46	0.20	1.82	0.27	0.12	11.4	24	10.4	7.26	8.00	3.47	5.07	3.35	1.45	3.71	1.55	0.67
18,000	3.20	0.99	0.43	2.52	0.56	0.24	2.05	0.33	0.14		66 in.		8.17	9.90	4.39	5.70	4.20	1.82	4.17	1.90	0.82
20,000	3.56	1.21	0.52	2.80	0.68	0.30	2.28	0.42	0.18	1.88	0.28	0.12	9.08	12.0	5.20	6.34	5.00	2.17	4.64	2.30	1.00
25,000	4.45	1.82	0.79	3.50	1.05	0.46	2.85	0.63	0.27	2.34	0.40	0.17		72 in.		7.92	7.70	3.34	5.80	3.60	1.56
30,000	5.34	2.70	1.17	4.20	1.45	0.63	3.42	0.90	0.39	2.81	0.56	0.24	2.36	0.36	0.16	9.51	10.7	4.64	6.96	5.00	2.17
35,000	6.23	3.40	1.47	4.80	1.90	0.82	3.99	1.20	0.52	3.28	0.76	0.33	2.76	0.48	0.21		84 in.		8.12	6.70	2.91
40,000	7.12	4.50	1.95	5.60	2.60	1.13	4.56	1.53	0.66	3.75	0.99	0.43	3.15	0.59	0.26	2.32	0.28	0.12	9.28	8.40	3.64
45,000	8.01	5.50	2.38	6.30	3.20	1.39	5.13	1.83	0.79	4.22	1.20	0.52	3.55	0.76	0.33	2.61	0.36	0.16		96 in.	
50,000	8.90	6.70	2.91	7.20	4.10	1.78	5.70	2.30	1.00	4.69	1.46	0.63	3.94	0.94	0.41	2.90	0.43	0.19	2.22	0.23	0.10
60,000	10.7	9.60	4.16	8.40	5.40	2.34	6.84	3.30	1.43	5.63	2.18	0.95	4.73	1.29	0.56	3.48	0.61	0.26	2.66	0.32	0.14
70,000		108 in.		9.80	7.10	3.08	7.98	4.30	1.86	6.57	2.77	1.20	5.51	1.79	0.78	4.06	0.80	0.35	3.11	0.42	0.18
80,000	2.80	0.30	0.13		120 in.		9.11	5.50	2.38	7.50	3.50	1.52	6.30	2.30	1.00	4.64	1.04	0.45	3.55	0.54	0.23
90,000	3.15	0.37	0.16	2.56	0.23	0.10		132 in.		8.45	4.42		7.09	2.80	1.21	5.22	1.30	0.56	3.99	0.68	0.29
100,000	3.50	0.46	0.20	2.84	0.27	0.12	2.34	0.18	0.08		144 in.		7.88	3.40	1.47	5.80	1.60	0.69	4.44	0.83	0.36
120,000	4.20	0.64	0.28	3.41	0.38	0.16	2.81	0.24	0.10	2.37	0.16	0.07		156 in.		6.96	2.25	0.98	5.32	1.16	0.50
140,000	4.80	0.84	0.36	3.98	0.53	0.23	3.28	0.33	0.14	2.76	0.22	0.10	2.35	0.15	0.07		168 in.		6.21	1.53	0.66
160,000	5.60	1.10	0.48	4.55	0.66	0.28	3.75	0.42	0.18	3.15	0.27	0.12	2.69	0.19	0.08	2.32	0.13	0.06		180 in.	
180,000	6.30	1.39	0.60	5.11	0.83	0.36	4.22	0.53	0.23	3.55	0.35	0.15	3.02	0.23	0.10	2.61	0.16	0.07	2.28	0.12	0.05
200,000	7.00	1.68	0.73	5.68	1.00	0.43	4.68	0.63	0.27	3.94	0.42	0.18	3.36	0.28	0.12	2.90	0.20	0.09	2.53	0.14	0.06
250,000	8.75	2.60	1.13	7.10	1.55	0.67	5.86	0.97	0.42	4.83	0.63	0.27	4.20	0.42	0.18	3.62	0.30	0.13	3.16	0.21	0.09
300,000	10.5	3.65	1.58	8.52	2.20	0.95	7.03	1.37	0.59	5.91	0.89	0.39	5.04	0.60	0.26	4.35	0.42	0.18	3.79	0.30	0.13
350,000	12.3	4.80	2.08	9.95	2.82	1.22	8.19	1.79	0.78	6.90	1.20	0.52	5.88	0.80	0.35	5.07	0.55	0.24	4.43	0.39	0.17
400,000	14.0	6.15	2.67	11.4	3.75	1.63	9.36	2.28	0.99	7.88	1.52	0.66	6.72	1.05	0.46	5.80	0.73	0.32	5.06	0.50	0.22

[1] For other values of C, multiply the friction heads or pressure drops by the following factors:

C value	60	65	70	75	80	85	90	95	100	105	110	115	120	125	130	135	140	145
Factor	2.58	2.22	1.93	1.71	1.51	1.38	1.22	1.10	1.00	0.913	0.838	0.771	0.712	0.661	0.615	0.573	0.536	0.502

[2] To convert gallons per minute to millions of gallons per day, divide by 694.4; to cubic feet per second, divide by 448.8; to thousands of pounds per hour, multiply by 500.4.

Class 1*b*.—K_s = 0.44 for new plate-metal full-riveted pipes from $\frac{3}{16}$ to $\frac{7}{16}$ in. thickness, with either taper or cylinder joints.

Class 1*c*.—K_s = 0.48 for new plate-metal full-riveted pipes from $\frac{1}{2}$ in. thickness up with either taper or cylinder joints, and for pipes from $\frac{1}{4}$ to $\frac{7}{16}$ in. thickness when butt jointed.

Class 1*d*.—K_s = 0.52 for new butt-strap full-riveted plate-metal pipes from $\frac{1}{2}$ in. thickness up.

Class 2.—K_s = 0.34 for new girth-riveted pipes.

Class 3.—K_s = 0.32 for new continuous-interior pipes. Welded steel pipes with welded field joints or connected with bolted couplers of the Dresser type belong in this class.

Convenient tables and charts for aid in solving the Scobey formula for iron and steel pipe will be found in *Bull.* 150 and in the "Handbook of Welded Steel Pipe."[1]

Scobey's formula for *wood-stave pipe*[2] written in three different forms is

$$h_\lambda = 0.419\,\frac{v^{1.8}}{d'^{1.17}} \qquad v = 1.62d'^{0.65}h_\lambda^{0.555} \qquad F = 1.272d'^{2.65}h_\lambda^{0.555}.$$

Scobey's formula for *concrete pipe*[3] written in three different forms with the inside diameter d given in inches is

$$h_\lambda = \frac{v^2}{C_s^2 d^{1.25}} \qquad v = C_s h_\lambda^{0.5} d^{0.625} \qquad F = 0.00546C_s d^{2.625} h_\lambda^{0.5}.$$

The following values are suggested for the coefficient C_s for concrete pipe:

Class 1.—C_s = 0.267. For concrete pipes laid with a generous supply of mortar without removal of mortar squeeze. This coefficient is recommended also for pipes of Class 2 when used to convey sewage.

Class 2.—C_s = 0.310. For "dry-mix" concrete pipes and monolithic concrete pipes or tunnel linings made over rough wood forms. Also for surfaces as left by cement-gun process.

Class 3.—C_s = 0.345. For small "wet-mix" concrete pipes in short units; for "dry-mix" pipes in long units; for average monolithic pipes made on steel forms.

[1] Published by the Welded Pipe Division of the California Corrugated Culvert Co., Berkeley, Calif. See also "Handbook of Water Control," published by the same company.

[2] See "The Flow of Water in Wood Stave Pipe," by Fred C. Scobey, *Bull.* 376, U. S. Dept. of Agriculture, 1916, rev., 1926.

[3] See "The Flow of Water in Concrete Pipe," by Fred C. Scobey, *Bull.* 852, U. S. Dept. of Agriculture, 1920. See also "Concrete Pipe Lines," by M. W. Loving, published 1942 by the American Concrete Pipe Association, Chicago.

Class 4.—$C_s = 0.370$. For glazed-interior concrete pipe lines; for large cement-lined iron pipes; for monolithic pipe lines where joint scars and all interior surface irregularities are removed. Particularly applicable to jointed lines of units made from wet, well-spaded concrete, deposited against oiled-steel forms.

Kutter and Manning Formulas.—These formulas, published in 1869 and 1890, respectively, are used extensively by civil engineers

TABLE XLV.—HORTON'S VALUES OF n FOR USE WITH KUTTER AND MANNING FORMULAS

Surface	Best	Good	Fair	Bad
Uncoated cast-iron pipe	0.012	0.013	0.014	0.015
Coated cast-iron pipe	0.011	0.012[1]	0.013[1]	
Commercial wrought-iron pipe, black	0.012	0.013	0.014	0.015
Commercial wrought-iron pipe, galvanized	0.013	0.014	0.015	0.017
Smooth brass and glass pipe	0.009	0.010	0.011	0.013
Smooth lockbar and welded "OD" pipe	0.010	0.011[1]	0.013[1]	
Spiral riveted steel pipe	0.013	0.015[1]	0.017[1]	
Corrugated steel pipe[2]		0.021[1]		
Vitrified sewer pipe	0.011	0.013[1]	0.015	0.017
Common clay drainage tile	0.011	0.012[1]	0.014[1]	0.017
Glazed brickwork	0.011	0.012	0.013[1]	0.015
Brick in cement mortar; brick sewers	0.012	0.013	0.015[1]	0.017
Neat cement surfaces	0.010	0.011	0.012	0.013
Cement mortar surfaces	0.011	0.012	0.013[1]	0.015
Concrete pipe	0.012	0.013	0.015[1]	0.016
Wood-stave pipe	0.010	0.011	0.012	0.013
Plank flumes:				
Planed	0.010	0.012[1]	0.013	0.014
Unplaned	0.011	0.013[1]	0.014	0.015
With battens	0.012	0.015[1]	0.016	
Concrete-lined channels	0.012	0.014[1]	0.016[1]	0.018
Cement-rubble surface	0.017	0.020	0.025	0.030
Dry-rubble surface	0.025	0.030	0.033	0.035
Dressed-ashlar surface	0.013	0.014	0.015	0.017
Semicircular metal flumes, smooth	0.011	0.012	0.013	0.015
Semicircular metal flumes, corrugated	0.0225	0.025	0.0275	0.030
Canals and ditches:				
Earth, straight and uniform	0.017	0.020	0.0225[1]	0.025
Rock cuts, smooth and uniform	0.025	0.030	0.033[1]	0.035
Rock cuts, jagged and irregular	0.035	0.040	0.045	
Winding sluggish canals	0.0225	0.025[1]	0.0275	0.030
Dredged earth channels	0.025	0.0275[1]	0.030	0.033
Canals with rough stony beds, weeds on earth banks	0.025	0.030	0.035[1]	0.040
Earth bottom, rubble sides	0.028	0.030[1]	0.033[1]	0.035
Natural stream channels:				
(1) Clean, straight bank, full stage, no rifts or deep pools	0.025	0.0275	0.030	0.033
(2) Same as (1), but some weeds and stones	0.030	0.033	0.035	0.040
(3) Winding, some pools and shoals, clean	0.033	0.035	0.040	0.045
(4) Same as (3), lower stages, more ineffective slope and sections	0.040	0.045	0.050	0.055
(5) Same as (3), some weeds and stones	0.035	0.040	0.045	0.050
(6) Same as (4), stony sections	0.045	0.050	0.055	0.060
(7) Sluggish river reaches, rather weedy or with very deep pools	0.050	0.060	0.070	0.080
(8) Very weedy reaches	0.075	0.100	0.125	0.150

[1] Values commonly used in designing.
[2] Added from other sources.

for gravity-flow design in both open channels and buried conduits carrying either water or sewage, particularly where the conduit runs only partly full.[1] The numerical constant $1.486/n$ in the Manning formulas is designed to suit the corresponding values of Kutter's n. Owing to the involved nature of Kutter's formula and the fact that equivalent results can be obtained from Manning's formula,[2] the latter only is reproduced here. It is suggested that those having occasion to use Kutter's formula refer to the logarithmic flow charts prepared by Prof. John H. Gregory.[3]

TABLE XLVI.—APPROXIMATELY EQUIVALENT VALUES OF ROUGHNESS COEFFICIENTS USED IN VARIOUS HYDRAULIC FLOW FORMULAS[1]

Williams and Hazen, C	Scobey		Manning and Kutter, n
	Steel, K_s	Concrete, C_s	
140	0.30	0.40	0.010
130	0.33	0.38	0.011
120	0.38	0.35	0.012
110	0.44	0.33	0.013
100	0.52	0.30	0.014
90	0.64	0.28	0.015
80	0.80	0.25	0.016
70	0.96	0.20	0.017

[1] For average conditions of about 5 ft per sec velocity and pipe diameters of 12 to 48 in. For equivalent values for individual pipe size and velocities see Table 13, pp. 98–99 of Scobey *Bull.* 150.

Using the same terms as in the preceding formulas (see page 82 for definitions of symbols), the Manning formula can be written:

$$v = \frac{1.486}{n} R^{2/3} h_\lambda^{1/2} \quad \text{and} \quad h_\lambda = \frac{n^2 v^2}{2.208 R^{4/3}},$$

where the hydraulic radius $R = \dfrac{\text{cross-sectional area in sq ft}}{\text{wetted perimeter in ft}}$.

[1] See "Determination of Kutter's n for Sewers Partly Filled," by C. Frank Johnson, *Proc. ASCE*, February, 1943, Vol. 69, with discussion April, 1943.

[2] See "Handbook of Hydraulics," by H. W. King, *op. cit.* This text contains extensive Manning formula tables which are supplemented by others published in "Hydraulic Tables," of U. S. War Dept., Corps of Engineers (copies of the latter can be obtained from Supt. of Documents, Washington, D.C.). King shows that the maximum carrying capacity of a circular conduit exists when the depth of water in the conduit is 0.938 of its inside diameter.

[3] Reproduced in "American Sewage Practice," Vol. I, "Design of Sewers," *op. cit.*

TABLE XLVII.—CONVERSIONS FROM GALLONS PER MINUTE TO MILLIONS OF GALLONS PER DAY, CUBIC FEET PER SECOND, AND THOUSANDS OF POUNDS PER HOUR AT A DENSITY OF 8.34 LB PER GAL (7.48 GAL PER CU FT)[1]

Gal per min	Millions of gal per day	Cu ft per sec	Thousands of lb per hr	Gal per min	Millions of gal per day	Cu ft per sec	Millions of lb per hr
100	0.144	0.223	50.04	10,000	14.40	22.28	5.004
120	0.173	0.267	60.05	12,000	17.28	26.74	6.005
140	0.202	0.312	70.06	14,000	20.16	31.19	7.006
160	0.230	0.356	80.06	16,000	23.04	35.65	8.006
180	0.259	0.401	90.07	18,000	25.92	40.10	9.007
200	0.288	0.446	100.1	20,000	28.80	44.56	10.01
250	0.360	0.557	125.1	25,000	36.00	55.70	12.51
300	0.432	0.668	150.1	30,000	43.20	66.84	15.01
350	0.504	0.780	175.1	35,000	50.40	77.98	17.51
400	0.576	0.891	200.2	40,000	57.60	89.12	20.02
450	0.648	1.003	225.2	45,000	64.80	100.3	22.52
500	0.720	1.114	250.2	50,000	72.00	111.4	25.02
600	0.864	1.337	300.2	60,000	86.40	133.7	30.02
700	1.008	1.560	350.3	70,000	100.8	156.0	35.03
800	1.152	1.762	400.3	80,000	115.2	178.2	40.03
900	1.296	2.005	450.4	90,000	129.6	200.5	45.04
1,000	1.440	2.228	500.4	100,000	144.0	222.8	50.04
1,200	1.728	2.674	600.5	120,000	172.8	267.4	60.05
1,400	2.016	3.119	700.6	140,000	201.6	311.9	70.06
1,600	2.304	3.564	800.6	160,000	230.4	356.4	80.06
1,800	2.592	4.010	900.7	180,000	259.2	401.0	90.07
2,000	2.880	4.456	1,001	200,000	288.0	445.6	100.1
2,500	3.600	5.570	1,251	250,000	360.0	557.0	125.1
3,000	4.320	6.684	1,501	300,000	432.0	668.4	150.1
3,500	5.040	7.798	1,751	350,000	504.0	779.8	175.1
4,000	5.760	8.912	2,002	400,000	576.0	891.2	200.2
4,500	6.480	10.03	2,252	450,000	648.0	100.3	225.2
5,000	7.200	11.14	2,502	500,000	720.0	1,114	250.2
6,000	8.640	13.37	3,002	600,000	864.0	1,337	300.2
7,000	10.08	15.60	3,503	700,000	1,008	1,560	350.3
8,000	11.52	17.82	4,003	800,000	1,152	1,782	400.3
9,000	12.96	20.05	4,504	900,000	1,296	2,005	450.4

[1] For conversions to miner's inches and acre-feet or acre-inches, see Table XIII, p. 1085.

And for round pipes running full $R = d'/4$, from which

$$v = \frac{0.589 d'^{2/3} h_\lambda^{1/2}}{n} \quad \text{and} \quad h_\lambda = \frac{n^2 v^2}{0.347 d'^{4/3}}.$$

Horton's[1] values of n shown in Table XLV are generally used with the Kutter and Manning formulas. According to Metcalf and Eddy, however, the value of $n = 0.013$ for vitrified crock

[1] "Some Better Kutter's Flow Coefficients," by R. E. Horton, *Eng. News*, February, 24, 1916; May 4, 1916.

TABLE XLVIII.—RELATIVE CARRYING CAPACITIES OF STANDARD-WEIGHT[2] AND OUTSIDE DIAMETER WROUGHT PIPE FOR WATER[1]

Actual inside diameter, inches	0.269	0.364	0.493	0.622	0.824	1.049	1.380	1.610	2.067	2.469	3.068	3.548
Nominal size	1/8	1/4	3/8	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2
1/8	1	0.475	0.222	0.122	0.0625	0.0333	0.0167	0.0114	0.0061	0.00392	0.00228	0.00158
1/4	2.1	1	0.475	0.263	0.130	0.0714	0.0357	0.0244	0.0130	0.00833	0.00485	0.00336
3/8	4.5	2.1	1	0.555	0.278	0.151	0.077	0.0526	0.0278	0.0178	0.0103	0.0072
1/2	8.2	3.8	1.8	1	0.500	0.270	0.139	0.091	0.050	0.0322	0.0185	0.0128
3/4	16	7.7	3.6	2	1	0.555	0.278	0.189	0.100	0.0645	0.0370	0.0263
1	30	14	6.6	3.7	1.8	1	0.500	0.344	0.182	0.1180	0.0666	0.0476
1 1/4	60	28	13	7.2	3.6	2	1	0.666	0.357	0.232	0.143	0.0910
1 1/2	88	41	19	11	5.3	2.9	1.5	1	0.526	0.345	0.200	0.139
2	164	77	36	20	10	5.5	2.8	1.9	1	0.625	0.370	0.256
2 1/2	255	120	56	31	15.5	8.5	4.3	2.9	1.6	1	0.588	0.400
3	439	206	97	54	27	15	7	5	2.7	1.7	1	0.715
3 1/2	632	297	139	78	38	21	11	7.2	3.9	2.5	1.4	1
4	867	407	191	107	53	29	15	9.9	5.3	3.4	2	1.4
5	1,525	716	335	188	93	51	26	17	9.3	6	3.5	2.4
6	2,414	1,133	531	297	147	80	41	28	15	9.5	5.5	3.8
8	4,795	2,251	1,054	590	292	160	80	54	29	19	10.9	7.6
10	8,468	3,976	1,862	1,042	516	282	142	97	52	33	19	13.4
12	13,292	6,240	2,923	1,635	809	443	223	152	81	52	30	21
14 in. O.D.	17,028	7,994	3,745	2,094	1,037	567	286	194	104	67	39	27
16 in. O.D.	24,199	11,361	5,322	2,976	1,474	806	406	276	148	95	55	38
18 in. O.D.	31,750	14,906	6,982	3,905	1,933	1,057	533	362	194	124	72	50
20 in. O.D.	41,928	19,685	9,221	5,157	2,553	1,396	703	478	256	164	95	66
24 in. O.D.	67,599	31,737	14,866	8,315	4,116	2,251	1,134	771	413	265	154	107

TABLE XLVIII.—(*Concluded*)

Actual inside diameter inches	4.026	5.047	6.065	7.981	10.02	12.00	13.25	15.25	17.00	19.00	23.00
Nominal size	4	5	6	8	10	12	14 in. O.D.	16 in. O.D.	18 in. O.D.	20 in. O.D.	24 in. O.D.
1/8	0.00115	0.000655	0.000415	0.000209	0.000118	0.000075	0.000059	0.000041	0.000032	0.000024	0.000015
1/4	0.00246	0.00139	0.000882	0.000445	0.000252	0.000160	0.000125	0.000088	0.000067	0.000051	0.0000032
3/8	0.00523	0.00298	0.00188	0.000948	0.000536	0.000342	0.000267	0.000188	0.000143	0.000108	0.00067
1/2	0.00934	0.00532	0.00336	0.00169	0.000959	0.000611	0.000477	0.000336	0.000256	0.000193	0.000120
3/4	0.0189	0.01075	0.00680	0.00342	0.00194	0.00124	0.000965	0.000678	0.000517	0.000392	0.000243
1	0.0345	0.0196	0.0125	0.00625	0.00355	0.00226	0.00176	0.00124	0.000945	0.000716	0.000444
1¼	0.0666	0.0384	0.0244	0.0125	0.00705	0.00449	0.00350	0.00246	0.00188	0.00142	0.000881
1½	0.1010	0.0588	0.0357	0.0185	0.0103	0.00657	0.00515	0.00362	0.00276	0.00209	0.001300
2	0.1887	0.1075	0.0666	0.0345	0.0192	0.0123	0.00961	0.0675	0.00515	0.00390	0.00242
2½	0.294	0.1665	0.1050	0.0526	0.0303	0.0192	0.0149	0.0105	0.00805	0.00610	0.00377
3	0.500	0.296	0.182	0.0917	0.0526	0.0333	0.0256	0.0182	0.0139	0.01050	0.00650
3½	0.715	0.417	0.263	0.1315	0.0746	0.0476	0.0370	0.0263	0.0200	0.01515	0.00935
4	1	0.555	0.357	0.1820	0.1020	0.0666	0.0500	0.0357	0.0270	0.0208	0.0128
5	1.8	1	0.625	0.322	0.1785	0.1150	0.0909	0.0625	0.0476	0.0370	0.0227
6	2.8	1.6	1	0.500	0.286	0.1820	0.1430	0.1000	0.0769	0.0555	0.0357
8	5.5	3.1	2	1	0.555	0.3570	0.278	0.2000	0.1515	0.1150	0.0714
10	9.8	5.6	3.5	1.8	1	0.6250	0.500	0.3450	0.2630	0.2000	0.1250
12	15	8.7	5.5	2.8	1.6	1	0.769	0.555	0.416	0.322	0.196
14 in. O.D.	20	11	7	3.6	2	1.3	1	0.714	0.526	0.400	0.250
16 in. O.D.	28	16	10	5.0	2.9	1.8	1.4	1	0.769	0.588	0.357
18 in. O.D.	37	21	13	6.6	3.8	2.4	1.9	1.3	1	0.769	0.476
20 in. O.D.	48	27	18	8.7	5	3.1	2.5	1.7	1.3	1	0.625
24 in. O.D.	78	44	28	14	8	5.1	4	2.8	2.1	1.6	1

[1] (See p. 288.) The carrying capacity for water varies as the square root of the fifth power of the actual inside diameter. The relative carrying capacity for air, steam, and gas is given in Table XIII on p. 89.

[2] Schedule 40 pipe 10 in. and smaller is identical with standard-weight pipe. The relative carrying capacities listed for standard-weight pipe also are approximate for Type K copper water tubes of same nominal size.

Example.—A 2-in. pipe has a water-carrying capacity equal to ten ¾-in. pipes, or 0.1887 that of a 4-in. pipe.

sewers should be increased to $n = 0.015$ to provide a margin for losses in manholes, branches, and changes in direction, and for possible retardation by air currents in small sewers only partly filled.

Comparison of Hydraulic Coefficients.—Approximately equivalent values of the coefficients used in the foregoing hydraulic flow formulas are given in Table XLVI. This comparison will be found useful in converting the coefficients for interior roughness stated for any one of these formulas into equivalent values usable in the others. The average conversions probably are as accurate as are warranted where the roughness of the interior surface is as indeterminate over a period of years as usually is the case, especially with iron or steel pipes.

Resistance Due to Fittings, Valves, etc.—The equivalent resistance of fittings and valves in terms of pipe length is given in Table XIV on page 100, with discussion on pages 95 to 102. Further information applicable to underground water-supply systems will be found on pages 1029 to 1082, and to hot-water heating systems on page 964. Entrance and exit losses will be found on pages 53 and 72; losses due to sudden enlargement and contraction on pages 102, 103; and the effect of nozzles and orifices on pages 54 to 63.

Relative Carrying Capacity of Water Pipes.—A derivation of the formula for determining the relative carrying capacity of different diameter pipes having the same length is given on page 87. The generally accepted relation for water pipes is that the carrying capacity varies as the square root of the fifth power of the inside diameter. The relative carrying capacities of standard-weight wrought water pipes as given by the above-mentioned formula are shown in Table XLVIII. The decimal fractions in the upper right-hand portion of the table are the reciprocals of the corresponding values in the lower left-hand portion. Through the use of this table, rather involved problems in carrying capacity can be worked out readily, as shown by the following examples:

Example 1.—What size water main has a carrying capacity equal to 80 1-in. standard-weight pipes? Referring to Table XLVIII, go down the 1-in. pipe column to 80, and opposite 80 read the corresponding pipe size as 6 in.

Example 2.—An 8-in. main supplies four 2-in. pipes and one 3-in. pipe. How many 4-in. pipes can be added without exceeding the relative carrying capacity of the 8-in. main? The fractional carrying capacities of the small pipes in terms of an 8-in. main, as taken from Table XLVIII, add up as follows:

Four 2-in. pipes	$4 \times 0.0345 = 0.1380$
One 3-in. pipe	$1 \times 0.0917 = \underline{0.0917}$
	0.2297

Subtracting 0.2297 from unity gives 0.7703 of the total capacity of the 8-in. main still available to supply 4-in. pipes. From Table XLVIII one 4-in. pipe has 0.1818 of the carrying capacity of an 8-in. pipe. Therefore, $0.7703 \div 0.1818 = 4.25$ 4-in. pipes, which could be added.

The answer, therefore, is four.

Example 3.—A building is supplied by three water service pipes: one 2-in., one 1½-in., and one 1¼-in. It is desired to add additional equipment requiring the capacity of a 1-in. pipe and at the same time replace the various water-supply pipes with a single service. What size pipe will have the same relative carrying capacity as the four mentioned above? Express the carrying capacity of each pipe in terms of the smallest by reference to Table XLVIII.

One 2-in. pipe	=	5.5 1-in. pipes.
One 1½-in. pipe	=	2.9 1-in. pipes.
One 1¼-in. pipe	=	2.0 1-in. pipes.
One 1-in. pipe	=	1.0 1-in. pipe.
Total capacity	=	11.4 1-in. pipes.

From Table XLVIII a 2½-in. pipe has the carrying capacity of eight 1-in. pipes, while a 3-in. has the capacity of fifteen 1-in. pipes. It will be necessary, therefore, to use a 3-in. pipe.

Flow through Complex Pipe Lines.—A convenient method of calculating flow relations in complex pipe lines is given on page 90 in conjunction with the general subject of "Friction Loss in Pipes." In that connection are considered complex-series pipe lines, divided circuits or loops, multiple-series pipe lines, and distribution networks. An easy and direct method is described there which was worked out originally for calculating the flow of natural gas in complex lines. References are cited there for computing the flow of water and other fluids in distribution networks.

Water Horsepower.—By definition a horsepower is the ability to do work at the rate of 33,000 ft-lb per min. The total horsepower developed by water in falling from a given height is the product of the weight of flow W in pounds per minute times the height h in feet divided by 33,000, or water horsepower $= Wh/33{,}000$. The actual horsepower available for doing useful work is less than the total by the amount consumed in friction and lost in velocity head. If the water is used to run a *water turbine or other engine*, the shaft horsepower is further diminished by the mechanical losses in the turbine. If h represents the total elevation head in feet, h_λ the head lost in friction and velocity head at exit, and E the mechanical efficiency of the water turbine, the shaft horsepower developed is

$$\text{Shaft horsepower} = \frac{EW(h - h_\lambda)}{33{,}000}.$$

The water flow to turbines is commonly measured in cubic feet per second. At 50 F water weighs 62.4 lb per cu ft.[1] If Q represents water flow in cubic feet per second, $W = 60Q \times 62.4$ and the above formulas may be written: water horsepower $= (60Q \times 62.4h) \div 33{,}000 = (Qh) \div 8.81$; and shaft horsepower $= EQ(h - h_\lambda) \div 8.81$.

Horsepower Required to Pump Water.—In a similar way the water horsepower required to pump water against a total head h is water horsepower $= Wh/33{,}000$. If E is the mechanical efficiency of the pump, the shaft horsepower $= Wh/(E \times 33{,}000)$, which is also the horsepower output of the motor or other engine driving the pump. If E' represents the efficiency of the driving engine, the input of the latter is line horsepower $= Wh/(EE' \times 33{,}000)$. If the input of a motor is desired in kilowatts, it can be obtained readily from the relation 1 hp = 0.746 kw. The electrical input of the motor then is

$$\text{Line kw} = \frac{0.746Wh}{EE' \times 33{,}000} = \frac{Wh}{EE' \times 44{,}230}.$$

In case it is desired to solve pumping problems in terms of gallons per minute rather than weight, the above formulas can be adapted readily from the relation that 1 gal of water weighs 8.34 lb at 50 F, hence $W = 8.34 \times \text{gpm}$, and the above formulas can be rewritten: water horsepower $= (8.34 \times \text{gpm} \times h) \div 33{,}000 = (\text{gpm} \times h) \div 3{,}957 = 0.000252\ \text{gpm} \times h$; shaft horsepower $= (8.34 \times \text{gpm} \times h) \div (E \times 33{,}000) = (\text{gpm} \times h) \div (E \times 3{,}957)$.

For the number of foot-pounds of work done per horsepower-hour or per kilowatt-hour at 100 per cent efficiency, see page 8. Due account must be taken of the pump-and-motor or the engine efficiency in converting work in foot-pounds into horsepower-hours or kilowatt-hours.

Time Required to Empty Tanks.—An open tank whose cross-sectional area is uniform throughout its depth (*i.e.*, a vertical cylindrical or prismatic tank) will empty itself through an orifice or a pipe (the discharge point being at the same level as the bottom of the tank), if there is no inflow, in just twice the time required to discharge the same amount of liquid under the initial head. For other shapes of reservoirs, the time may be computed by dividing the volume into several layers and computing the time for each

[1] For problems involving water at any other temperatures, substitute the density or weight per gallon corresponding to that temperature, as given in Table XL, p. 267, in these formulas.

layer to discharge under the average head for that layer. In case the outlet of the pipe is located some distance below the bottom of the tank, the time required will be that corresponding to the average head at the outlet during the process of emptying.

WATER HAMMER

Occurrence of Water Hammer.—If the velocity of water or other liquid flowing in a pipe is suddenly diminished, the energy given up by the liquid will be divided between compressing the liquid itself, stretching the pipe walls, and frictional resistance to wave propagation. Water hammer is manifest as a series of shocks, sounding like hammer blows, which may have sufficient magnitude to rupture the pipe or damage connected equipment. It may be caused by the nearly instantaneous or too rapid closing of a valve in the line, or by an equivalent stoppage of flow such as would take place with the sudden failure of electricity supply to a motor-driven pump. The shock pressure is not concentrated at the valve, and if rupture occurs it may take place near the valve simply because it acts there first. The pressure wave due to water hammer travels back upstream to the inlet end of the pipe, where it reverses and surges back and forth through the pipe, getting weaker on each successive reversal. The velocity of the wave is that of an acoustic wave in an elastic medium, the elasticity of the medium in this case being a compromise between that of the liquid and the pipe. The excess pressure due to water hammer is additive to the normal hydrostatic pressure in the pipe, and depends on the elastic properties of the liquid and pipe, and on the magnitude and rapidity of change in velocity. Complete stoppage of flow is not necessary to produce water hammer, as any sudden change in velocity will create it to a greater or lesser degree, depending on the conditions mentioned above.

The phenomena of water hammer have been analyzed by a number of investigators.[1] In stating the formulas for velocity

[1] The following references are listed from the large number available:

(*a*) "Theory of Water Hammer," by Lorenzo Allievi, translated by Eugene E. Halmos, Topography Riccardo Garroni, Rome, Italy, 1925.

(*b*) "Hydraulics and Its Applications," by A. H. Gibson, D. Van Nostrand Company, Inc., New York, 1923, p. 222.

(*c*) "Hydraulics," by R. L. Daugherty, McGraw-Hill Book Company, Inc., 1937, p. 300.

(*d*) "Hydraulics of Pipe Lines," by W. F. Durand, D. Van Nostrand Company, Inc., New York, 1921, p. 84.

(*e*) "Symposium on Water Hammer," ASME Hydraulic and ASCE Power

of wave travel, pressure rise, etc., the following symbols and definitions of units have been chosen as best agreeing with other portions of this text. While this does not correspond exactly with the statement of similar formulas in certain of the references listed below, the principles are the same and the results in general are identical.

LIST OF SYMBOLS USED IN CONNECTION WITH WATER HAMMER

E = modulus of elasticity in tension for pipe material, psi.
g = acceleration due to gravity = 32.2 ft per sec^2.
K = bulk modulus of elasticity of liquid, psi (approximately 300,000).
K' = virtual modulus of elasticity for liquid and pipe combination, psi.
l = length of pipe from valve to inlet end, ft.
λ = Poisson's ratio (coefficient of lateral contraction) for pipe material (see Table XLIX, page 296, and Table I, page 344).
p = normal or initial pressure of liquid in pipe, psi.
p_0 = excess pressure due to water hammer, psi.
r = inside radius of pipe, in.
S = velocity of wave travel, ft per sec, in rigid pipe.
S' = velocity of wave travel, ft per sec, in elastic pipe.
t = thickness of pipe wall, in.
T = time, sec.
v = initial velocity in pipe, ft per sec.
v_0 = reduction in velocity causing water hammer, ft per sec.
y = density of liquid, lb per cu ft.

Wave Propagation in a Rigid Pipe.—If a column of liquid flowing at a steady velocity v through a *rigid* pipe of uniform diameter and of length l feet has its motion instantaneously retarded by the partial or complete closure of a rigid valve (or equivalent stoppage), the phenomena experienced are due to the elasticity of the column alone and are analogous to those obtaining in the case of the longitudinal impact of an elastic bar against a rigid wall.

Division, 1933 and 1937. Papers of the 1933 symposium were published in a separate bulletin for sale by the American Society of Mechanical Engineers, 29 West 39th St., New York 18, N.Y. Papers of the 1937 symposium were published in the *Trans. ASME*, November, 1937, Vol. 59, No. 8.

(*f*) "Water Hammer," by Miss O. Simin, *Proc. AWWA*, Vol. 24, 1904, pp. 335–424. Summary of work in Moscow by Prof. N. Joukowsky.

For other references see pp. 1031 to 1038 and 1317 to 1322.

At the instant of closure, the velocity of the layer of water in contact with the valve is retarded by an amount designated as v_0, and the kinetic energy corresponding to $v_0^2/2g$ is converted into resilience or energy of strain, with a consequent sudden rise in pressure. This checks the adjacent layer, with the result that a state of reduced velocity and increased pressure (this at any point being designated as p_0 above the pressure p obtaining at that point with steady flow) is propagated as a wave along the pipe with velocity S. Assuming instantaneous retardation, a *rigid* pipe, and *complete reflection* of the wave at the valve, which represent worst conditions of water hammer, the velocity of wave travel is $S = 12\sqrt{gK/y}$. (This is the well-known formula of physics for the velocity of an acoustic wave in an elastic medium.) This wave reaches the inlet end of the pipe after T seconds, where $T = l \div S$. At this instant the column of liquid has been retarded to its new uniform velocity of $v - v_0$ and is in a state of maximum compression of $p + p_0$ where $p_0 = (v_0/12)\sqrt{yK/g} = ySv_0/144g$. This is derived from the relation that if a cube of unit side be subjected to a pressure increasing from p to $p + p_0$, the change in volume will be $p_0 \div K$, and since the mean increase in pressure during compression is $p_0 \div 2$, the work done in inch-pounds is $p_0^2 \div 2K$. Hence, equating the change in energy $yv_0^2/(2g \times 144) = p_0^2/2K$ and $p_0 = (v_0/12)\sqrt{yK/g}$. The second expression for p_0 is obtained directly by substituting the value of K obtained from the preceding equation.

This is not a state of equilibrium, since the pressure $p + p_0$ existing inside the pipe is greater than the applied pressure p from the external medium. In consequence, the strain energy due to compression is reconverted into kinetic energy, the internal pressure falls to that of the external source, and the liquid rebounds with a velocity almost equal to $v - 2v_0$ in a direction opposite to that of initial flow. At this instant the whole of the column is at normal pressure p and is moving with a velocity of approximately $v - 2v_0$ toward the inlet end of the pipe. The end of the column tends to leave the valve but cannot do so without causing a reduction in pressure at that point which in turn acts to retard the reversed flow. Motion is consequently checked and the kinetic energy of reversed flow goes to reduce the strain energy to a value below that corresponding to normal pressure. The pressure drops suddenly by an amount equal to that through which it originally rose, and a wave of zero velocity and of pressure

p_0 below normal is transmitted along the pipe, to be reflected from the inlet end as a wave of normal pressure and velocity toward the valve. When this wave reaches the valve $4l \div S$ seconds after the instant of closure, the conditions are the same as at the beginning of the cycle, and the whole is repeated. This cycle would continue indefinitely were it not for viscosity of the liquid and friction against the pipe walls. The completeness of wave reflection at the valve and at the inlet end of the pipe will vary in different cases—the above explanation applies to worst conditions, consisting of complete reflection at both ends, a rigid pipe, and instantaneous reduction in velocity at the valve. The condition of instantaneous closure applies, provided the time of closure (whether partial or complete) does not exceed that required for the wave to travel from the valve to the inlet end of the pipe and back again, *i.e.*, provided the time of closure does not exceed $(2l \div S)$. It is apparent, therefore, that if severe water hammer is to be avoided, the time of closure should exceed the time corresponding to $(2l \div S)$, and the longer the time taken in closure, the less will be the shock. The excess pressure resulting from water hammer can be kept within any desired limit by computing the change in velocity which will produce that excess pressure, and then arranging to have the rate of valve closure such that the time T exceeds $2l \div S$. For the time required to close gate valves, see page 1032 to 1034.

Wave Propagation in an Elastic Pipe.—Owing to elasticity of the pipe walls, part of the kinetic energy of the moving column is expended in stretching these, with a resulting increase in the complexity of the phenomena, a reduction in the maximum pressure attained, and an increase in the rate at which the pressure waves die out. The elasticity of the pipe may modify the results in two ways:

1. If the pipe is free to move in a longitudinal direction, the impact of the moving column of water against the closed or partially closed valve will tend to drive the pipe ahead in the direction of flow. As the compression wave in the liquid reflects and surges in the opposite direction, there is a tendency for the pipe to rebound in the opposite direction. This pulsation will continue until the disturbance inside the pipe dies down. Hence, the necessity to secure boiler-feed lines and similar piping with anchors or shock absorbers, especially where reciprocating pumps are used. The longitudinal stretching of a pipe free to move in this

direction superimposes the effect of a wave in the pipe material on the wave in the liquid, although the net effect on the liquid wave is slight and usually may be neglected.

2. The second effect of the elasticity of the pipe line is due to the fact that, since the walls extend both longitudinally and circumferentially under pressure, the apparent diminution of volume of the fluid under a given increment of pressure is greater than in a rigid pipe. The effect of this is to reduce the value of K obtaining with a rigid pipe to a virtual value of K' for elastic pipe. The value of K' is influenced by the lateral contraction of the pipe material under longitudinal strain, which is taken into account by introducing Poisson's ratio in the equation. The equation for determining K' is[1]

$$K' = \frac{1}{\dfrac{1}{K} + \dfrac{r}{2tE}(5 - 4\lambda)}.$$

If the pipe is so anchored that all longitudinal extension is prevented, but that circumferential extension is free, this becomes

$$K' = \frac{1}{\dfrac{1}{K} + \dfrac{2r}{tE}}.$$

The velocity of wave travel through a liquid in an elastic pipe then is, as explained in connection with rigid pipes,

$$S' = 12\sqrt{\frac{gK'}{y}},$$

and the excess pressure due to water hammer is

$$p_0 = \frac{v_0}{12}\sqrt{\frac{yK'}{g}} = \frac{yS'v_0}{144g}.$$

Under identical conditions, the velocity of wave travel and the excess pressure due to water hammer are always less in an elastic pipe than in a rigid pipe.

Numerical Values of Coefficients.—The bulk modulus of elasticity K of water and other liquids is approximately 300,000 psi, within a limit of about 10 per cent plus or minus, varying somewhat with the pressure and temperature (see page 266), which

[1] The derivation of this equation is too long to include here. Reference may be made to footnote 1(*d*) on p. 291. The symbol $1/\sigma$ used there is identical with λ used here for Poisson's ratio.

is sufficiently accurate in view of the uncertainty of other factors entering the problem. Poisson's ratio and average values of the modulus of elasticity at room temperature for common piping materials are given in Table XLIX.

TABLE XLIX.—POISSON'S RATIO AND AVERAGE VALUES OF THE MODULUS OF ELASTICITY FOR COMMON PIPING MATERIALS AT ROOM TEMPERATURE

	Poisson's ratio λ	Modulus of elasticity in tension E
Brass or bronze	0.333	14,000,000
Copper	0.333	16,000,000
Cast iron	0.270	12,000,000
Cement asbestos	0.10 to 0.20	3,400,000
Concrete	0.10 to 0.20	3,000,000
Lead	0.450	2,500,000
Steel	0.303	30,000,000
Wrought iron	0.278	28,000,000
Wood	....	1,500,000

Numerical Values for Wave Velocity and Excess Pressure Due to Water Hammer.—The numerical values for steel, wrought-iron, and cast-iron pipe shown in Table L were computed by the above formulas. The values of p_0 are for instantaneous reduction in velocity of 1 ft per sec and complete reflection of

TABLE L.—NUMERICAL VALUES OF TERMS USED IN FORMULAS FOR WATER HAMMER

(Pipe free to extend both longitudinally and circumferentially)

r/t	Cast iron $E = 15,000,000$ $\lambda = 0.270$			Wrought iron $E = 28,000,000$ $\lambda = 0.278$			Steel $E = 30,000,000$ $\lambda = 0.303$		
	K'	S'	p_0[1]	K'	S'	p_0[1]	K'	S'	p_0[1]
Rigid pipe	300,000	4,720	63.5	300,000	4,720	63.5	300,000	4,720	63.5
5	251,000	4,320	58.1	271,500	4,490	60.5	274,000	4,510	60.7
10	215,700	4,000	53.8	248,300	4,293	57.7	252,000	4,325	58.2
15	189,100	3,750	50.5	228,500	4,119	55.5	233,900	4,165	56.1
20	168,000	3,535	47.5	212,000	3,964	53.4	217,900	4,025	54.2
25	151,500	3,355	45.2	197,100	3,827	51.6	203,500	3,890	52.4
30	137,500	3,195	43.0	184,700	3,702	49.9	191,300	3,775	50.8
35	126,500	3,063	41.2	173,700	3,588	48.4	180,400	3,665	49.3
40	116,900	2,945	39.6	163,600	3,485	47.0	170,700	3,565	48.0
45	108,500	2,840	38.2	155,000	3,390	45.6	161,800	3,463	46.6
50	101,300	2,745	36.9	146,900	3,302	44.5	154,000	3,385	45.6

[1] Values of p_0 per unit change in velocity ($v_0 = 1$) where time of valve closure T is less than $2l \div S$ and with complete reflection of the wave.

the wave. Corresponding values of p_0 for any other reduction in velocity can be obtained by multiplying these values by that reduction in velocity. The values of K', S', and p_0 given opposite different ratios of r/t are for pipe which is free to extend both longitudinally and circumferentially.

Numerical values for maximum excess pressure due to water hammer p_0 can be computed directly from the table without recourse to the formula as follows: divide the radius of the pipe by its thickness to obtain the ratio r/t; opposite this value of r/t read the excess pressure p_0 due to a change in velocity of 1 ft per sec, multiply this value by the change in velocity v_0 produced by valve closure to obtain the excess pressure p_0 caused by v_0 change in velocity. In case of complete stoppage of flow in a time less than $2l \div S'$, v_0 should be taken as the initial velocity v.

Example.—What will be the excess pressure due to water hammer in a 12-in. standard-weight steel pipe for an instantaneous reduction in velocity of 5 ft per sec?

Solution.—In this case $r = 6$ in., $t = 0.375$ in., and $r/t = 16$. Interpolating in the table to find the value of p_0 for $r/t = 16$ gives an excess pressure of 55.7 psi for a change in velocity of 1 ft per sec. For an instantaneous change in velocity of 5 ft per sec, the excess pressure will be $55.7 \times 5 = 279$ psi.

For cement-asbestos and concrete pipe the maximum shock pressure can be computed from the formulas and coefficients given or, for the usual r/t ratios of about 5 obtaining with these products, the excess pressure can be assumed to be of the order of 40 to 45 psi for each 1 ft per sec change in velocity within the critical period. In the case of reinforced concrete pipe the values of E, λ, K', and S' are difficult to determine accurately owing to its composite structure, but here again the r/t values are about 5 and it seems reasonable to assume a possible maximum shock pressure of 40 to 45 psi for each 1 ft per sec change in velocity within the critical period.

End Restraint.—The effect of end restraint is apparent from a comparison of the first and second formulas on page 295. For $\lambda = 0.250$ these formulas become identical and it follows that the excess pressure due to water hammer would be the same for either free or restrained ends. For materials having λ greater than 0.250 the excess pressure would be slightly higher for the assumption of pipe free to extend longitudinally. By the same token, the excess pressure with materials having λ less than 0.250 would be slightly less for the free end condition. Actually the differences

are too small to be significant in the face of the possible variations in λ, E, or the rate of valve closure. Hence for practical purposes it usually does not matter whether or not the pipe is assumed to be free to extend longitudinally.

Water-hammer Suppressors.—The rise in pressure caused by water hammer may be minimized by the use of relief valves, surge tanks, or air chambers. To obtain greatest effectiveness the relief valve or other form of suppressor should be located as close as possible to the source of the disturbance. Shock absorbers for hydraulic power transmission piping are discussed on page 1319. Equations for determining the relieving capacity of relief valves and the proportions of air chambers are given in footnote 1(*e*), page 291. The necessity for replenishing the air in air chambers should be recognized in considering their use as water-hammer suppressors. In some cases, restricting the passages between the pipe line and the air chambers increases the effectiveness of a given size of air chamber. Suppressors, as a general rule, do not eliminate shock entirely but will reduce it by 10 to 40 per cent which often is sufficient to remove the clanking sound. For a description of the use of relief valves and cushioned check valves, see pages 1031 to 1038.

The amount of kinetic energy set up by water hammer that has to be at least partially dissipated by a suppressor may be computed from the relation given on page 293; *viz.*, the kinetic energy of the liquid which has been transformed into increased pressure has a value of $p_0^2 \div 2K$ in.-lb per cu in. of liquid involved between the point of arrested motion and the point of normal pressure. In the case of a branch pipe, for instance, this may involve the entire column of liquid flowing in the branch. The suppressor must be designed to absorb a good part of this energy and return it to the liquid in the proper part of the cycle or, in the case of a relief valve, to spill a certain amount of liquid from the pipe.

Design Values for Water Hammer.—In the case of pipe made of steel or other relatively elastic materials it is sometimes permissible to assume that occasional shock pressures can be looked after satisfactorily within the designed factor of safety allowed for working pressure. This is the usual practice with boiler-feed lines and other steel piping used to convey water in power and industrial plants. The adequacy of this practice depends on the ratio of shock pressure to working pressure and, while it might do well enough to have a shock pressure of 200 psi in a

boiler-feed line designed for 1,000-psi working pressure, this might not do at all in a large city-water line designed for 50-psi working pressure.

With relatively brittle materials such as cast iron and cement or concrete, it is customary to design for an internal pressure which represents the sum of the nominal working pressure plus a reasonable allowance for shock pressure. This may be desirable also with *steel* pipe used for underground water service, owing to the likelihood of wastage of the pipe wall through corrosion. What shock pressure should be designed for may be a matter of engineering judgment, or it may be determined by empirical rules. Devices are often employed, particularly in large-diameter underground water-supply systems, which act to prevent building up the maximum shock pressures theoretically possible if all the change in velocity took place within the critical time period. As a first step, an effort should be made to lengthen the time interval for valve closure or other stoppage of flow; as a second step, where needed, means may be provided for cushioning or relieving shock as described under "Water-hammer Suppressors." Both possibilities are developed in the references listed in the footnote on page 291 and in the discussion of shock pressure in water-supply systems on pages 1031 to 1038.

Hence in waterworks practice it is customary to design for shock pressures considerably less than would be expected for the instantaneous loss in velocity of 5 ft per sec corresponding to usual maximum flow conditions in such lines. Many years ago the following schedule of shock pressures based on experience and varying with pipe diameter was suggested by Dexter Brackett,[1] when engineer of the Boston water works.

ALLOWANCES PROPOSED BY DEXTER BRACKETT FOR SHOCK PRESSURE DUE TO WATER HAMMER, PSI

Nominal pipe diameter, in.....	3 to 10	12 and 14	16 and 18	42 to 60
Allowance for water hammer,	120	110	100	70

These allowances have found wide acceptance in the waterworks industry and are the basis for rules set up for the design of cast-iron

[1] See "A Proposed New Method for Determining Barrel Thickness of Cast Iron Pipe," by T. H. Wiggin, M. L. Enger, and W. J. Schlick, *Jour. AWWA*, Vol. 31, p. 843, May, 1939. Brackett is said to have recommended these allowances prior to 1895.

pipe in the American Standard Code for Pressure Piping, ASA B31.1, and in the American Recommended Practice Manual for the Computation of Strength and Thickness of Cast Iron Pipe, ASA A21.1.

In long-distance gravity-flow water-supply conduits and other places where economic considerations forbid designing the line for water-hammer allowances as generous as those suggested by Brackett, it becomes necessary to hold shock pressures to satisfactory limits through the use of cushioned check valves and overflow pipes, surge tanks, or relief valves of either the direct-acting or relay-operated types as discussed on pages 1031 to 1038 in Chap. XII on Water-supply Piping.[1]

Entirely different conditions are encountered with high-pressure piping for operating hydraulic presses and similar equipment where water velocities of 20 to 40 ft per sec are customary (see pages 1313 to 1317) with almost instantaneous stoppage of the press plunger at the end of each stroke. Under these circumstances shock pressures of 2,000 psi or more may be experienced unless adequate suppressors are provided.

[1] See also "Water Hammer Correctives," by Richard Bennett, *Water Works & Sewerage*, Vol. 89, pp. 92–97, 1942. This article describes a number of relieving and suppressing devices with illustrations.

CHAPTER III

METALLURGY OF PIPING MATERIALS

SELECTION OF SUITABLE MATERIALS

The choice of materials for any given application is prescribed in a general way through the interrelation of safety codes, dimensional standards, and material specifications which are abstracted elsewhere in this handbook. A certain amount of discretion remains with the designing engineer, however, in choosing between two or more materials which are listed as equally acceptable for a specific purpose. Hence some discussion of the considerations involved in making such choices would seem in order and is presented here. The subject of corrosion-resisting materials and protective coatings is deemed of sufficient importance to cover separately in Chap. XVII on Corrosion. Some further discussion of corrosion as encountered in specific applications will be found in Chap. XII on Water-supply Piping and in Chap. XV on Gas Piping.

Stock Materials.—No serious difficulty is experienced in obtaining satisfactory piping materials for the ordinary service conditions which have been common practice for many years past. The strength of such materials and their limitations as to temperature, corrosion, etc., in usual operating practice are well known. To ensure getting satisfactory materials for such service it is only necessary to select the proper metal, specify certain well-known physical or chemical properties, or both, and assure oneself by inspection, manufacturer's certification, etc., that the correct article is furnished. In buying valves, fittings, pipe, etc., for ordinary service from reputable manufacturers, it is frequently sufficient to select from their catalogues the articles which meet one's requirements, and accept them, subject to proving satisfactory in use, without resorting to any elaborate inspection by the purchaser before or after shipment. In such cases it is assumed that routine tests made by the manufacturer are sufficient to

maintain a uniform and satisfactory quality of product. When unusually large orders are placed with a manufacturer, it is sometimes deemed desirable to station a resident inspector at his plant, employ the services of an inspection agency, or send on a traveling inspector at appropriate times. The great bulk of common piping material is accepted, however, on the manufacturer's reputation for quality of product and business integrity.

Proper Specification Requirements.—From a strength standpoint, the acceptability of a piping material for any given service hinges on whether the operating temperature exceeds the safe working temperature of that material. In general, if it is a well-established fact that the physical properties of the material at the working temperature are greater than, or approximately equal to, the same properties at atmospheric temperature, the physical properties at atmospheric temperature may be used as a basis of design and acceptance test. The chemical properties and heat treatment are of interest, particularly as a criterion of what physical properties and endurance may be expected of the material. In many instances a considerable variation in chemical properties and heat treatment will give equally satisfactory results. For this reason it is frequently permissible to specify acceptable physical properties for some general type of material with the statement that the chemical properties and heat treatment shall be such as will give these desired physical properties. If it is definitely known that certain elements are objectionable in the material, it is advisable to specify limits on the percentage content of these elements which will be tolerated: for instance, in the case of steel castings, forgings, and bolts, it is customary to specify the maximum allowable sulphur and phosphorus contents. An attempt on the part of a purchaser of piping material to specify simultaneously in exact terms its physical and chemical properties and heat treatment should be based only on sound metallurgical experience and definite knowledge that those requirements are the most suitable. In many instances it is preferable to specify the minimum acceptable physical properties of some general type of material with a clause to the effect that the chemical properties and heat treatment shall be a matter of agreement. This leaves the door open to a number of varieties of that type of material, with possibilities for improvement over what the purchaser originally had in mind. Incidentally, this procedure opens the field to a larger source of supply which stimulates competition and

should result in a more satisfactory price from the consumer's standpoint. An alternative way of accomplishing an equivalent result, where simultaneous physical and chemical properties and heat treatment are specified, is to insert a clause inviting vendors to quote, also, on some material which they prefer to offer as a substitute. A good example of where such procedure is advisable is the purchase of alloy-steel bolts where a number of equally good alloys are available. This is recognized in ASTM Specification *A*96 (see Chap. IV) from which the following is quoted: "Nickel, chrome-nickel, chrome-vanadium, chrome-manganese or any other type of alloy steel approved by the purchaser may be submitted under these specifications."

In ordering piping materials it is advisable, where practicable, to refer to existing specifications which are well known, as those of the ASTM, rather than to write a specification of one's own. A great deal of confusion is avoided with a standard specification, and its use is apt to secure a better price through supplying a standard rather than a special article. In case special requirements are imperative, it is often desirable to say that the material shall conform to a certain standard specification except in such and such respects. Unless the exceptions are too extensive, this is less confusing than to attempt to write an entirely new specification.

In placing an order for material to be used in high-temperature service, it is important to specify at what temperature it will operate and to require the manufacturer to guarantee that it will prove satisfactory for the purpose intended. This is not only a necessary precaution for self-protection on the purchaser's part, but it gives the manufacturer a fair chance either to furnish material that he knows is right, or else to decline the order. A discussion of the suitability of different metals for high-temperature service is given in later paragraphs.

Properties of Common Metals.—The data presented in Tables I and II, which have been collected from various sources, are offered as typical of the metals listed. Additional data on seat metals are given in Table III. All of these tables are grouped at the end of this chapter. Other data on piping materials will be found in the abstracted specifications in Chap. IV.

STRESS-STRAIN RELATIONS FOR METALS

When external forces act on a solid body, they are resisted by reactions within the body which are termed "stresses" and

which are related to the elastic and cohesive properties of the material. A tensile stress is one which resists a force tending to pull a body apart; a compressive stress is one which resists a force tending to crush or squeeze a body; a shearing stress is one which resists a force tending to make one layer of a body slide across another layer; a torsional stress is a form of shearing stress which resists a moment tending to twist a body. Tensile and compressive stress may be developed in a material either by a direct acting force or by a bending moment. These various types of stress frequently occur in combination.

A brief explanation of the stress-strain relations of metals will be of assistance in understanding the terms designating these relations as used in later paragraphs on the properties of piping materials. Much confusion has arisen through the use in textbooks on engineering mechanics of the term *stress* to denote *force in pounds* whereas its meaning should have been restricted to *unit load in pounds per square inch.* Likewise *strain* has been used wrongly to denote *total elongation* instead of *elongation per unit of length.* Loose use of the terms "elastic limit," "proportional limit," "yield point," and "yield strength" also has been prevalent. The following definitions abstracted from the "Standard Definitions of Terms Relating to Testing," ASTM Designation E6-36, should assist in clarifying these terms.

Stress.—The intensity (measured per unit area) of the internal distributed forces or components of force which resist a change in the form of the body. Stress is measured in force per unit area (pounds per square inch, kilograms per square millimeter, etc.).

NOTE.—It is customary to compute stress on the basis of the original dimensions of the cross section of the body.

Strain.—The change per unit of length in a linear dimension of a body, which accompanies a stress. Unit strain is measured in inches per inch of length or in millimeters per millimeter.

NOTE.—Under tensile or compressive stress, strain is measured along the dimensions under consideration. Shearing strain is measured at right angles to the dimension under consideration. In torsion tests, which involve shearing stress, it is customary to measure angle of twist, which may be translated into terms of strain.

Stress-strain Diagram.—A diagram plotted with values of stress as ordinates and values of strain as abscissas (see Fig. 1).

NOTE.—The use of the term "stress-strain diagram" is frequently extended to cover diagrams plotted with values of applied load or of applied moment as

ordinates and with values of stretch, compression, deflection, or twist as abscissas. Methods of plotting stress-strain diagrams with equal increments of stress, equal increments of strain, and with two values of increments of stress are described in detail in ASTM Standard E6-36.

Elastic Limit.—The greatest stress which a material is capable of developing without a permanent deformation remaining upon complete release of the stress.

NOTE.—Values found for elastic limit by means of observations of permanent deformation (set) after release of stress do not differ widely from the values found for proportional limit, the latter being defined as:

Proportional Limit.—The greatest stress which a material is capable of developing without deviation from the law of proportionality of stress to strain (Hooke's law). Since the determination of proportional limit is much more readily made than is the determination of elastic limit, it is customary to accept the proportional limit as equivalent to the elastic limit for such materials (metallic materials), and hence the proportional limit is frequently called the "proportional elastic limit."

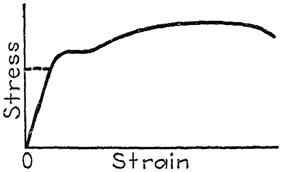


FIG. 1.—Stress-strain diagram.

Yield Strength.—The stress at which a material exhibits a specified permanent set.

NOTE.—The limit of usefulness of many materials, especially metals, in members subjected to approximately static loading at ordinary temperatures is therefore determined by a measurable value of plastic yielding of the material above which the material is considered to be damaged and below which the damaging effects are considered to be negligible.

The following methods are recommended for determining the yield strength of a material:

1. For material that has a "sharp-kneed" stress-strain diagram and hence exhibits at a certain stress the special characteristic of yielding without increase in stress, two satisfactory methods are in use, as follows:

(*a*) *"Drop of the Beam" Method.*—In this method the load is applied to the specimen at a steady rate of increase and the operator keeps the beam in balance by running out the poise at approximately a steady rate. When the yield strength of the material is reached the increase of load stops, but the operator runs the poise a trifle beyond the balance position, and the beam of the machine drops for a brief but appreciable interval of time. In a machine fitted with a self-indicating load-measuring device, there is a sudden halt of the load-indicating pointer corresponding to the drop of the beam. The load at the "halt" or the "drop" is recorded, and the corresponding stress is taken as the yield strength.

This method of determining the yield strength requires only one man to conduct a test.

(*b*) *Total Strain Method Using Dividers.*—This method is frequently called the "dividers method." In this method an observer with a pair of dividers, or other suitable apparatus, watches for visible elongation between two section marks on the specimen. When visible stretch is observed, the load at that instant is noted, and the stress corresponding to the load is taken as the yield strength. For the higher strength steels, a gauge length of less than 8 in. is recommended.

Note on Yield Point.—The above two methods determine what is usually called the yield point which is commonly defined as follows:

Yield Point.—The stress in a material at which there occurs a marked increase in strain without an increase in stress.

It should be noted that only materials that exhibit this unique phenomenon of yielding have a yield point. The term yield point should not be used in connection with material whose stress-strain curve in the region of yield is a smooth curve of gradual curvature.

2. For material whose stress-strain diagram in the region of yield is a smooth curve of gradual curvature.

(*a*) *Offset Method.*—This method can be used, if desired, for materials having "sharp-kneed" stress-strain diagrams, but is especially adapted to materials whose stress-strain diagram in the yield range is a smooth curve of gradual curvature.

For nearly all materials, if at any point on the stress-strain diagram such as r in Fig. 2, the load is released, the diagram for decreasing load will follow a line, r_m, approximately parallel to the initial portion, OA, of the diagram for increasing load. The offset Om will then give the approximate value of the permanent set after release of the stress OR. The value of this set is given in percentage of the original gauge length. Thus, to determine the yield strength by the "offset method," it is necessary to secure data (autographic or numerical) from which a stress-strain diagram may be drawn. Then with the stress-strain diagram (Fig. 2) lay off O equal to the specified value of the set and draw mn parallel to OA and thus locate r, the intersection of mn with the stress-strain diagram. Draw Rr parallel to the X axis, and then OR gives the value of the yield strength.

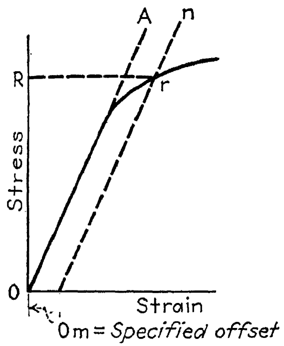


FIG. 2.—Offset method of determining yield strength.

In reporting values of yield strength obtained by this method, the specified value of "offset" used should be stated in parenthesis after the term "yield strength."

The "offset" method is devised for determining a stress corresponding to a well-marked plastic deformation, or set, and it is not feasible to specify a small value of the permissible "offset" *Om.*

In the specification for Seamless Alloy-steel Pipe for Service at temperatures from 750 to 1100 F (ASTM Designation A158), a yield strength by the offset method of 0.2 per cent is specified for materials not exhibiting a definite yield point.

Tensile Strength.—The maximum tensile stress which a material is capable of developing.

NOTE.—In practice, it is considered to be the maximum stress developed by a specimen representing the material in a tension test carried to rupture, under definite prescribed conditions. Tensile strength is calculated from the maximum load carried during a tension test and the original cross-sectional area of the specimen.

Compressive Strength.—The maximum compressive stress which a material is capable of developing.

NOTE.—In the case of a material that fails in compression by a shattering fracture, the compressive strength has a very definite value. In the case of materials that do not fail in compression by a shattering fracture, the value obtained for compressive strength is an arbitrary one depending upon the degree of distortion which is regarded as indicating complete failure of the material.

Modulus of Elasticity.—The ratio, within the elastic limit of a material, of stress to corresponding strain. For numerical values, see Table I, page 344.

NOTE.—As there are three kinds of stress, so there are three moduli of elasticity for any material: the modulus in tension, the modulus in compression, and the modulus in shear. The value of the modulus of elasticity in tension is nearly the same, for most metals, as the value of the modulus of elasticity in compression. The value of the modulus of elasticity in shear is smaller than the value of the modulus of elasticity in tension. The modulus of elasticity is expressed in pounds per square inch (kilograms per square millimeter).

The modulus of elasticity is a measure of the stiffness of any material and determines the slopes of its stress-strain line—the higher the modulus of elasticity, the stiffer the material and the steeper the line. Elasticity is the property that enables deformed bodies to resume their original shape after removal of load. A point that is frequently overlooked is that the modulus of elasticity changes with temperature (see Fig. 39, page 830). In general, as the temperature increases, there is a tendency for the modulus to

decrease. This change is of importance in considering the elastic deformation under load and is of especial significance in connection with the study of stresses and reactions resulting from thermal expansion.

Additional Stress-strain Definitions.—A definition of certain stress-strain terms not covered in the above ASTM definitions, together with an amplification of certain definitions there given, will aid in understanding the use of these terms in later paragraphs.

Bursting Strength.—In the case of steel pipe its bursting strength as determined by the Barlow or equivalent formulas (see pages 31 to 42) closely approximates the tensile strength of the material as determined from standard tensile test coupons. With cast-iron pipe this is not the case, however, since failure almost invariably occurs at a lower pressure than would be computed by substituting the tensile strength of the material in either the Barlow or the "common" formula. This difference in behavior probably comes about because bursting pressure subjects the material to other stresses besides pure tension, to which condition cast iron is not so ductile in adjusting itself as is the case with steel. According to the research department of the American Cast Iron Pipe Co.,[1] cast-iron pipe which shows a tensile strength in the bursting test of only 20,000 psi, may give 30,000 to 40,000 psi when tensile specimens are *machined* from the pipe wall. Testing of *strips* cut from the wall of a pipe from the same lot will give a tensile strength of say 25,000 to 30,000 psi, somewhat lower than the machined specimens because of irregularities of the cast surfaces.

The *ultimate strength* or *maximum strength* of a material is the same as the tensile strength or compressive strength, as defined above.

The *breaking strength* of a ductile material is the load at failure divided by the reduced area at the neck where failure occurs in a tension-test specimen (see Fig. 3). The breaking strength ordinarily exceeds the tensile strength but is not of much commercial significance and is seldom reported.

Ductility is the property of elongation, above the elastic limit, under tensile stress. The measure of ductility is the percentage of elongation of the fractured test bar over an initial length of

[1] See "An Explanation of the New Law of Design for Cast Iron Pipe as Developed by Committee A21 of the American Standards Association," presented at the AGA Distribution Conference, April, 1939.

usually either 2 or 8 in. According to the standards of the ASTM, the measurement of elongation after fracture of test specimens can be made with sufficient accuracy by means of a pair of dividers and a scale. The elongation should not be reported for any tension-test specimen which breaks outside the middle third of the gauge length (see Fig. 3). It has been found approximately true that the same results will be obtained from two test pieces if they are of similar form when it is not possible to make

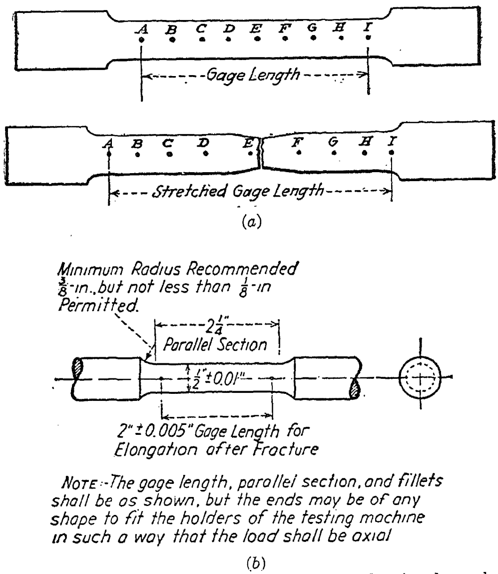


FIG. 3.—Tension-test specimens: (a) strip specimen showing how elongation is measured; (b) standard round specimen with 2-in. gauge length.

them of identical dimensions. This is known as *Barba's law*, or the law of proportionality or similarity, and is expressed by the formula $P = l/\sqrt{a}$, where P is the proportionality, l the gauged length, and a the cross-sectional area of diameter d. For the standard ASTM test specimen where $l = 2$ in. and $d = 0.505$ in., $P = 4.5$. Hence, $l = 4.5\sqrt{a} = 4d$ (approximately), and a proportional test specimen of other dimensions should have a length approximately four times its diameter.

Reduction of area at the neck where failure takes place (see Fig. 3) may be determined from direct measurement with a micrometer at the smallest section of the fractured specimen. Reduction of area is expressed as a percentage obtained from the ratio of

$$\frac{(\text{original area}) - (\text{area at neck})}{\text{original area}}.$$

Number of Tests.—A tension test from which the above physical properties can be determined is usually made on one or more specimens from each melt or pour or heat-treatment charge of the material. A test of one specimen is sufficient to determine these properties or to plot a stress-strain diagram—one or more additional specimens are frequently tested as a check. Attention is called to the simplicity of such a test at atmospheric temperature as compared with the extensive work required to determine these values over an extended temperature range, as discussed in later paragraphs.

Effect of Change in Temperature.—The test results described above are such as can be obtained from pulling a single tension-test specimen at atmospheric temperature. This test is the kind called for in ordinary specifications where a report on tensile strength, yield point, reduction in area, and elongation is required. Usually two or more tests are required from each lot for check, but the tests are all made on supposedly similar samples under approximately identical conditions. When it is desired to study the change in physical properties with temperature, it is necessary to pull a number of identical specimens at various temperatures under very carefully regulated conditions. Obviously such tests are of a research rather than a plant routine nature, and are too expensive for ordinary inspection purposes. Tests of this kind are useful in studying the physical properties of a metal over its operating temperature range. The curves of Fig. 4, which are plotted from the results of such a test on an alloy-steel bolting material, are included here as a typical illustration of this type of test. A number of identical specimens were made up from the same lot of bolting material coming from the same pour and having undergone the same heat treatment. Each specimen was then pulled at a different temperature and all the physical characteristics shown in the curves were observed. By repeating these tests at 50 or 100 F steps throughout the temperature range, sufficient data were obtained to plot the curves.

HARDNESS AND TOUGHNESS

Hardness is the property of metals which enables them to resist indentation, scratching, or abrasion. In valve seats and disks and similar parts, hardness coupled with toughness is of great importance in resisting abrasion and erosion. There are a number of instruments available for measuring hardness, among

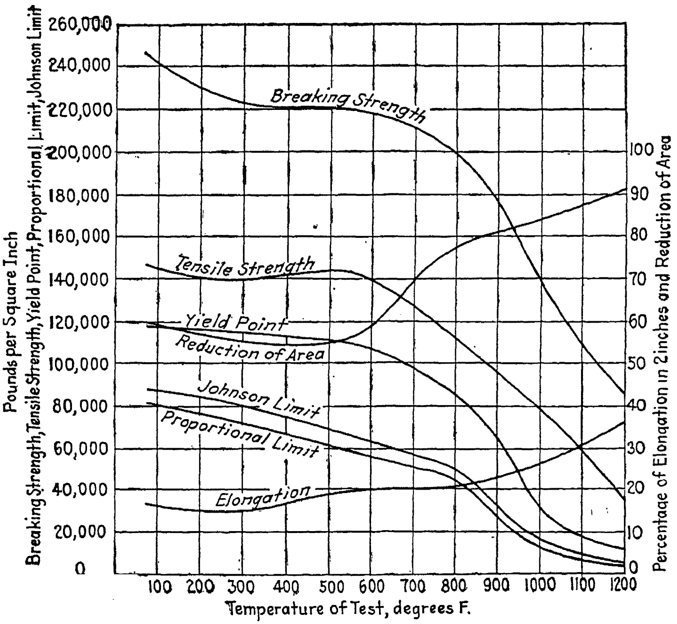


FIG. 4.—Tensile tests at elevated temperatures. SAE 3135 steel.

which are the Brinell and the Rockwell hardness testers, and the Shore sceleroscope. While the nature of the tests made with these three instruments varies considerably, there is apparently some relation between the results.[2] The Brinell test which is better known and more widely used than the others, is frequently given among the requirements in ASTM and other specifications.

[2] For complete discussion of hardness testing, see pp. 114–128 of ASM "Metals Handbook," 1939 ed.

The fact that there is a rather definite relation between the hardness and the tensile strength of metals is of importance in that the tensile strength can be approximately estimated from a very simple and inexpensive hardness test. This is of great value in making individual tests on large numbers of small objects, such as bolts, etc. The relation of Brinell hardness to tensile strength is discussed at more length on page 313.

In the standard *Brinell tester*, which is a hand-operated hydraulic press, a load of 3,000 kg is applied to a 10-mm ball for at least 10 sec for steel, and for at least 30 sec for other metals. For brass and other soft materials, a load of 500 kg is applied for 60 sec. The diameter of the impression is measured and the Brinell hardness number obtained from tables based on the formula

$$H = \frac{P}{\frac{\pi D}{2}\left(D - \sqrt{D^2 - d^2}\right)}$$

in which H = hardness number; P = applied load in kg; D = diameter of ball in millimeters; and d = diameter of impression in millimeters.

A variation of the Brinell tester, in which the hardness number is read from a dial while the load is on, is called a "direct reading" Brinell machine. Most machines of this type divide the total load of 3,000 kg into two parts: a preload, or seating load, and the load that actuates the dial. Since the total load is 3,000 kg, as in the standard Brinell tester, the impression diameter can later be read under a glass to give a true Brinell hardness number.

The Brinell numbers for steels range from 100 for mild steel to 350 or higher for heat-treated high-carbon and alloy steels. Standard methods of Brinell-hardness testing of metallic materials are explained at length in ASTM Standard E10, along with a very complete table of the Brinell numbers corresponding to different indentation diameters which saves the labor of computing the result.

The *Shore sceleroscope* consists of a small steel weight carrying a diamond point and a means for dropping it, within a glass tube, upon the specimen from a height of about 10 in. The rebound of the weight is observed on a vertical scale. The softer the metal, the farther the diamond point penetrates it and consequently the less the rebound, and vice versa. The scale is an arbitrary one calibrated by means of a standard steel specimen.

The *Rockwell hardness test* consists essentially in applying a minor initial load to a standard penetrator, adding to this a comparatively large increment called the "major load," releasing the major load, and reading the amount of movement of the penetrator with respect to the test piece due to the major load. The initial and final readings of penetration are read from an indicating dial while the minor load is on the penetrator. The Rockwell hardness is expressed as a number which is equal to a constant minus the number of gauge units on penetration under the major load. This is done in order that high readings may be expressed by large numbers. The dial is arranged so that the number may be read directly. Because of its greater speed and the fact that a flat surface is not so essential as in the case of the Brinell test, the Rockwell test is commonly used in checking uniformity of boiler tubes, bolting, and the like.

Toughness is the property of a material to resist impact, or to withstand repeated reversals of stress, or to absorb energy when stressed beyond the elastic limit. In piping work the resistance to shock is a desirable attribute of valve bodies and fittings, etc., which are subject to water hammer and similar forms of surge. An approximate indication of the toughness of metals can be obtained by means of a pendulum impact machine. A small notched bar of the material is prepared and fixed in position in the path of a relatively heavy pendulum. In the Charpy type of impact test the specimen is supported as a simple beam, whereas in the Izod type the specimen is clamped at one end to form a cantilever beam. The general requirements as to rigidity of machine, dimensions of specimens, methods of notching, and necessary precautions in testing are covered in Methods of Impact Testing of Metallic Materials, ASTM Specification E23.

RELATION OF BRINELL AND ROCKWELL HARDNESS TO TENSILE STRENGTH

As pointed out on page 312 in connection with hardness tests, a more or less definite relation exists between the Brinell hardness and tensile strength of each material. After this relation has once been experimentally determined for any particular metal, it is possible to estimate the tensile strength of a piece of that metal from a relatively simple Brinell or Rockwell test. While tensile strengths estimated in this way are not so reliable as those determined by pulling a specimen in a tension-testing machine,

they are extensively used as check tests, especially on large lots of small objects, such as alloy-steel bolts. If the results of such Brinell tests seem to show that the metal does not have the desired

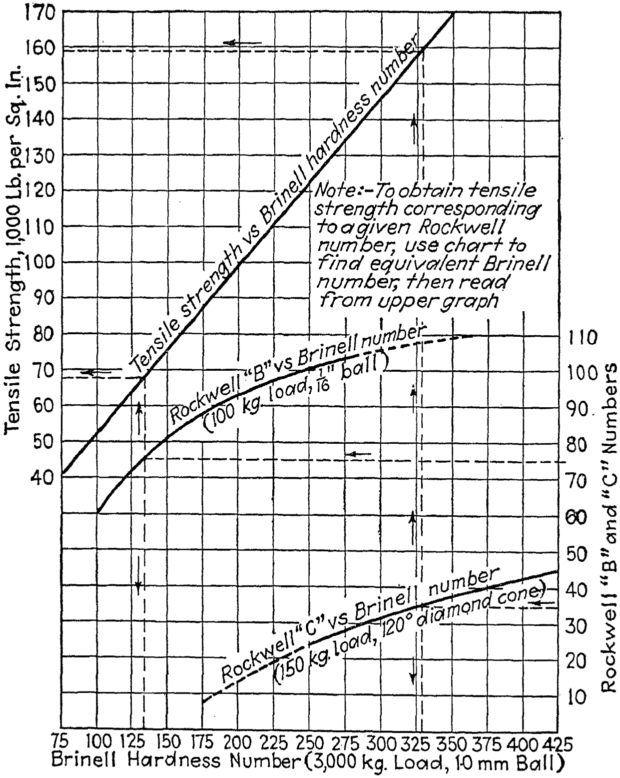


FIG. 5.—Conversion chart for Brinell and Rockwell hardness numbers, giving corresponding tensile strength for steel. Based on hardness conversion table (*ASM* "*Metals Handbook*," pages 127–128, 1939 *ed.*)

tensile strength, it is the usual practice to pull one or more tension-test specimens to settle the question. Where Brinell hardness and tensile strength are both used as requirements in a specification, a clause is frequently inserted to the effect that the Brinell tests shall not be a basis of rejection without confirming tensile tests.

Brinell tests of this kind are applicable to a considerable variety of metals, but they are most frequently used in estimating the tensile properties of steel (see Fig. 5).

TEMPERATURE LIMITATIONS OF METALS

The selection of suitable materials for high pressures alone is not a difficult problem. For many years past, piping systems have been operated handling water or air under pressures of 2,000 to 3,000 psi at atmospheric temperatures. There is no particular difficulty in obtaining satisfactory materials for such service, but the safe handling of oil or superheated steam under comparatively moderate pressures at temperatures of 900 F or higher is quite a different matter. The reduction in central-station fuel consumption obtainable through the use of steam temperatures of 900 F and higher has encouraged the development of new materials having satisfactory load-carrying ability at such temperatures. Similar demands in the oil industry for heat- and corrosion-resistant materials for use at 1000 to 1200 F have resulted in the production of many new alloy steels.

In determining whether a material is suitable for high-temperature service, a study must be made of the physical properties of the metals at the operating temperature, such as tensile strength, proportional limit, resistance to shock, endurance or fatigue limit, torsional elastic limit, toughness or wearing qualities, resistance to erosion, and, finally, ability to withstand continued load over periods of long duration without distortion or undue plastic flow. Excellent physical properties at elevated temperatures are not the only criteria to be used in selecting materials, as scaling and oxidation are decided factors. The real difficulty in high-temperature work is in combining the necessary physical properties with sufficient chemical stability so as to obtain satisfactory performance during the useful life of the equipment.

Tensile Strength at Elevated Temperatures.—The tensile strengths of some of the common piping materials tend to increase with reference to the corresponding values at atmospheric temperatures for a few hundred degrees and then fall off rapidly with further rise in temperature (see Fig. 6). Certain other materials show a continuous decrease in strength with increase in temperature. A great deal of experimental work has been done in the last few years to determine the properties of materials at elevated temperatures.

In June, 1931, a Symposium on Effect of Temperature on the Properties of Metals was held at a joint meeting of the American Society of Mechanical Engineers and the American Society for Testing Materials in Chicago. The curves for chrome-nickel and carbon cast steel, rolled medium- and low-carbon steels, cast iron, cast brass, and rolled copper of Fig. 6 are taken from that symposium. The curves for 18 per cent chromium—8 per cent nickel,

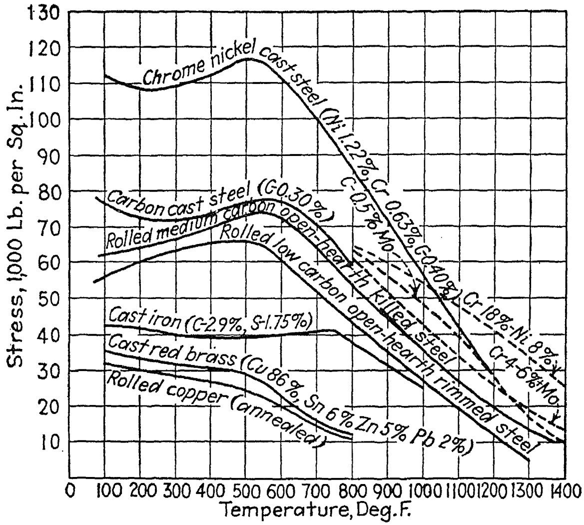


FIG. 6.—Short-time tensile strength of various piping materials at elevated temperatures. Solid curves are taken from the 1931 *Symposium on Effect of Temperature on the Properties of Metals.* Broken-line curves are plotted from Table III of National Tube Company's *Bulletin,* "Properties of Steel Pipe and Tubes Applicable to High Temperature and High Pressure Service," 1936.

4 to 6 per cent chromium —0.50 per cent molybdenum, and carbon-molybdenum (0.50 per cent molybdenum) steels are taken from the National Tube Company's Bulletin, "Properties of Steel Pipe and Tubes Applicable to High Temperature and High Pressure Service," 1936. The latter data were obtained in accordance with the specified procedure given in ASTM Specification E21.

This test code for short-time tests at elevated temperature was developed by the ASTM-ASME Joint Research Committee on Effect of Temperature on the Properties of Metals in order to

eliminate as many variables as possible from such tests. Fourteen laboratories cooperated in an investigation[3] of a special rolling of an open hearth 0.35 per cent carbon killed steel to secure information for this committee. Reasonably comparable results were obtained from 12 of the cooperating laboratories. The average of the reported tensile values at 850 F was approximately 64 per cent of the values at room temperature. This reduction in short-time tensile strength is somewhat greater than indicated for rolled medium-carbon killed steel on Fig. 6. The unusual heat treatment given the material for the cooperative tests to secure uniformity may have been responsible for this difference.

In any event, estimates of load-carrying ability at temperatures above 750 F should be based on long-time flow or "creep" tests and the chemical and structural stability of the material rather than on short-time tests.

Temperature Limitation of Gray Cast Iron and Malleable Iron.—Gray cast iron and malleable cast iron exhibit quite constant tensile strengths on short-time tensile tests up to approximately 800 F. (See curve for cast iron on Fig. 6.) Few creep tests have been conducted on cast iron, but one 2,000-hr test reported by R. J. Allen[4a] indicated that a good grade of gray iron would support a stress of 10,500 psi at 700 F without excessive creep. J. W. Bolton[4b] also reported a creep test in which a stress of 10,000 psi was applied to cast iron at 600 F with only slight creep.

Despite the retention of reasonably good properties at moderately elevated temperatures, as shown by such tests, the use of cast iron at elevated temperatures is limited by the phenomenon termed "growth." Growth refers to the permanent increase in volume which occurs under certain conditions of heating and cooling and is considered to be independent of stress, although it is considerably affected by the presence of superheated steam and certain corrosive fluids. In severe cases of growth, the volume may increase as much as 50 per cent with an attendant loss of strength and development of brittleness which make it worthless as an engineering material under such conditions.

[3] "Short-time Tensile Tests at 850 F of the 0.35 Per Cent Carbon Steel Material," K20, *Trans. ASME*, February, 1936, RP58-4, p. 97.

[4a] Discussion by R. J. Allen of paper on "Properties of Gray Iron," by J. W. Bolton and Hyman Bornstein, 1931 *Symposium on Effect of Temperature on Properties of Metals*, ASTM-ASME, 1931, p. 459.

[4b] Closure of above paper, *ibid.*, p. 464.

J. W. Bolton has summarized the cause of growth under the following headings: (1) expansion due to graphitization, (2) corrosion, (3) thermal gradients, (4) allotropic changes, and (5) pressure from occluded gases.

These factors are stated to act in an interdependent manner, and most of them are cumulative in effect.

Below a temperature of 700 to 800 F at which graphitization is of importance, corrosion appears to be the important factor. In the case of coarse-grained cast iron, severe growth and disintegration have been observed with superheated steam at temperatures below 600 F. Apparently the graphite flakes in combination with porosity allow penetration of oxidizing gases. The resulting products of oxidation of the silicon and iron occupy greater volume and tend to destroy the continuity of the structure.

Rapid heating or cooling, nonuniform sections, and local heating set up temperature gradients or differences which may result in stresses sufficient to cause cracks. Since gray cast iron has little ductility, even slight overstress may result in cracks. These cracks further facilitate infiltration of the corroding gas or liquid. Pressure from occluded gases may be a contributing factor in the formation of such cracks.

Growth due to graphitization in the temperature range 900 to 1300 F and growth due to allotropic changes (change of the pearlite-ferrite mixture to austenite), which occur at about 1340 F, have little practical significance in piping practice. The ASME Boiler Code limits the use of cast iron for pressure-containing parts to temperatures below 450 F. The ASA Code for Pressure Piping likewise limits the use of cast-iron pipe, valves, and fittings to temperatures not in excess of 450 F. A further limitation that cast-iron pipe shall not be used for oil at a temperature above 300 F is made in the latter code in both the "Power Piping" and "Oil Piping" sections.

Specific limitations for cast-iron pipe and fittings as to pressure, pipe sizes, and allowances for water hammer are contained in the various sections of the Code for Pressure Piping. These restrictions in conjunction with the temperature limitations are designed to prevent the use of cast iron under conditions which might constitute a hazard to life or property by reason of its low ductility and tendency to growth.

The resistance of gray cast iron to growth may be definitely improved by care in design to avoid nonuniform sections, by secur-

ing a dense grain, by reducing the total carbon content so that thinner graphite flakes will result, and by the addition of small amounts of nickel, chromium, and molybdenum, or other alloys. While it is possible to double the room-temperature tensile strength by these measures and even to secure a certain degree of ductility, cast iron is not to be considered a high-temperature engineering material. High-alloy cast irons containing 14 per cent nickel with additions of copper, chromium, or silicon have been developed which resist growth and oxidation at temperatures up to 1500 F. These alloy cast irons are termed "austenitic" cast irons since they retain a portion of the carbon in solid solution. The thermal expansion of these austenitic cast irons is approximately 50 per cent greater than ordinary cast irons.

Malleable cast iron is produced by prolonged heating at temperatures up to 1750 F and slow cooling of cast iron which in the as-cast condition has a white fracture and is extremely hard and brittle. The heat treatment converts the combined carbon which exists as iron carbide into nodules of graphite surrounded by ferrite. Because the graphite in fully malleableized cast iron exists as rounded nodules of free carbon rather than heavy flakes as in gray cast iron, it does not afford paths for the infiltration of oxidizing gases. The lower silicon content of malleable cast iron reduces the effect of oxidation of that element while its greater ductility minimizes the effect of temperature gradients.

At temperatures up to about 900 F, there appears to be satisfactory creep strength and little evidence of growth of malleable cast iron.[5] The Code for Pressure Piping permits the use of malleable cast iron at temperatures up to 500 F, provided the pressure is limited to 300 psi. The Boiler Code permits the use of malleable cast iron for boiler and superheater connections such as pipes, fittings, water columns, valves, etc, for pressures not to exceed 300 psi, provided the steam temperature does not exceed 450 F.

Malleable cast iron, if properly malleableized and if not embrittled by improper galvanizing, possesses good resistance to shock. Its ductility as indicated by an elongation in 2 in. of 10 to 15 per cent and a reduction of area of 15 to 20 per cent makes it a useful engineering material for small-pipe fittings and other light sectioned parts.

[5] See "Some Creep Studies on Cupola Malleable Cast Iron," by J. J. Kanter and Glen Guarnieri, *Proc. ASTM*, 1942, Vol. 42, pp. 659–667.

Standard specifications for malleable cast iron for railroad, motor-vehicle, agricultural-implement, and general-machinery purposes are given in ASTM Specification A47. Cupola malleable iron is covered by ASTM Standard Specification A197. The properties of both types will be found in Tables I and II on pages 344 to 345.

By proper modification of heat treatment it is possible to produce malleable iron castings with widely different physical properties. These different properties are in general obtained by arresting the graphitizing process at different stages so the matrix surrounding the globules of graphite or temper carbon will retain a greater or less amount of carbon in the combined state. These matrices may be ferritic, as in fully graphitized iron, pearlitic, or sorbitic in character, or they may be in the spheroidized condition. In this latter condition tensile strengths in excess of 100,000 psi with an elongation of some 7 per cent in 2 in. have been obtained.[6]

Temperature Limitations of Nonferrous Alloys.—Copper tubing, copper and brass pipe, brass valves, and copper-alloy valve-trim materials have great commercial importance. Their use for elevated temperatures is limited to temperatures below the lower recrystallization temperature for the particular alloy, the lower recrystallization temperature in this instance being defined as the temperature at which cold-worked specimens begin to soften. This recrystallization is usually accompanied by a marked reduction in tensile strength and a corresponding increase in ductility.[7] Grain size and degree of cold working have an important effect on creep resistance.[8] Exceptionally coarse-grained nonferrous material, either annealed or cold worked, may become embrittled and fail suddenly even though exhibiting a high creep resistance. For annealed 70–30 brass, creep resistance increases with grain size at 300 to 500 F for stresses within the range normally used. Finer grain metal may be superior at temperatures below 300 F, at least it has a greater load-carrying capacity without appreciable deformation because of a higher yield strength. White and Clark[9]

[6] "Malleable Castings," by Albert Sauveur and Harry L. Anthony, *Trans. ASM*, June, 1935, p. 409.

[7] See "Properties of Copper and Some of its Important Industrial Alloys at Elevated Temperatures," by W. B. Price, *Symposium on Effect of Temperature on Properties of Metals*, ASTM-ASME, 1931, p. 341.

[8] See "The Creep Characteristics of Some Copper Alloys at Elevated Temperatures," by H. L. Burghoff, A. I. Blank, and S. E. Maddigan, *Proc. ASTM*, 1942, pp. 668–689.

[9] Discussion by A. E. White and C. L. Clark of reference 7, p. 365.

concluded that brasses containing 70 per cent or more of copper may be used successfully at temperatures up to 400 F, while those containing only 60 per cent of copper should not be used over 300 F.

The ASME Boiler Construction Code limits the use of brass and copper pipe and tubing intended for piping service, as distinguished from heater tubes, to temperatures not exceeding 406 F. The ASA Code for Pressure Piping limits the use of brass and copper pipe and tubing to temperatures not over 406 F for steam, gas, and air piping. Bronze or brass for fittings and valves is limited to 406 to 500 F, depending on the composition. The section on "Oil Piping" of the Code for Pressure Piping states that "Bronze or brass (fittings and valves) shall not be used over 350 F, or where a fire hazard is involved. Bronze or brass (trim for valves) shall not be used over 400 F, or where a fire hazard is involved." Brass and copper pipe or tube for oil piping is also limited to 350 F.

Nickel, nickel-copper, nickel-manganese, and nickel-copper-zinc alloys may be used successfully at considerably higher temperatures than copper and copper-zinc alloys. The temperature limits, as in the case of the copper-zinc alloys, depend to a large extent on the temperature at which softening or recrystallization begins. This temperature for nickel and the higher nickel-content alloys may be roughly set at 800 F. Tests[10a] on an annealed nickel at 800 F indicated that a stress of 2,000 psi would cause the material to creep at a rate of approximately 3 per cent in 100,000 hr. No appreciable creep was found for a stress of 10,000 psi at 550 F. Hot-rolled monel metal was reported by Clark and White[10b] as supporting a stress of 26,000 psi at 600 F and 19,000 psi at 800 F with creep at the rate of 1 per cent in 100,000 hr. At 1000 F the stress producing the same rate of creep was found to have dropped to only 1,650 psi.

Nickel, monel (70 per cent nickel, 30 per cent copper), and various modifications thereof have been used extensively in turbine blading, valve trim, and miscellaneous power-plant accessories handling steam. While some evidence exists of steam embrittlement of nickel and monel at high steam temperatures, Crawford

[10a] Tests conducted by Westinghouse Electric and Manufacturing Company, Paper on "Nickel Alloys," by C. A. Crawford and Robert Worthington, Symposium on Effect of Temperature on Properties of Metals, ASTM-ASME, 1931, p. 559.

[10b] Discussion by C. L. Clark and A. E. White, *ibid.*, p. 383.

and Worthington[10c] conclude that its effect is not serious in the temperature range 600 to 800 F.

Monel-metal gaskets have been used successfully in the flanged joints of a steam line supplying steam to a 10,000-kw 1000 F turbine.[11]

The presence of even small quantities of sulphur in a reducing or neutral furnace atmosphere will result in surface embrittlement of nickel and monel at temperatures of 700 to 1200 F.

Numerous copper-nickel alloys (60 to 70 per cent copper, 30 to 40 per cent nickel) with varying amounts of iron, zinc, tin, or aluminum additions have been developed for valve trim, turbine nozzle-blocks tubing, and the like. Some of these alloys have given exceptionally good performance up to 850 F on superheated-steam service while others have failed by intergranular cracking, pitting, etc., at less than 750 F. With such uncertain behavior it is impossible to set definite temperature limits for the copper-nickel alloys, but the present tendency is to substitute 12 to 14 per cent chromium stainless steel or some of the hard-surfacing cobalt-chromium tungsten alloys (stellite) for valve trim on superheated-steam service.

Cast Steel.—Carbon cast-steel fittings, valve bodies, and companion flanges are used extensively for temperatures between 450 and 750 F. The development of welding-end valves has extended the use of cast steel into the lower temperature ranges for special purposes such as oil lines where the elimination of flanged joints and cast iron valves is considered desirable.

Alloy cast steels[12] of the carbon-molybdenum, chrome-nickel-molybdenum, or similar low-alloy types are used for service involving temperatures from 750 to about 950 F. In the temperature ranges of 950 to 1100 F, higher alloy additions, such as 4 to 6 per cent chromium plus molybdenum, and the austenitic 18 per cent chromium 8 per cent nickel castings appear to give the greatest promise, although carbon-molybdenum castings have been used successfully for 1000 F service.[11]

Although it is probable all of these alloy cast steels can be used at higher temperatures than indicated above, provided the working

[10c] Conclusion of above paper, *ibid.*, p. 580.

[11] Discussion by P. W. Thompson of paper on "Development and Performance of American Power Plants" by A. G. Christie, *Mechanical Engineering*, February, 1937, pp. 103–104.

[12] ASTM Specifications covering alloy-steel castings for high-temperature service are abstracted in Chap. IV.

stresses are kept sufficiently low,[13] it is possible that the excessively heavy construction required may make their use under such conditions economically undesirable. It is pertinent to remark in this connection that considerable variation in alloy content may be made in valve-body castings without affecting the cost to any extent.

Although the particular composition of alloy casting material to be used for service at temperatures above 750 F should be selected on the basis of "creep" resistance and chemical and structural stability of the material at the particular operating temperature,[14] the soundness of the casting and the adequacy of its heat treatment can best be judged by tests at room temperature. The room-temperature physical properties for a given analysis and heat treatment should be established high enough to ensure that only test specimens which are sound and properly heat treated will pass. If the specified physical properties for a particular composition are set too low, tests on conventional test bars show only a perfunctory compliance with the specifications.

Tensile specimens to check conformance with specified values usually are machined from lugs cast attached to valve bodies when of sufficient size, otherwise from separately cast bars. It is assumed that values obtained from these bars represent the best that can be expected for the particular composition and heat treatment under ideal conditions, and allowances are made for porosity, shrinkage cavities, and other imperfections in the casting proper by making the metal thickness considerably heavier than might be deemed necessary for rolled material having the same nominal physical properties.

In the case of welding-end valves which have the ends machined so the wall thickness approaches that of the adjoining pipe, it is highly important that the cast material in the welding ends be as sound and reliable as the pipe material. An extensive investigation[15] of the physical properties of specimens cut from various sections of large welding-end valve bodies made of carbon-molybdenum cast steel showed that the welding ends were the soundest

[13] For discussion of safe working stresses at elevated temperatures, see pp. 336–345.

[14] For a discussion, see the article, "Selection of Materials for Cast Valves and Fittings," by A. M. Houser and H. L. Moe, *Steel*, Nov. 26, 1934.

[15] "Physical Property Uniformity in Valve-Body Steel Castings," by A. E. White, C. L. Clark, and Sabin Crocker, *Trans. ASME*, Vol. 58, No. 8, pp. 643–647, November, 1936.

part of the valve body and that they possessed physical properties commensurate with those found for the separately and integrally cast test bars.

Macrographs and tensile and impact tests on specimens cut from the heavy bonnet flange and from the shell or center portion of the valve body showed considerable porosity. But the lack of soundness of these sections was more than compensated for by the greater metal thickness. It was concluded from this investigation that further study with respect to the location and size of gates and risers, pouring temperature, and heat treatment should enable castings with more uniform properties to be produced.

A comparison of the above findings with those reported in previous papers on steel castings[16] indicates that considerable improvements in casting technique has taken place during the past few years.

The heat treatment used depends upon the composition of the casting, the purpose for which it is intended, and the preference of either the manufacturer or the purchaser. For extremely large castings, such as turbine castings, where possible distortion caused by release of cooling strains is not tolerable, it is customary to give the castings a full anneal,[17] *i.e.*, slow cooling in the furnace from above 1600 F.

Considerable improvement in grain refinement,[18] ductility, and impact resistance can be obtained in castings for valves, fittings, and other high-grade steel castings by normalizing and drawing in place of full annealing. Normalizing consists of heating to above 1600 F, followed by cooling in still air. A typical heat treatment used for chrome-nickel-molybdenum valve-body castings is as follows: (1) raise to 1900 F, hold for 3 hr; (2) cool in furnace to 1650 F, followed by cooling in still air; (3) draw by reheating to 1100 to 1200 F.

Variations in this heat treatment such as holding for a longer time at 1750 to 1800 F, reheating to 1500 to 1650 F and air quenching, or drawing at higher or lower temperatures may be made to

[16] See "Symposium on Steel Castings," *Journal of American Society of Naval Engineers*, February, 1933; also, "Alloy Steel Castings Regularly Made," by A. W. Gregg, Bonney-Floyd Co., *Iron Age*, Oct. 5, 1933, p. 15.

[17] An exception to this practice was noted in the heat treatment of the chrome-nickel-molybdenum turbine casing and valve chest of a 10,000-kw 1000 F turbine installed at the Delray Plant of The Detroit Edison Co. These castings were oil quenched and drawn subsequent to a normalizing treatment.

[18] See "Heat Treatment for Grain Size," by C. E. Sims, American Steel Foundries, *Metal Progress*, September, 1934.

secure a better balance of impact resistance, yield strength,[19] etc.

The practice of securing a "pilot" casting for thorough examination by X ray or gamma ray followed by subsequent sectioning for visual examination has been followed to some extent by the U.S. Navy Department. Conclusions from such tests[20] are that all castings poured to the same pattern by the same methods and given the same heat treatment will be very much alike. If a pronounced defect is found in one casting of a lot, there is apt to be the same defect in each of the others.

Standard specifications for carbon-steel castings for valves, flanges, and fittings for high-temperature service have been prepared under the procedure of the American Society for Testing Materials. These specifications under the ASTM Designation A95 are abstracted in Chap. IV. Similar specifications for alloy-steel castings under the ASTM Designation A157 cover the temperature range from 750 to 1100 F. Six ferritic casting alloys and two austenitic alloys are included in this latter specification. Minimum requirements as to tensile properties at room temperatures are included, subject to modification, if material is intended for welding.

The minimum tensile properties specified for some of these alloy cast steels are given in Table II, page 345, and in the abstracts of these specifications in Chap. IV.

Bolting Material.—Common practice with respect to bolting material used in piping which operates at low pressures and moderate temperatures permits the use of mild steel and, sometimes, wrought-iron bolts and nuts. These are usually made up with square heads and hexagonal nuts, the nuts being to American Standard Heavy Series dimensions in order to ensure adequate wrench grip and to give sufficient bearing over the ⅛-in. oversize bolt holes. Galvanized or cadmium-plated bolts and nuts are frequently used for flanged joints in underground piping, or a bitumastic preparation is sometimes applied if plain steel or iron bolts are used.

Steam plants and oil refineries operating at pressures in excess of 250-lb pressure and temperatures of 600 F and upwards have

[19] A good discussion of the properties of alloy-steel castings as affected by heat treatment is given in a paper on "Properties of Some Cast Alloy Steels," by T. H. Armstrong, *Trans. ASM*, March, 1935, p. 286.

[20] "Weight Reductions Emphasize Benefits of Radiographic Casting Exploration," by Lieutenant Commander E. B. Perry, *Iron Age*, Apr. 30, 1936, p. 33.

experienced considerable trouble with ordinary bolt material.[21] Instances have been reported where the metal became so brittle after a few months' service under these conditions, that the bolt heads would snap off when tapped with a hammer. Investigation usually showed that the material was mild-carbon steel, wrought iron, or screw stock containing high sulphur and phosphorus, and that unequal strains were set up in the bolt by forging the head.[22] For these reasons it was deemed advisable, in framing the American Steel Flange Standards, to recommend the use of heat-treated alloy-steel stud bolts with two hexagonal nuts.

Alloy-steel bolting material suitable for flanged joints for operating temperatures up to 750 F customarily is specified in accordance with the requirements of ASTM Specification A96.[23] The chemical composition of bolting covered by this specification other than phosphorus and sulphur is a matter of agreement between manufacturer and purchaser, the following types of alloy steels being recommended: nickel, chrome nickel, chrome vanadium, and chrome molybdenum.

Tentative specifications for alloy-steel bolting material for service at temperatures from 750 to 1100 F have been issued under the ASTM Designation A193. Several grades of bolting material are included, classified in accordance with chemical compositions and room-temperature physical properties. The steels are not necessarily all suitable for the full range of temperature up to 1100 F. But, if care is exercised in making selections for the particular service, these specifications will be found to facilitate greatly the purchase of these more or less special bolting materials.[24]

Owing to the fact that the nuts have internal threads engaging external threads on the studs and are, consequently, under less unit stress than the studs, it is generally believed that nuts made from ordinary screw stock having a sulphur content of not to exceed 0.15 per cent and a phosphorus content of not to exceed 0.05 per cent are satisfactory. This permits the use of commercial

[21] "Bolts for Use in Power Plant Construction," by William P. Wood, *Trans. ASME*, 1925, p. 747.

[22] "Investigation of Bolt Steels," by V. T. Malcolm and John Juppenlatz, *Trans. ASST*, Vol. XI, January–June, 1927, p. 177.

[23] See abstract of ASTM A96, Chap. IV, p. 537.

[24] For discussion of these bolting materials as well as fundamental data on selection and performance of nut and bolting materials, see article on "Nut and Bolting Materials for High Pressure and High Temperature Service," by H. A. Wagner, *Heating, Piping and Air Conditioning*, March, 1937, pp. 147–151.

nuts which are relatively easy to thread. Some engineers, on the other hand, feel that it is desirable to use a better grade of carbon-steel nuts having sulphur and phosphorus content below 0.045 and 0.05 per cent, respectively.[25] In either case it is customary to semifinish the nuts, which consists in machining the under-bearing surface and chamfering the upper corners. These nuts correspond to Class 0 and Class 1 of ASTM Specification A194.

Embrittlement of alloy bolting materials has been the subject of numerous investigations reported by English metallurgists.[26] In general, nickel-chromium (3–4 per cent nickel, 0.75 per cent chromium) steels have been found most susceptible to embrittlement as measured by reduction in impact strength. Lower nickel, plus additions of molybdenum or tungsten, has been found to give less susceptible steels. Occasional fracture of an alloy-steel bolt stud during assembly or disassembly is encountered with most of the alloys used for such purposes. But attention to grain size and care in manufacture of the steel apparently offset the tendency to embrittlement observed by these English investigators since nickel-chromium bolt steels of similar analysis made in this country have not become embrittled in prolonged service at 700 F.

Nuts suitable for use with these bolts are covered in ASTM Specification A194. Six classes of material for services varying in degree of severity are covered: Class 0 for service under the least exacting conditions; Classes 3 and 4 for the most severe service; and Classes 1, 2, and 2H for service conditions between these extremes.

While Class 1 carbon-steel nuts have been used satisfactorily on alloy bolt studs at temperatures up to 1000 F, instances have been reported of carbon-steel nuts deforming badly at such temperatures. Based on tests at 975 F, the report[27] of the (British) Pipe Flanges Research Committee concludes that alloy-steel nuts are preferable for such service.

[25] For a discussion of this point refer to the above-mentioned papers by Wood and by Malcolm and Juppenlatz.

[26] "The Effect of Time and Temperature on the Embrittlement of Steels," by A. M. McKay and R. N. Arnold, *Engineering*, Dec. 15, 1933, p. 647. "Embrittlement of Steels at High Temperatures," by H. A. Dickie, *Engineering*, Aug. 4, 1933, p. 108.

[27] "First Report of the Pipe Flanges Research Committee of the British Standards Institution," *Engineering*, Feb. 28, 1936, pp. 243–245, and Mar. 6, 1936, pp. 271–273.

Bolt studs are commonly made up in one of the three following designs (illustrated in Fig. 7); (1) threaded on ends with full-diameter shank in the middle, (2) threaded the full length, and (3) threaded on ends with shank turned down to a diameter equal to that at the root of the threads. Each design has certain advantages for particular operating conditions although cost of manufacture may offset the advantages, real or imagined, of one design over another.

Bolt studs threaded on the ends with full-diameter shanks have proved entirely satisfactory for power-plant uses at temperatures up to 750 F when used with flanges conforming to the American Steel Flange Standards. Where extremely rigid flanged connections are used, provided the operating temperature is low enough so that creep of the bolts is not the primary consideration, the greater flexibility of the full-threaded and turned-down shank bolts will somewhat reduce the stress caused by temperature difference and may be of advantage if the bolt stress approaches the yield strength of the bolting material.

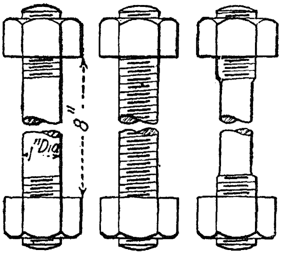


FIG. 7.—Bolt-stud designs.

For operating temperatures in the neighborhood of 1000 F where creep of the material is the important factor in joint design, full-shank bolts are preferable[28] because of the lesser length of bolt subjected to high stress and consequent higher rate of creep. The conclusion that full-shank bolts are superior to full-threaded or reduced-shank bolts for service where creep is of significance is supported through tests reported by the (British) Pipe Flanges Research Committee.[27]

Where working stress in the bolting is kept sufficiently low, however, the use of full-threaded bolts, as is almost standard practice in valve-bonnet joints, will give entirely satisfactory results. The preference for full-threaded bolt studs in valve-bonnet joints is based to a large extent on cost and manufacturing convenience since long bars of continuously threaded bolt stock are obtainable

[28] "Designing High Temperature Steam Piping," Part II, "Pipe Joints for 1000 F Service," by Arthur McCutchan, *Heating, Piping and Air Conditioning*, November, 1931, pp. 918–923.

from the bolt manufacturers. Under conditions producing fatigue failure, as in connecting-rod bolts of reciprocating engines, the use of reduced-shank bolts is supported by both tests and service experience, but in usual piping applications the greater resilience and lesser stress concentration obtained with such bolts are of little practical significance.[29]

Difficulty with nuts seizing on bolt threads after a period of service at high temperatures frequently has been encountered. The use of dissimilar materials or materials of different hardness for the bolt studs and nuts has tended to lessen this difficulty. In a series of tests of a number of bolt-stud and nut combinations,[30] it was found that a carbon-steel bolt and carbon-steel nut which could be loosened satisfactorily after a period at 392 F seized badly after the same period at 752 F. A chrome-molybdenum bolt with a nut of the same material seized after a short period at 572 F, while the same bolt stud with a carbon-steel nut gave no evidence of seizure at 932 F.

Forged Steel.—Forged-steel flanges, welding fittings, and valve bodies are used extensively in high-temperature services. Although valve bodies up to 6 in. have been forged, the general field of forged valves is in sizes 2 in. and smaller. In the low-pressure services, valves machined from bar stock have achieved considerable popularity. The grain refinement obtained by forging, plus the ability to inspect the billets and bars during the process of manufacture, is commonly considered to give a more reliable product than castings, although attention must be given to securing proper flow lines in the final forging if this condition is to be realized. Materials which forge easily do not in general have the best resistance to creep or flow at high temperatures. For this reason, somewhat heavier cast structures may be economically more desirable for severe high-temperature service.

Forged- or rolled-steel pipe flanges for service at temperatures up to 750 F are covered by ASTM Specification A105. For the temperature range 750 to 1100 F, reference should be made to ASTM Specification A182 for forged or rolled alloy-steel pipe flanges, forged fittings, and valves and parts. These specifications are abstracted in Chap. IV, page 348.

[29] "Effect of Screw Threads on Fatigue," by S. M. Arnold, *Mech. Eng.*, July, 1943, pp. 497–505.

[30] "Seizing of Screw Threads," *The Engineer* (Supplement) (London), April 24, 1936.

A comparison of the strength and rigidity of cast and forged flanges[31] under room-temperature conditions apparently has shown that cast flanges cup or deflect less easily than forged flanges. Since the modulus of elasticity, which is the relation between stress and deformation, of cast and forged material is not appreciably different, there should be no significant difference in the behavior of cast and forged flanges if the bolt loading is such that the yield strength of the flange material is not exceeded. In these particular tests the reported yield point of the cast material was approximately 40 per cent higher than the forged flange material which may account for the greater deformation found for the forged flanges.

Wrought-steel Pipe.—Some 10 specifications for wrought-steel pipe have been prepared under the jurisdiction of the American Society for Testing Materials. These specifications cover the entire range of pipe requirements from commercial steel pipe for ordinary uses, such as low-pressure steam, liquid, and gas lines, to alloy-steel pipe for service at temperatures from 750 to 1100 F.

These specifications embrace the different kinds of steel such as Bessemer, open-hearth, and electric furnace, the different grades of steel as defined by tensile strength, and the different processes of pipe manufacture, as, welded, seamless, electric-resistance welded, fusion welded, riveted, etc. The scope of these specifications is given in an abstract in Chap. IV covering the principal features of each specification.

Wrought Iron.—Genuine wrought iron has been used to a considerable extent for flange bolts and pipe under conditions of moderate temperature and pressure. As steel pipe has largely superseded wrought-iron pipe, it should always be specified whether wrought iron or steel is required when either would not be acceptable. "Wrought pipe" is a term used to designate a process of manufacture, as distinguished from cast pipe or spiral-riveted pipe, etc., and applies to both wrought iron and wrought steel. Hence, if wrought-iron pipe is the article actually wanted, it should be so described. Satisfactory properties, methods of manufacture, and tests for wrought-iron pipe and bolts are given in Chap. IV.

As in the case of pipe, the use of wrought-iron bolts has been largely superseded by the use of steel. The use of wrought-iron

[31] "Cast and Forged Flanges," by Lester Benoit and Raymond L. Collier, *Power*, March, 1937, p. 146.

pipe or bolts is not considered good practice for temperatures in excess of 500 F.

Seat Metals.—Among the principal factors influencing the performance of seat metals are: (1) tensile properties and chemical stability at the operating temperature; (2) hardness and toughness; (3) a coefficient of expansion which corresponds closely to that of the valve body; (4) enough difference in the properties of seat and disk facings to prevent seizing of their surfaces when one slides over the other. These factors are discussed on pages 331 to 333, and a table of physical properties and chemical compositions is given on page 346.

1. It is essential that a seat metal retain adequate tensile properties at the operating temperature and that it be immune from "growth" or other chemical change due to temperature. If it is to retain the smooth surface which is essential to tightness, a seat metal must resist oxidation and corrosion under service conditions. A metal such as brass or bronze, which may be entirely satisfactory for water or saturated steam, will not withstand superheated steam or oil temperatures of 500 F or more. For temperatures in the range 500 to 750 F, monel[32] metal (70 per cent nickel, 30 per cent copper) and what is sometimes termed "reversed monel" (70 per cent copper, 30 per cent nickel) have given good service. Modifications[33] of both these alloys having additions of tin, iron, zinc, chromium, silicon, or aluminum have been used up to 850 F with reasonably good success.

Straight chromium stainless steels (12 to 14 per cent chromium) have found wide acceptance as valve-trim materials. In a few cases austenitic stainless steels (18 per cent chromium, 8 per cent nickel) have been used where corrosive conditions are severe or for applications to which it seems particularly suited as in steam safety-valve nozzles. Nitralloy seats and disks have been used on valves for superheated steam and hot oil with good success. Some pitting has occurred where nitralloy has been used on intermittent saturated-steam service. For the more severe service conditions welded-on overlays of stellite or colmonoy (see

[32] Monel metal is the trade name for a product of the International Nickel Company, Inc. It originally was applied to the alloy made from an ore containing nickel and copper in approximately a 70–30 ratio but has been extended to cover modifications such as K-Monel, S-Monel, etc., which contain additions of aluminum, manganese, and silicon.

[33] Everbrite, Davis Metal, Platnam, Lunkenheimer NT7, Crane No. 49, are trade names for some of these modifications of monel or reversed monel.

Table III, page 346) hard-surfacing alloys have given excellent results.

2. Hardness and toughness are properties which tend to prevent wear caused by sliding one surface over the other and erosion due to cutting action of the fluid. The proper balance between hardness and toughness to secure good resistance to cutting and other abrading influences is dependent on a number of factors which differ with each material. Steam-impingement tests conducted on a large number of materials[34] tend to show that greater hardness does not necessarily indicate greater resistance to erosion. A forged 12 to 14 per cent chromium stainless steel, heat treated to give a Brinell hardness of 302, showed only about one-sixth the depth of erosion measured for the same material with a Brinell hardness of 444. Nitrided materials having an equivalent Brinell of 1,150 did not show up so well as the moderately hard stainless material. The cobalt-chromium-tungsten (stellite) deposited material likewise showed superior resistance in these steam-impingement tests.

3. Similar coefficients of expansion for seat metals and valve bodies are desirable in order to prevent loosening of the rings after repeated heating and cooling. Leakage around seat rings is one of the more common difficulties with valves on high-temperature service. Despite the fact that brass and bronze used for valve trim on cast-iron valves expands approximately 50 per cent more than cast iron, it is used quite successfully at temperatures up to 450 F. Under conditions of frequent heating and cooling, however, such rings occasionally loosen. The expansion of monel and modified versions of reversed monel used for valve trim is about 20 per cent greater than carbon cast steel, although the straight chromium stainless steels expand about 15 per cent less than the carbon- and low-alloy steels used for valve bodies. The austenitic stainless steels (18 per cent chromium, 8 per cent nickel) expand about one-third more than carbon- and low-alloy steels.

It is apparent from consideration of the coefficients of expansion of the foregoing seating materials that it is improbable that a screwed seat ring made of these materials will act as a unit with the valve body. The fact that screwed seat rings made of a modified version of reversed monel have performed reasonably satisfactorily on 850 F service must be explainable on the basis of

[34] "New Method of Studying Erosion Aids Selection of Valve Seat Materials," by F. R. Venton, *Heating, Piping and Air Conditioning*, January, 1937, p. 34.

ability of the rings to deform elastically and plastically to compensate for these differences in expansion.

In respect to similarity of expansion, screwed seat rings of nitrided steel or of carbon or alloy steel with welded-on overlays should be a better choice. The recent use of welded-in seat rings of course eliminates difficulties with leakage around the seat rings. The greater permanency of the welded-on-overlay seats makes provision for ease of replacement less necessary.

4. Galling or seizing occurs when certain metals are in sliding contact. The use of dissimilar materials for seats and disks, such as seats of a ferrous alloy and disks of a nickel-copper alloy, has been a common method of avoiding galling. The use of the same materials heat treated to give different hardness values also has been used successfully with the straight-chromium stainless-steel valve-trim materials. The development of free-machining stainless steels having sulphur and selenium additions has made it possible to use seats and disks of the same hardness with no tendency to seize in some applications. Nitrided seats and disks show no tendency to gall or seize at temperatures below about 900 F, although this trouble was experienced in an experimental installation at 1100 F. Because of its extremely low coefficient of friction and other properties peculiar to its composition, no evidence of galling has been found in the seats and disks faced with cobalt-chromium-tungsten welded-on overlays. The following composition[35] of stellite has been used extensively for welded-on overlays: cobalt, 60 per cent; chromium, 30 per cent; tungsten, 4 per cent. The physical properties and chemical analysis of the more common seat metals are given in Table III, page 346.

Stainless Steels.—Stainless steels are those steels which possess greater resistance to oxidation and corrosion than usually obtainable with ferrous materials. The straight chromium steels (12 to 16 per cent chromium) were originally developed for the cutlery trade. These steels are resistant to such substances as caustic soda, alkali ammonia, nitric acid, fresh and salt water, superheated steam, and the atmosphere. Various modifications of these 12 to 16 per cent chromium steels are used for valve trim, valve stems, and turbine blades.

An austenitic[36] type of stainless steel containing approximately 18 per cent chromium and 8 per cent nickel and popularly known

[35] "Federal Specification 10,300," *The Welding Journal*, September, 1936.

[36] Steels which are austenitic contain the carbon, chromium, nickel, and other

as "18–8" possesses even greater resistance to many corroding mediums than does the straight chromium type. In addition, it possesses unusual strength at high temperatures and unequaled shock resistance at extremely low temperatures. This material has been used for oil-refinery still tubes operating at temperatures of 1200 F and higher and for dewaxing coils at −70 F. Because of its resistance to atmospheric attack and its work hardenability which enable tensile strengths of 200,000 psi to be obtained in the cold-rolled condition, stainless steel has been used for structural parts of light-weight trains and airplanes.

The austenitic stainless steels have been used to a limited extent for steam piping and superheater tubing in power-plant work[37] at temperatures of 1000 to 1100 F. Although austenitic steels have given good service under these conditions, the high cost of the material has stimulated search for low-alloy substitutes having reasonably satisfactory properties at elevated temperatures. Under corrosive conditions as in the paper, oil, and food industries,[38] welded austenitic pipe must be heated as a unit to about 1900 F and quenched in water to restore its corrosion resistance. The addition of titanium or columbium is claimed to permit welding of this material without impairment of its corrosion-resistant properties.

Nitralloy or Nitrided Steel.—Nitralloy is the general name for those alloy steels which can be surface hardened by exposure to ammonia gas at high temperatures. Most of these steels contain approximately 1 per cent aluminum and 1½ per cent chromium, with additions of nickel or molybdenum for certain grades. The metal is forged or cast, and machined to its final shape before being placed in a closed retort where it is kept for 15 to 90 hours at 875 to 1200 F, the temperature depending upon the practice followed by the particular manufacturer. Anhydrous

---

elements in solid solution in the gamma iron. Austenitic alloys are characterized by low heat conductivity, comparatively high coefficients of expansion, non-magnetic properties, inability to harden by heat treatment, good work-hardening properties, and satisfactory toughness.

[37] Discussion by Sabin Crocker of paper on "High-temperature Steam Piping," by F. W. Martin, 1931 *Symposium on Effect of Temperature on Properties of Metals*, p. 61. Also paper on "High-temperature Steam Experience at Detroit," by P. W. Thompson and R. M. Van Duzer, Jr., *Trans. ASME.*, Vol. 56, pp. 497–506, 1934.

[38] "Piping 18-8," by A. W. Beatty and A. P. McCurdy, Pittsburgh Piping & Equipment Co., *Steel*, Mar. 4, 1935, p. 38.

ammonia gas is circulated over the material and dissociates into hydrogen and nitrogen at this temperature. The nitrogen unites with the surface layer constituents of the steel, forming an extremely hard case, principally of aluminum and chromium nitrides.

The depth of case varies from 0.010 to 0.030 in., about one-half of which is of extreme hardness and clearly visible in the fractured sample. The hardness decreases gradually until the case merges into the core without any line of demarcation. The hardness imparted to the outer surface by the foregoing treatment will scratch glass with ease, and even special testing files are worn smooth without effect on this extremely hard surface.

The composition and physical properties reported for valve-seat rings furnished for high-pressure feed-water service are given below:

Composition

Carbon, percentage	0.30 to 0.40
Manganese, percentage	0.40 to 0.60
Silicon, percentage	0.30 max
Aluminum, percentage	0.75 to 1.25
Chromium, percentage	1.00 to 1.50
Molybdenum, percentage	0.15 to 0.25

Physical Properties

Tensile strength, psi	138,000
Yield point, psi	110,000
Elongation, percentage	4
Reduction of area, percentage	17
Brinell number	1,000

Material for nitriding may be quenched and drawn at different temperatures to secure desired hardness and toughness of the core. The nitriding treatment takes the place of the drawing in some instances.

Nitrided bushings in combination with various alloy stem materials are used quite generally on turbine control valves. Wear, galling, and seizing tests have demonstrated the superiority of nitrided bushings for this service. While nitralloy has been used successfully on superheated-steam and hot-oil lines, excessive pitting has taken place in some instances on saturated-steam and boiler-feed service.

SAFE WORKING STRESS AT ELEVATED TEMPERATURES

Design stresses for carbon and alloy steels used in pipe, valves, and fittings for service at temperatures up to about 750 F usually are chosen as some fraction of the room-temperature tensile or yield strength. For a rational basis in the selection of design stresses above this temperature reference should be made to the results of long-time tensile or "creep" tests.

Creep is defined as the increase in deformation with time which occurs at elevated temperatures when a material is subjected to stress. Creep at moderately elevated temperatures is regarded as the result of a running balance between yielding of the material and the strengthening effect of strain hardening, while the rate of creep at the higher temperatures is a compromise between the rate of strain hardening and the rate of recrystallization. The load which a given material can support over an extended period at elevated temperatures in general can be predicted with assurance only from prolonged service experience or comprehensive long-time creep tests.

Utilization of Creep Data.—A digest of mathematical and engineering theories and experiments bearing on the utilization of creep data was presented by J. J. Kanter as part of the Symposium on the Plastic Working of Metals.[39] The author pointed out that the acceptance of creep as a factor in the design of parts for high-temperature service necessitated a critical interpretation of laboratory-test results and that working stresses should be selected with due regard for the total deformation permissible for a given service.

The list of references appended to that paper, 59 in number, affords ample evidence of the importance of the subject.[40]

Correlating Creep Data with Experience.—In the selection of pipe for high-temperature service, the primary consideration

[39] "Interpretation and Use of Creep Results," by J. J. Kanter, *Trans. ASM*, December, 1936, p. 870.

[40] Additional references on utilization of creep data are:

(*a*) "The Creep Curve and Stability of Steels at Constant Stress and Temperature," by S. H. Weaver, *Trans. ASME*, Vol. 58, RP-58-16, pp. 745–751, November, 1936.

(*b*) "The Interpretation of Creep Tests for Machine Design," by C. Richard Soderberg, *Trans. ASME*, Vol. 58, RP-58-15, pp. 733–743, November, 1936.

(*c*) "Design of Members Subject to Creep at High Temperatures," by Joseph Marin (*Journal of Applied Mechanics*), *Trans. ASME*, Vol. 4, p. A21, March, 1937.

is to keep the hoop tension or bursting stress low enough to avoid excessive thinning of the pipe wall which accompanies increase in diameter under the internal-pressure loading. Where sufficient high-temperature experience and creep data are available, as in the case of carbon steel, it has been possible to establish reasonably satisfactory values of working stress for temperatures up to about 900 F by reducing the allowable stress at 750 F in proportion to the reduction in creep strength.

This procedure, with minor modifications, was followed in arriving at the allowable stress values for steel pipe given in the Code for Pressure Piping, which are abstracted in Table V, page 43, for the Power and District Heating Sections. Corresponding values for the other code sections will be found in the respective chapters of this handbook. In many cases this method of correlating creep measurements with the background of experience has proved more workable than direct application of creep results to design. The values of allowable stress given in Table V are intended to be used in selecting pipe wall thicknesses for the various service conditions indicated. The formulas to be used in determining pipe wall thickness, with allowances for corrosion, threading, etc., are given in Chap. II, page 42. In determining allowable S values for temperatures of 750 F and lower, factors of safety and joint efficiency have been adjusted to suit service conditions involved.

Deformation before Fracture.—Under prolonged loading at elevated temperatures some materials fracture with only slight display of ductility. As deformation of the material affords the only information as to impending failure, it is desirable that the material should possess reasonable ductility under slow fracture as evidenced in creep-test loading. For example, a low-carbon steel tested at 1000 F which gave an elongation of 42 per cent in 2 in. after fracture in a short-time tensile test at that temperature showed an elongation of only 11 per cent after fracturing in a 14,000-hr creep test. In this case the observed embrittlement was attributed to intercrystalline oxidation. This tendency to fracture with little evidence of ductility is characteristic of carbon-molybdenum steel.[41]

[41] See "The Properties and Mode of Rupture of a Molybdenum and a Molybdenum-vanadium Steel Judged from Prolonged Creep Tests to Fracture," by H. J. Tapsell, C. A. Bristow, and C. H. M. Jenkins, *Proc. Inst. Mech. Engrs.* (British), 1941, Vol. 146, No. 5, pp. 208–222, published in March, 1942, journal.

Creep Characteristics.—A paper on "Creep Characteristics of Metals," by C. L. Clark and A. E. White, presented as part of a symposium on plastic working of metals, summarizes the present state of knowledge of creep phenomena.[42] The following discussion of creep is largely abstracted from that paper. Figure 8 illustrates the relation between deformation and time observed during three principal stages of creep. When a stress is applied to a steel at elevated temperatures, a deformation *OA* immediately

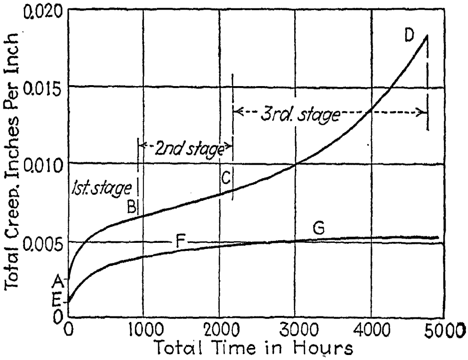


FIG. 8.—Typical creep curves.

results. This deformation may be entirely elastic, or elastic plus plastic, depending upon the material, the temperature, and the stress. Under suitable combination of stress and temperature this deformation increases as the time of load application is extended, following a curve such as *ABCD*. This increase in deformation with time under a constant stress represents the most common type of creep determination.

As indicated in Fig. 8, the complete creep curve consists of three stages. In the first stage creep occurs at a decreasing rate, in the second at an approximately constant rate, and in the third stage at an increasing rate. In the majority of creep tests the stress and temperature conditions are selected so that the third stage is not entered and the curve corresponds to the lower curve *EFG* of Fig. 8. In general it is found that the duration of each stage tends to vary inversely with the stress.

[42] *Trans. ASM*, December, 1936, p. 831. Also paper on "The Rupture Strengths of Steels at Elevated Temperatures," by A. E. White, C. L. Clark, and R. L. Wilson, *Trans. ASM*, September, 1937, p. 863.

On the basis of data presented in that paper and a thorough study of the extensive literature on the subject of creep, these investigators conclude that the ability of a metal to resist creep may be greatly influenced by factors some of which have little or no effect on the physical properties at room temperature. The manner in which these factors influence creep is discussed below.

Chemical Composition.—In the lower temperature ranges creep resistance may be increased by the addition of elements which enter into solid solution with the ferrite, such as nickel and manganese, as well as by the addition of chromium, tungsten, molybdenum or vanadium which unite with the carbon to form carbides. These carbides form both within the crystals and in the amorphous or severely strained crystalline matter between the crystals. The carbides within the crystals are considered to have a keying action which decreases the ease of slippage or creep along the slip planes of the crystals.

In the higher temperature ranges only the elements forming carbides are effective and only the carbides located in the matter between the crystals are considered to be involved, as it is felt that creep at the higher temperatures occurs largely by movement of crystals with respect to one another, rather than by slippage along planes within the crystals. Creep resistance of a material at the lower temperatures thus depends primarily upon cohesion across the surfaces within the crystals, while creep at the higher temperatures depends upon cohesion across the bounding surfaces of the crystals.

Equicohesive Temperature.—The temperature at which these two cohesions are of equal strength is termed the equicohesive temperature. This temperature varies for each material and for each condition of the material. Although it cannot be determined directly, it is generally considered to coincide with the temperature at which a cold-worked material begins to recrystallize. This temperature is believed to be of fundamental importance as far as creep behavior is concerned. At temperatures below the equicohesive temperature cold working or overstraining the material results in a change in the crystal orientation which makes it more and more difficult for slip to occur. At temperatures above the equicohesive temperature the strain-hardening effect of slippage in the material between the crystals is continuously destroyed by the formation of new crystals, and creep represents a compromise between strain hardening and the rate of recrystallization.

Structural Stability.—The structure of a material may be defined as the shape, dimensions, nature, and arrangement of its constituent parts. The greater the resistance to change of any of these factors, the greater the structural stability. In general it may be stated that the greater the structural stability, the greater will be the creep resistance. It is questionable, however, whether any type of structure possesses complete stability at the more elevated temperatures.

One of the more common changes of structure under prolonged exposure to high temperature below the lower critical temperature[43] is a change in the form of the carbides from elongated bodies or layers as they exist in normal pearlitic steels to rounded bodies. This change, usually associated with a reduction in creep resistance, is termed "spheroidization." The rapidity and degree of spheroidization are dependent principally upon the temperature. The higher the temperature, if below the lower critical temperature, the less time required to effect a given degree of spheroidization.

Structural Uniformity.—Nonuniform distribution of the various constituents due to segregation, banding, and the presence of nonmetallic inclusions will in general decrease the creep resistance, since the strength of the metal as a whole will be affected by that of the weakest portion.

Heat Treatment.—Creep is influenced by heat treatment in so far as it affects the uniformity and stability of the structure and the grain size. For the higher operating temperatures, annealing is considered to give a more stable structure. However, normalizing followed by a draw at a temperature about 200 F above the operating temperature will, in general, produce greater creep resistance. Quenching in water or oil, followed by a draw about 200 F above the operating temperature, may be used but is not considered to give either as good creep resistance or as great structural stability.

Grain Size.—The influence of grain size on creep resistance depends to a considerable extent on whether the temperature involved is below, or above, that at which the material begins to recrystallize after work hardening. The fine-grained steels possess the greater resistance when slip occurs through the crystals or grains, as is considered to be the mode of slippage at the lower temperatures, while at temperatures above the equicohesive

[43] NOTE.—The lower critical temperature is the temperature at which the structure of the steel begins to change from pearlite to austenite upon heating.

temperature, where grains are considered to slip over each other, the coarse-grained steels are superior.

Method of Manufacture.—Creep resistance is likewise influenced by the melting process, melting practice, and casting conditions. Since the superiority of any given process may be obscured by variations in other factors mentioned, no hard and fast rules can be given, although in general fully deoxidized or "killed" steel is found to be superior to the open or rimmed type.

Previous Deformation.—Initial cold deformation, such as cold rolling or cold drawing, affects the temperature at which recrystallization begins as well as the creep resistance at the more elevated temperatures. Under prolonged testing or service periods, the effect of initial cold deformation probably is minimized.

Oxidation Resistance.—Under prolonged test or service in an oxidizing atmosphere, general oxidation of the surface as well as intergranular oxidation may have a profound effect on the load-carrying ability of a material. The occurrence of intergranular oxidation makes the extrapolation of creep data extremely hazardous. For this reason the susceptibility of a given steel to intergranular oxidation under the particular operating condition should be taken into account. General resistance to oxidation gives some indication of the relative susceptibility of a material to intercrystalline oxygen attack (see Chap. XVII on Corrosion).

COMPARATIVE OXIDATION RESISTANCE OF STEELS:[44] LOSS OF WEIGHT, GRAMS PER SQUARE INCH OF EXPOSED SURFACE, IN AIR

	1000 F	1250 F	1500 F
Low-carbon steel	0.07	1.92	3.81
Carbon-0.50 Mo	0.08	2.00	3.95
4-6 Cr-0.50 Mo	0.04	1.06	2.60
18Cr-8Ni	0	0	0

The above values were obtained on samples heated in air for 1,000 hr in an electric furnace. Somewhat greater scaling was observed in an atmosphere of products of combustion containing CO_2 and water vapor. The behavior of various alloys when exposed to steam has been investigated by others.[45]

[44] Bulletin on "Properties of Steel Pipe and Tubes Applicable to High Temperature and High Pressure Service," National Tube Company, 1936.

[45] (*a*) "Investigation of the Oxidation of Metals by High-temperature Steam," by A. A. Potter, H. L. Solberg, and G. A. Hawkins, *Trans. ASME*,

Reliability of Creep Data.—As the factors which influence creep have become better appreciated, there has been an increased reluctance on the part of creep investigators to attempt to predict the behavior of materials for service periods of 10 to 20 years. Some even hesitate to make predictions beyond the period of actual test.

In 1936 ASTM Committee A1 on Steel published creep data in pamphlet form (entitled "Data on Nominal Creep Strength of Steels at Elevated Temperatures") for several steels covered by the specifications of its Subcommittee XXII for High-temperature Piping Materials. Since in many cases information was meager as to the full characteristics and heat history of the material tested and the creep rates observed by different reputable laboratories were at wide variance, the subcommittee deemed it inadvisable to recommend limiting creep stresses for these materials and confined its report merely to presenting a compilation of the laboratory data, leaving it to the individual designer to set his own allowable stresses. The following quotation from that report is of interest in setting such stresses:

> Until more is known of the mechanism of creep and until a much larger body of test results is available, the committee is of the opinion that a simple compilation of results is all that should be presented at this time. To restrict utilization of all material to conditions based on the lowest value would be to throttle progress, while to authorize utilization in accordance with the highest value without caution would be foolhardy. However, the committee feels that if due caution is exercised with respect to the possible variations in strength, the results presented may form a useful means of comparison.

The above expression of opinion by the committee is regarded by the authors as a fair caution concerning most existing creep data.

Need for Acceptance Test.—Since it is impracticable to make a long-time creep test on each heat of steel or heat-treatment charge intended for service at high temperatures, there is a widespread search for some method of test which will assure that a particular lot of a given material has creep properties consistent with those determined for material conforming to the same specifications in regard to chemical composition, room-temperature physical properties, and heat treatment. In general, the require-

---

Vol. 59, pp. 725–732, 1937. Also, same authors in Vols. 64 and 65.

(*b*) "High-temperature-steam Corrosion Studies at Detroit," by I. A. Rohrig, R. M. Van Duzer, Jr., and C. H. Fellows, *Trans. ASME*, Vol. 66, No. 4, pp. 277–290, May, 1944.

ments for such a test are that it can be run within 24 hr, that it will detect off-color heats of steel, and that the conditions of test, reproducibility of results, and correlation with creep data are such that mutual agreement can be reached between the manufacturer and purchaser. Metallographic examination to determine grain size and condition of carbides has possibilities as a means of securing material with consistent high-temperature properties.

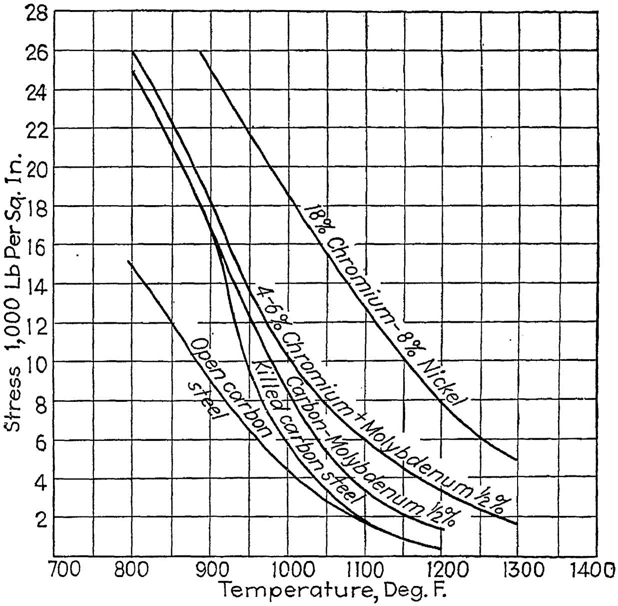


FIG. 9.—Creep properties of pipe steels, stress causing elongation at the rate of 1 per cent in 10,000 hr.

Representative Creep Values.—Values of stress which may be expected to cause an elongation of 1 per cent in 10,000 hr are plotted on Fig. 9. These values are of direct application to oil piping where creep of the order of 1 per cent in a year is considered permissible. For power-plant piping, periods of 15 to 20 years are the accepted basis of design, and a creep rate of 1 per cent in 100,000 hr is the more usual value considered. Since, however, values for 1 per cent in 100,000 hr are based entirely on extrapolation from tests of a few hundred or few thousand hours, it appeared

Table I.—Elastic and Physical Constants for Metals and Alloys Used for Pipe, Valves, and Fittings[1]

Material	Spec. grav. relative to water	Weight, lb per cu ft	Weight, lb per cu in.	Specific heat, Btu per lb per deg F	Melting point, deg F	Poisson's ratio, λ	Modulus of elasticity in 1,000,000 psi: Tension and compression, E	Modulus of elasticity in 1,000,000 psi: Shear and torsion, G
Aluminum	2.56	160	0.093	0.218	1220	0.333	10	3.8
Brass (copper-zinc, etc.)	8.59	535	0.310	0.094	1700	0.333	14	5.2
Bronze (copper-tin, etc.)	8.59	535	0.310	0.104	1700	0.333	14	5.2
Colmonoy No. 6	6.65	415	0.240	.....	1925			
Copper	8.93	558	0.323	0.093	1980	0.333	16	6.0
Davis metal	8.75	546	0.316	.....	2250	.....	24	
Everdur 1000	8.15	509	0.295	.....	1830	.....	15	
Hastelloy C	8.94	558	0.323	.....	.....	.....	28	
Inconel	8.55	534	0.307	0.109	2540	.....	31	
Iron:								
Gray cast iron	7.22	450	0.260	0.130	2200	0.270	12	4.5
Semisteel	7.30	456	0.264	0.120	2300	0.270	15	6.0
Malleable iron	7.40	461	0.267	0.122	2400	0.265	24	11
Duriron	7.00	437	0.253	0.120	2300			
Ni-resist	7.55	471	0.273	.....	2200	.....	18	
Wrought iron	7.70	480	0.280	0.114	2740	0.278	28	12
Lead	11.4	710	0.411	0.031	620	0.450	2.5	0.8
Magnesium	1.74	108	0.063	0.248	1200	0.330	6.5	2.5
Mercury (60 F)	13.6	847	0.490	0.032	−39.5			
Monel	8.84	551	0.319	0.127	2400	0.315	26	9.5
Nickel	8.90	555	0.321	0.130	2625	0.310	30	11
Nitralloy, N125	7.85	490	0.283	0.117	2500	0.303	30	12
Solder:								
Plumbers	10.0	624	0.361	0.043	360			
Silver (Sil-fos)	8.80	550	0.318	0.091	1200			
Steel:								
Carbon steel	7.85	490	0.283	0.117	2500	0.303	30	12
Carbon, ½ Mo	7.85	490	0.283	0.117	2500	0.303	30	12
5% Cr, ½ Mo	7.85	490	0.283	0.117	2550	0.303	28	11
12% Cr	7.66	477	0.276	0.110	2750	0.303	28	11
2–5% Ni	7.85	490	0.283	0.117	2500	0.303	29	11
18 Cr, 8 Ni	7.93	495	0.286	0.125	2600	0.305	29	11
Stellite No. 6	8.38	523	0.303	0.101	2327	.....	30	
Tin	7.35	458	0.265	0.055	450	0.330	8.0	2.9
Zinc	7.15	447	0.259	0.094	787	0.300	13	5.0

[1] Average values at atmospheric pressures or temperatures. Owing to commercial variations in materials about which specific information is lacking, drawing fine distinctions between the constants for slightly different compositions or between processes of manufacture, such as cast, forged, rolled, or wrought, is not warranted.

For chemical analysis, tensile strength, Brinell hardness, etc., see Tables II and III. For coefficient of expansion and modulus of elasticity at elevated temperatures, see Tables I and II, pp. 756–758, and Fig. 39, p. 830, respectively.

desirable to limit the period covered by these curves to that approximated by actual tests.

Where it is desired to reduce the rate of elongation to 0.1 per cent in 10,000 hr, or its numerical equivalent one per cent in 100,-

TABLE II.—PHYSICAL PROPERTIES (MINIMUM) OF COMMON PIPING MATERIALS AT ATMOSPHERIC TEMPERATURE[1]

Material	ASTM or other designation	Tensile strength, psi	Yield strength, psi	Elongation		Reduction in area, %	Brinell hardness number, range
				Per cent	Gauge length		
Bolting:							
Carbon steel, hot-rolled, Grade 22	A107	60,000[2]	35,000[2]	25[2]	2[2]	50[2]	110–125[2]
Heat-treated	A261	100,000	75,000	16	2	45	200–260
Wrought-iron commercial	A189	48,000	28,800	25	8	40	90–110[2]
Alloy steel, Class *A*	A96	95,000	70,000	20	2	50	190–250
Class *B*	A96	105,000	80,000	20	2	50	210–270
Class *C*	A96	125,000	105,000	16	2	50	260–320
Grade *B7*	A193	125,000	105,000	16	2	50	260–320[2]
Grade *B8*	A193	75,000	30,000	35	2	50	135–185[2]
Grade *B14*	A193	125,000	105,000	16	2	50	260–320[2]
Nuts:							
Carbon steel, Class 0	A194	....	....	..	..	..	120 min
Class 1	A194	....	....	..	..	..	120 min
Class 2*H*	A194	....	....	..	..	..	248–352
Cast brass and bronze:							
Red brass or ounce metal	B62	30,000	14,000	20	2	15[2]	50–60[2]
Steam metal or valve bronze	B61	34,000	16,000	22	2	15[2]	52–62[2]
Hard bronze, 88-8-4	B60	38,000	16,000	22	2	25[2]	55–62[2]
No. 1 Manganese bronze (valve trim)	B147	65,000	25,000	25	2	20[2]	110–120[2]
Leaded yellow brass, Class *A*	B132	60,000	20,000	20	2	15[2]	80–100[2]
Class *B*	B132	80,000	32,000	12	2	10[2]	140–170[2]
Cast-iron valves, flanges, fittings, or pipe:							
Regular gray iron, Grade *A*	A126	21,000	....	..	..	..	130–160[2]
Semi steel, Grade *B*	A126	31,000	....	..	..	..	160–200[2]
Alloy or regular, Grade *C*	A126	41,000	....	..	..	..	200–250[2]
Gas pipe and fittings, pit-cast	AGA	18,000	....	..	..	..	130–160[2]
Soil pipe	A74	21,000	....	..	..	..	130–160[2]
Water fittings	AWWA	20,000	....	..	..	..	130–160[2]
Water pipe, ASA pit-cast	A21.2	11,000[3]	....	..	..	..	130–160
Centrifugally cast	18/40	18,000[3]	....	..	..	..	160–200
Centrifugally cast	25/50	25,000[3]	....	..	..	..	160–200
Malleable iron:							
Cupola	A197	40,000	30,000	5	2	..	110–145[2]
Furnace, Grade 32510	A47	50,000	32,500	10	2	..	110–145[2]
Cast-steel valves, flanges, and fittings:							
Carbon steel, flanged	A95	70,000	36,000	22	2	30	140–160[2]
welding, Grade *WCB*	A216	70,000	36,000	22	2	35	130–150[2]
Carbon molybdenum, Grade *C1*	A157	70,000	45,000	22	2	35	150–175[2]
welding, Grade *WC1*	A217	70,000	45,000	22	2	35	140–160[2]
Nickel-chrome-molybdenum, Grade *C4*	A157	100,000	65,000	18	2	30	225–250[2]
Grade *WC4*	A217	80,000	55,000	20	2	35	180–200[2]
4–6% chrome, Grade *C5A*	A157	90,000	60,000	18	2	30	200–220[2]
3% nickel (low temperature)	.....	67,500	45,000	25	2	45	140–160[2]
18% chrome, 8% nickel, Grade *C9*	A157	70,000	30,000	35	2	40	135–185[2]
Forged steel valves, flanges, and fittings:							
Carbon steel, as forged, Grade I	A181	60,000	30,000	22	2	35	115–135[2]
Grade II	A181	70,000	36,000	18	2	24	140–160[2]
heat-treated, Grade I	A105	60,000	30,000	25	2	38	115–135[2]
Grade II	A105	70,000	36,000	22	2	30	140–160[2]
Carbon molybdenum, heat-treated, Grade *F1*	A182	70,000	45,000	25	2	35	150–175[2]

[1] For chemical composition, etc., and for the properties of wrought pipe, s abstracted in Chap. IV.
[2] Not included in specification indicated, but approximately true.
[3] Bursting tensile strength computed by "common formula for bursting pressure," see p. 33.

000 hr, the National Tube Company *Bulletin*,[44] from which these curves (except that for killed carbon steel) were taken, suggests that the corresponding stress for creep at the rate of 1 per cent in 100,000 hr may be considered as 60 per cent of the values given on Fig. 9.

The curve for killed carbon steel above 900 F was taken from a bulletin issued by The Timken Steel and Tube Company.[46] Although it is believed that the values obtained by applying a factor of 60 per cent to the curves of Fig. 9 are ultraconservative for some of the materials, they correspond reasonably well with the values of allowable stress given in Tables V and VI on pages 43 and 44.

TABLE III.—SPECIAL ALLOYS FOR VALVE TRIM OR CORROSION RESISTANCE

Section A.—Minimum physical properties at atmospheric temperature

Trade name	Tensile strength, psi	Yield strength, psi	Elongation in 2 in., %	Reduction of area, %	Brinell hardness number range, 10-mm ball	Rockwell hardness number range, *C* scale
Aluminum, cast	12,000	....	40	..	15–20	
Colmonoy No. 6, cast	55,000	....	..	..	600	55–60
Davis metal, cast	67,000	40,000	15	..	120	
Everbrite, cast	75,000	45,000	14	..	160–180	
Everdur 1000, cast	50,000	20,000	20	25	70–90	
Hastelloy C, cast	72,000	45,000	10	11	175–215	
Inconel, wrought, annealed	80,000	25,000	35	60	110–170	
Lead	1,900	700	50	..	4.5–6	
Monel, cast	65,000	33,000	25	..	125–150	
Monel, silicon, cast	90,000	70,000	..	..	280–325	
Nickel, cast	55,000	20,000	15	..	100–130	
Ni-resist cast iron	25,000	....	2	..	120–170	
Nitralloy, N125, core	130,000	115,000	18	54	Surface 1000, core 250	
Platnam metal, cast	72,000	60,000	..	..	225	
Silicon cast iron (Duriron, etc.)	17,000	....	..	..	....	52
Stellite No. 6, cast	105,000	....	1	..	....	30–40
Stainless steel:						
AISI Type 410, annealed	65,000	35,000	25	60	135–165	B75–B85
Hardened	100,000	60,000	10	25	290–400	
AISI Type 420, annealed	85,000	40,000	25	40	170–200	
Hardened	225,000	185,000	3	5	450–550	
AISI Type 440, annealed	90,000	40,000	20	40	170–210	
Hardened	250,000	190,000	1.5	2	500–600	
Tin	2,000	200	..	..	4–6	
Zinc, annealed	18,000	....	30			

[46] "Digest of Steels for High-Temperature Service," The Timken Steel and Tube Company, November, 1934.

TABLE III.—SPECIAL ALLOYS FOR VALVE TRIM OR CORROSION RESISTANCE.—(*Concluded*)

Section B.—Chemical Analysis (Percentage by Weight)

Material	Essential Nominal Chemical Composition,[1] Per Cent
Aluminum	Al, 98-99
Colmonoy No. 6	Ni, 68; Cr, 18; B, 4; Fe, C, Si, 10 max
Davis metal	Cu, 67; Ni, 27.5; Fe, 4; Mn, 1.5; C, Si, 0.5
Everbrite	Cu, 60; Ni, 30; Fe, 3; Si, 3; Cr, 3
Everdur 1000	Cu, 94.9; Si, 4; Mn, 1.1
Hastelloy C	Ni, 60; Mo, 17; Cr, 15; Fe, 8
Inconel	Ni, 80; Cr, 13; Fe, 6.2; Mn, 0.25; Si, 0.25; Cu, 0.2; C, 0.1
Lead	Pb, 99.8+
Monel	Ni, 67; Cu, 30; Fe, 1.5; Si, 1.25; Mn, 0.25
Monel, silicon	Ni, 64; Cu, 29; Si, 3.75; Fe, 2.5, Mo, 0.5; C, 0.1
Nickel, cast	Ni, 96.7, Si, 1.5; Fe, 0.5; C, 0.5, Mn, 0.45
Ni-resist cast iron	Fe, 71; Ni, 14; Cu, 6; C, 3; Cr, 2.5; Si, 1.75; Mn, 1.25; S, 0.1 max
Nitralloy, N 125	Fe, 96.5; Cr, 1.15; Al, 1.15, Mn, 0.5; C, 0.25; Si, 0.25; Mo, 0.2
Platnam metal	Ni, 54; Cu, 30; Sn, 15; Fe, 1
Silicon cast iron (Duriron, etc.)	Fe, 84; Si, 14; C, 0.8; Mn, 0.4
Stellite No. 6	Co, 60; Cr, 30; W, 4; Fe, Mn, Si, C, 6
Stainless steel:	
AISI Type 410	Cr, 12; C, 0.15 max
AISI Type 420	Cr, 13; C, over 0.15
AISI Type 440	Cr 16; C, over 0.12
Tin	Sn, 99.99+
Zinc	Zn, 99+; Pb, 0.12

[1] KEY.—Al, aluminum; B, boron; C, carbon; Co, cobalt; Cr, chromium; Cu, copper; Fe, iron; Mo, molybdenum; Mn, manganese; Ni, nickel; Pb, lead; S, sulphur; Si, silicon; Sn, tin; W, tungsten; Zn, zinc.

CHAPTER IV

PIPE, VALVES, AND FITTINGS

STEEL PIPE

The bulk of wrought tubular products used in industry at the present time for conveying fluids is made from steel produced in open-hearth furnaces and, to a diminishing extent, in Bessemer converters. Duplexed and electric-furnace steels are used to a limited extent in the manufacture of wrought pipe. The designation "wrought" is used to distinguish from cast pipe, steel, iron, copper, or brass pipe which is manufactured by a rolling or drawing process. The term "pipe" is restricted to apply to the dimensions commonly used for pipe lines and connections, formerly known as "Iron pipe size" (IPS). The material, diameter, wall thickness, and quality of pipe required largely determine the manufacturing process which is employed to produce pipe for a given class of service. Numerous processes of manufacture have come into use through the years, among which the most common are: furnace welding, both butt and lap; seamless; electric resistance welding; electric-fusion welding; and spirally wound plate with the edges joined by either riveting or electric-fusion welding.

MANUFACTURE OF FURNACE-WELDED STEEL PIPE

Butt Welded.—Pipe 3 in. in diameter and smaller for ordinary uses, such as low-pressure steam, liquid, or gas lines, may be made by the butt-welding process from Bessemer steel, which is used because of its low cost and superior furnace-welding characteristics imparted by its high phosphorus content. Rephosphorized open-hearth steel has been used to a limited extent. In the butt-welding process a strip or plate termed "skelp," which has been heated in a furnace to welding heat, is pulled through a ring die or bell at welding heat, the tube being formed and the edges welded together in a single operation (Fig. 2). In the continuous butt-weld process the end of one strip is resistance-welded to the

next coil. A second pass through a welding roll is used in some cases to ensure more complete welding and more accurate finished dimensions of the pipe.

Butt-welded steel pipe may be purchased under ASTM Specifications A53 and A120, and API Specification 5L for line pipe. These specifications are much alike except that no physical tests of the material are required by A120. All three specifications require a hydrostatic test of each length of pipe.

Lap Welded.[1]—Pipe 4 in. in diameter and larger for ordinary purposes is made to a large extent by the lap-welding process, although considerable fusion-welded, electric-resistance-welded, and seamless pipe are used also for applications to which they are peculiarly suited. Open-hearth steel is commonly used for lap

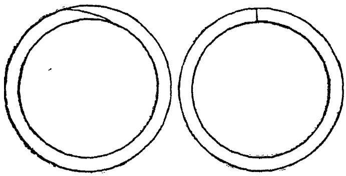


Fig. 1. Fig. 2.
Fig. 1.—Lap-welded pipe.
Fig. 2.—Butt-welded and electric-resistance-welded pipe.

welding, but Bessemer steel is used to some extent also, especially in the smaller sizes. Although the "rimmed" variety of open hearth usually is considered good enough for the purposes to which lap-welded pipe will be put, "killed" steel of a better grade sometimes is used where service conditions warrant.

As implied by the name, lap-welded pipe is made from flat plate or skelp with scarfed edges which is heated and bent to tubular form with the edges overlapping (Fig. 1). Following this, the rolled-up plate is reheated to welding heat and welded together between a mandrel inside the pipe and the two grooved welding rolls. Subsequent rolling operations are employed to straighten and finish the pipe.

Lap-welded steel pipe may be purchased under ASTM Specifications A120, A53, and A106, and API Specification 5L for line pipe. The usual lengths are 16 to 22 ft.

[1] "The Manufacture of Steel Tubular Products," National Tube Co., *Bull.*, 1931; "Seamless Steel Pipe," Babcock and Wilcox, *Bull.*, 1927; *Bull.* A1A, File 29B8, Republic Steel Corp., 1930; "Steel Products Manual," Sec. 18 on Steel Tubular Products, American Iron and Steel Institute, see latest issue.

MANUFACTURE OF SEAMLESS PIPE

Seamless pipe may be made from a large variety of carbon and alloy steels. Three processes of making seamless pipe are employed. The Mannesman plug-mill process is used almost exclusively in this country for high-quality seamless pipe intended for power-plant use. The Pilger process has wide application to commercial thin-walled pipe for cross-country oil and gas lines, oil-well casing, and the like. Seamless pipe made by the plug-mill process also is used for these latter services while Pilger pipe has been used for power-plant purposes abroad and to a limited extent in this country. The cupping process is used for odd sizes and for small orders.

Plug-mill Process.—In the Mannesman plug-mill process a round billet of high-quality killed steel is heated to a high forging temperature and forced by revolving rolls over the rounded nose of a piercing mandrel, resulting in a thick-walled seamless tube. In some cases, a second piercing mandrel is used to expand the tube to a greater diameter before passing to the rolls. A plug or "ball" is inserted in the pierced hole and the tube passed through a series of rolls that reduce the wall thickness and elongate the tube. The tube is passed through cross rolls which straighten and true up the surface of the pipe. The pipe is then reheated and passed through sizing rolls. If thin-walled pipe larger than about 16 in. is desired, a rotary expander is used immediately after the first rolling operation. Pipe with an outside diameter of 28 in., ½-in. wall, and a length of 50 ft has been produced by this process, although it is usually limited to 24 in. outside diameter.

Pilger Process.[1]—In the Pilger process a round cast-steel ingot or rolled-steel billet of either open or killed steel is heated and pierced on a heavy type of Mannesman roll piercer. A mandrel about 10 ft long of a diameter approximating the inside diameter of the finished pipe is forced through the pierced ingot or billet by a hydraulic ram. The mandrel encased in the ingot is placed between the rolls of the Pilger mill. These rolls have a cam-shaped contour and revolve counter to the direction in which the ingot is forced by a hydraulic ram and air-cylinder mechanism. Rotation of the rolls produces the equivalent of hammer blows which forge the ingot wall down against the mandrel and drive ingot and mandrel back against the ram. As the relieved portion

[1] *Blast Furnace and Steel Plant*, June, 1927; *Iron Trade Review*, Jan. 10, 1929.

of the rolls comes into position with further rotation of the rolls, the plunger of the air cylinder and hydraulic ram again forces the ingot into position between the rolls. The pipe is rotated 90 deg after each blow of the rolls. It may be noted that in the Pilger process a short thick-walled cylinder is swaged down into a long thin-walled pipe of approximately the same inside diameter as the cylinder, whereas in the plug-mill process a thick-walled pipe of small diameter is expanded to form a thinner walled pipe of larger diameter.

Cupping Process.—In the cupping process plates 2 to 7 ft square, $\frac{3}{8}$ to 4 in. thick, are cut into circular disks, heated to forging temperature, placed concentric with a bottom die, and pushed through with a round-nose plunger. The resulting cup is reheated to forging temperature and forced through a smaller die that leaves it practically cylindrical. The closed-end cylinder is again heated and pushed, closed end foremost, through a series of 3 to 12 dies of successively decreasing diameters which are mounted on a horizontal bench. The cylinder is reheated as many as nine times between these drawing operations. Following this the closed end is cut off, and the pipe straightened. The maximum length of pipe produced by the cupping process varies from 10 to 20 ft depending upon size and wall thickness of pipe.

Specifications.—Seamless pipe may be purchased under the following ASTM specifications or under API Specification 5L for line pipe, or AWWA Specification 7A.4 for water pipe.

A120.—Commercial steel pipe for ordinary uses, where no physical tests are required.

A53.—Commercial steel pipe for ordinary uses where physical tests are required.

A106.—Pipe for high-temperature service. Temperatures up to 850 F for power piping and up to 1000 F for oil piping.

A158.—Alloy-steel pipe for service at temperatures from 750 to 1100 F.

A206.—Carbon-molybdenum alloy-steel pipe for service at temperatures from 750 to 1000 F.

MANUFACTURE OF ELECTRIC-WELDED STEEL PIPE

Electric-welded steel pipe is made with one or more longitudinal welded seams or with spiral welded seams. Electric-resistance welding is used in one process, whereas other processes employ various sorts of electric-fusion welding, according to the thickness

of the material and the service for which the pipe is intended. Distinctive features of the various types are explained below in some detail.

Resistance Welded.—Electric-resistance-welded pipe made of open-hearth or electric-furnace steel is sold in competition with furnace-welded pipe for general purposes and is used also, to a considerable extent, for cross-country gas and oil lines. Flat plate of the thickness desired in the final pipe is sheared to the proper width, shot blasted or pickled, and formed cold into tubular shape by a series of rolls. As the formed plate is fed into a welding machine, consisting of three adjustable pressure rolls and two circular electrodes, current is passed from one electrode to the other across the abutting edges. The resistance offered to passage of the current produces a welding temperature. At the same time the pressure rolls force the edges together to complete the weld under pressure. Immediately after welding and while the weld is still hot, the pipe passes between rolls and over an inside mandrel which iron out the metal extruded by forcing the edges together in making the weld. The pipe then passes through sizing rolls which round up the tube to desired outside diameter. A cross roll straightener completes the operation (see Fig. 2).

Electric-resistance-welded pipe is available in lengths up to 50 ft and may be purchased under ASTM Specifications A53 or A135 (abstracted on pages 373 and 394, respectively), API Specification 5L (see page 113), or AWWA Specification 7A.4 (see page 406).

Electric-fusion-welded Pipe.—Particularly in the larger diameters, pipe can be made economically by rolling plate to a circular shape and electric-fusion-welding the longitudinal seam or seams. Two distinct types of such fusion-welded pipe are produced: (1) a superior grade of pipe intended for high-temperature and high-pressure service made under conditions equivalent to requirements of the ASME Boiler Construction Code, Par. U68, fusion-welded vessels involving radiographic examination and stress relieving; and (2) pipe for ordinary uses such as air-, gas-, and water-transmission lines. Longitudinal-seam fusion-welded pipe for high-temperature, high-pressure service may be ordered to ASTM Specification A155, (see page 397) while pipe for ordinary service is covered by ASTM Specifications A134 (see page 399), and A139 (see page 400), API Specification 5L (see page 1131), and AWWA Specifications 7A.3 and 7A.4 (see pages 406 to 416).

Spiral welding is adapted to making lightweight pipe for transmission lines, dredge piping, irrigation, and like purposes where the

stiffening effect of the spiral seam helps to offset any undue tendency to collapsibility arising from the thin wall. Spiral-welded pipe usually is made in 40-ft lengths from strip corresponding to U. S. Standard gauge numbers (see Table III, page 22, for gauge numbers). The use of 20-ft lengths of spiral-welded pipe in conjunction with Victaulic or similar couplings and automatic pumping units has made it possible to construct temporary portable gasoline and fuel oil lines for military purposes (see page 429 for Victaulic couplings).

Spiral-welded pipe is made by winding coils of rather narrow strip steel into cylinders with the abutting or overlapping edges of the strip forming a helix. In one variety of this product the edges of the strip come together so that they can be fused by the electric arc into a butt weld. In another variety the edges overlap so that the outside edge can be fillet-welded to the underlying strip by the electric-arc process. Advantages are claimed for their respective products by the manufacturers of both. The overlapping variety has the inner edge of the plate scarfed and the outer edge flared so that the helix fits together flush on the inside and with a projecting rim on the outside to facilitate making the fillet weld.

Spiral-welded pipe can be ordered to the following specifications:

Medium wall, large diameter: ASTM A134 (see page 399) and AWWA 7A.3 (see page 416).

Medium wall, small diameters: ASTM A139 (see page 400) and AWWA 7A.4 (see page 408).

Light wall (U. S. Standard gauge Nos. 16 to 8) 4 to 48 in. in diameter: ASTM A211 (see page 402) and AWWA 7A.3 and 7A.4 (see pages 416 and 408).

Line joints for lightweight pipe may be coupled, flanged, or welded as best suits installation requirements. Couplings usually are of the Dresser or Victaulic types described on pages 428, 429. If couplings are to be used, the pipe ends should be specially prepared. Flanges and fittings suitable for use with lightweight pipe are described on pages 585 to 587. Where pipe lighter than No. 8 gauge is to have welded line joints, reinforcing bands may be required at the pipe ends.

MANUFACTURE OF MISCELLANEOUS LARGE PIPE

Riveted pipe is made from steel or wrought-iron plate, usually with the edges overlapped to form a lap joint that may be longitudinal or spiral. It is customary to use rivets of about the same

material as the plate and to calk the overlapping edge to obtain a fluid-tight joint. This type of construction is suitable for large-diameter pipe of rather thin wall that is intended for cold-water lines and similar service at relatively low pressure. Riveted-steel pipe may be bought to ASTM Specification A138 or AWWA 7A.1, and riveted wrought-iron to ASTM A138. With the development of fusion welding this method of joining the edges of the sheets has largely superseded the use of rivets. The advantages of welded pipe over riveted pipe are greater joint strength and greater carrying capacity. Double-riveted lap joints have an efficiency of only 70 per cent; single-riveted, 45 per cent.

Forge-welded or hammer-welded pipe is made in sizes 14 to 96 in. outside diameter and wall thicknesses of ¼ to 1¼ in., inclusive, by rolling plate into cylindrical shape with the edges overlapping. Following this, the joint is welded by heating a short length of the seam at a time and hammering or pressing the edges of the plate together on an anvil. This process has been superseded largely by electric-fusion welding. Forge-welded steel pipe may be ordered to ASTM Specification A136.

Lock-bar pipe is made in large sizes for cold-water and gas lines and similar services by rolling plate into cylindrical shape after the edges have been planed and upset to make a dovetail which engages in the groove of an H-shape lock bar to form the longitudinal joint of the pipe. After assembly of the plates and lock bars, the latter are closed by cold pressing, thus making a tight fit over the dovetail edges of the plate. The line joints of lock-bar pipe are made by overlapping and riveting the ends together. Lock-bar steel pipe may be ordered to AWWA Specification 7A.2 (see page 402).

DIMENSIONS OF STEEL PIPE

Pipe up to 12 in. diameter usually is designated by its nominal inside diameter, which differs from the actual inside diameter to a greater or less degree depending upon the wall thickness and size of pipe. The outside diameters are the same for all weights of pipe, while the greater the wall thickness, the less the actual inside diameter. Pipe 14 in. and larger is designated by its outside diameter. There is a growing tendency in the trade to refer to all pipe by actual outside diameter and wall thickness, as, for example, 6⅝-in. O. D. pipe with 0.28-in. wall, or simply 6⅝ × 0.28 in. pipe.

Attention is called to the need for care in the use of *nominal pipe diameters* since AWWA specifications for pipe made by different

processes differ in this regard. Nominal diameters of cast-iron pipe and riveted steel pipe refer to inside diameters while, with pipe fabricated by fusion welding longitudinal or spiral seams, the nominal diameter refers to the outside diameter, for sizes 14 to 28 in. inclusive and to the inside diameter for sizes 30 in. and larger.

The *dimensions and weights* of "standard-weight," "extra-strong," and "double-extra-strong" steel pipe are given in Tables I, II, and III. Although these designations eventually will be superseded by pipe-thickness "schedules," as in the Tentative American Standard for Wrought-iron and Wrought-steel Pipe, ASA B36.10, general familiarity with the former terminology makes it desirable to retain these tables for the time being. Dimensions and test pressures of large-outside-diameter lap-welded and seamless pipe are given in Table IV. The dimensions of standard-weight steel pipe in sizes 10 in. and smaller are identical with those for Schedule 40 given in Table V which is reproduced from ASA B36.10. The dimensions of extra-strong pipe 8 in. and smaller are identical with Schedule 80 pipe of Table V. The dimensions of double-extra-strong pipe given in Table III are identical with the dimensions given for double-extra-strong steel pipe in the appendix to ASA B36.10.

Adherence to the standard schedule thicknesses and weights given in Tables V and VI will reduce unnecessary duplication in the manufacture of pipe and facilitate purchase in small lots. The schedule numbers used in these tables and in Tables IX to XII inclusive of inside-diameter functions approximate the values of the expression $1{,}000 \times P/S$. Thus Schedule 40 is roughly suitable for a service pressure of 400 psi, provided an allowable stress of 10,000 psi, is satisfactory for the particular pipe material under the type of service and temperature conditions involved. However, final selection of pipe-wall thickness should be computed by using a recognized design formula, such as that of the ASA Code for Pressure Piping in equation (16*e*) on page 42 of Chap. II, which sets definite allowable stress values for certain materials under particular service conditions and provides allowances for corrosion or method of end attachment Attention is called to the fact that in accordance with trade practice all tabular data for wall thickness and weight of wrought pipe are based on nominal dimensions rather than on minimum thickness allowed on inspection as used in equation (16*e*). Departures from nominal dimensions are set by the wall-thickness tolerances allowed

in the specification under which the pipe is bought. Hence the established difference between nominal and minimum thickness should be taken into account in ordering pipe to meet code requirements.

The *inside-diameter functions* for different kinds of pipe given in Tables VII to XII inclusive are intended for use in the fluid flow formulas given in other chapters.

SPECIFICATIONS FOR STEEL PIPE

Varieties of steel pipe may be ordered in accordance with the corresponding specifications of the ASTM, which are quoted or abstracted on the following pages.[1] Several of these specifications have been adopted as American Standard and bear an additional serial designation of the ASA. Furnace-welded, electric-welded, and seamless pipe may be purchased also in accordance with API Specification 5L for line pipe (abstracted on page 1131); AWWA Specification 7A.3 and 7A.4 (abstracted on pages 406 to 416); and specifications of various departments and bureaus of the Federal government.

U. S. COMMERCIAL STANDARDS

Among the numerous commercial standards promulgated by the U.S. Department of Commerce through the National Bureau of Standards the following are of particular interest to piping designers:

Simplified Practice Recommendations:[2]

No. 57. Wrought-iron and wrought-steel pipe, valves, and fittings (see simplified list of pipe sizes, pages 357 to 359).

No. 183. Bronze or brass valves—pressure ratings.

No. 184. Iron body valves—pressure ratings.

No. 185. Pipe fittings (gray cast iron, malleable iron, and brass or bronze).

R 201. Iron and steel pop safety valves.

Commercial Standards:[2]

CS5. Pipe nipples; brass, copper, steel, and wrought-iron.

CS7. Standard weight malleable iron or steel screwed unions.

[1] Copies of ASTM specifications may be obtained from the American Society for Testing Materials, 260 South Broad St., Philadelphia, Pa. Specifications bearing the ASA designation also may be obtained from the American Standards Association, 29 West 39th St., New York. Specification for Line Pipe, API 5L, may be obtained from the American Petroleum Institute, Dallas, Tex.

[2] These simplified practice recommendations and commercial standards are for sale by the Superintendent of Documents, Washington, D.C.

TABLE I.—DIMENSIONS OF STANDARD-WEIGHT STEEL PIPE[1]

Nominal size	Diameters, in. External	Diameters, in. Internal	Thickness, in.	Weight per ft, lb Plain ends	Weight per ft, lb Threads and couplings	Threads per inch	Couplings Diameter, in.	Couplings Length, in.	Couplings Weight, lb	Circumference, in. External	Circumference, in. Internal	Transverse area, sq in. External	Transverse area, sq in. Internal	Transverse area, sq in. Metal	Length of pipe, ft per sq ft External surface	Length of pipe, ft per sq ft Internal surface	Length of pipe in ft, containing 1 cu ft
1/8	0.405	0.269	0.068	0.24	0.24	27	0.563	1 3/16	0.03	1.272	0.845	0.129	0.057	0.072	9.431	14.200	2,533.8
1/4	0.540	0.364	0.088	0.42	0.42	18	0.719	1 3/16	0.07	1.696	1.144	0.229	0.104	0.125	7.073	10.494	1,383.8
3/8	0.675	0.493	0.091	0.57	0.57	18	0.875	1 3/16	0.09	2.121	1.549	0.358	0.191	0.167	5.658	7.748	754.36
1/2	0.840	0.622	0.109	0.85	0.85	14	1.063	1 9/16	0.17	2.639	1.954	0.554	0.304	0.250	4.547	6.141	473.91
3/4	1.050	0.824	0.113	1.13	1.13	14	1.313	1 5/8	0.24	3.299	2.589	0.866	0.533	0.333	3.637	4.635	270.03
1	1.315	1.049	0.133	1.68	1.68	11 1/2	1.576	2	0.38	4.131	3.296	1.358	0.864	0.494	2.904	3.641	166.62
1 1/4	1.660	1.380	0.140	2.27	2.28	11 1/2	1.900	2 1/16	0.44	5.215	4.335	2.164	1.495	0.669	2.301	2.768	96.275
1 1/2	1.900	1.610	0.145	2.72	2.73	11 1/2	2.200	2 1/16	0.60	5.969	5.058	2.835	2.036	0.799	2.010	2.372	70.733
2	2.375	2.067	0.154	3.65	3.67	11 1/2	2.750	2 1/8	0.93	7.461	6.494	4.430	3.355	1.075	1.608	1.848	42.913
2 1/2	2.875	2.469	0.203	5.79	5.81	8	3.250	3 1/8	1.82	9.032	7.757	6.492	4.788	1.704	1.328	1.547	30.077
3	3.500	3.068	0.216	7.58	7.61	8	4.000	3 1/4	2.98	10.996	9.638	9.621	7.393	2.228	1.091	1.245	19.479
3 1/2	4.000	3.548	0.226	9.11	9.20	8	4.625	3 3/8	4.20	12.566	11.146	12.566	9.886	2.680	0.954	1.076	14.565
4	4.500	4.026	0.237	10.79	10.88	8	5.000	3 1/2	4.51	14.137	12.648	15.904	12.730	3.174	0.848	0.948	11.312
5	5.563	5.047	0.258	14.62	14.81	8	6.296	3 3/4	8.25	17.477	15.854	23.306	20.002	4.300	0.686	0.757	7.199
6	6.625	6.065	0.280	18.97	19.18	8	7.390	4	10.88	20.813	19.054	34.472	28.89	5.581	0.576	0.629	4.984
8[2]	8.625	8.071	0.277	24.70	25.00	8	9.625	5 1/4	23.46	27.096	25.356	58.426	51.162	7.265	0.443	0.473	2.815
8	8.625	7.981	0.322	28.55	28.80	8	9.625	5 1/4	23.46	27.096	25.073	58.426	50.027	8.399	0.443	0.478	2.878
10[2]	10.750	10.192	0.279	31.20	32.00	8	11.750	5 3/4	32.02	33.772	32.019	90.763	81.585	9.178	0.355	0.375	1.765
10[2]	10.750	10.136	0.307	34.24	35.00	8	11.750	5 3/4	32.02	33.772	31.843	90.763	80.691	10.072	0.355	0.377	1.785
10	10.750	10.020	0.365	40.48	41.13	8	11.750	5 3/4	32.02	33.772	31.479	90.763	78.854	11.908	0.355	0.381	1.826
12[2]	12.750	12.090	0.330	43.77	45.00	8	14.000	6 1/8	49.92	40.055	37.982	127.676	114.80	12.876	0.299	0.316	1.254
12	12.750	12.000	0.375	49.56	50.70	8	14.000	6 1/8	49.92	40.055	37.699	127.676	113.10	14.579	0.299	0.318	1.273

[1] National Tube Co.
[2] These sizes are not in the list included in the simplified practice recommendations of the U.S. Department of Commerce.

TABLE II.—DIMENSIONS OF EXTRA-STRONG STEEL PIPE[1]

Size	Diameter		Thickness	Weight per foot, plain ends	Circumference		Transverse area			Length of pipe per square foot		Length of pipe containing 1 cu ft
	External	Internal			External	Internal	External	Internal	Metal	External surface	Internal surface	
	Inches	Inches	Inches	Pounds	Inches	Inches	Square inches	Square inches	Square inches	Feet	Feet	Feet
1/8	0.405	0.215	0.095	0.31	1.272	0.675	0.129	0.036	0.093	9.431	17.766	3,966.4
1/4	0.540	0.302	0.119	0.54	1.696	0.949	0.229	0.072	0.157	7.073	12.648	2,010.3
3/8	0.675	0.423	0.126	0.74	2.121	1.329	0.358	0.141	0.217	5.658	9.030	1,024.7
1/2	0.840	0.546	0.147	1.09	2.639	1.715	0.554	0.234	0.320	4.547	6.996	615.02
3/4	1.050	0.742	0.154	1.47	3.299	2.331	0.866	0.433	0.433	3.637	5.148	333.02
1	1.315	0.957	0.179	2.17	4.131	3.007	1.358	0.719	0.639	2.904	3.991	200.19
1 1/4	1.660	1.278	0.191	3.00	5.215	4.015	2.164	1.283	0.881	2.301	2.988	112.26
1 1/2	1.900	1.500	0.200	3.63	5.969	4.712	2.835	1.767	1.068	2.010	2.546	81.487
2	2.375	1.939	0.218	5.02	7.461	6.092	4.430	2.953	1.477	1.608	1.969	48.766
2 1/2	2.875	2.323	0.276	7.66	9.032	7.298	6.492	4.238	2.254	1.328	1.644	33.976
3	3.500	2.900	0.300	10.25	10.996	9.111	9.621	6.605	3.016	1.091	1.317	21.801
3 1/2	4.000	3.364	0.318	12.51	12.566	10.568	12.566	8.888	3.678	0.954	1.135	16.202
4	4.500	3.826	0.337	14.98	14.137	12.020	15.904	11.497	4.407	0.848	0.998	12.525
5	5.563	4.813	0.375	20.78	17.477	15.120	24.306	18.190	6.112	0.686	0.794	7.916
6	6.625	5.761	0.432	28.57	20.813	18.099	34.472	26.067	8.405	0.576	0.663	5.524
8	8.625	7.625	0.500	43.39	27.096	23.955	58.426	45.664	12.763	0.443	0.500	3.154
10	10.750	9.750	0.500	54.74	33.772	30.631	90.763	74.662	16.101	0.355	0.391	1.929
12	12.750	11.750	0.500	65.42	40.055	36.914	127.676	108.43	19.242	0.299	0.325	1.328

[1] National Tube Co.

TABLE III.—DIMENSIONS OF DOUBLE-EXTRA-STRONG STEEL PIPE[1]

Nominal size	Diameter		Thickness	Weight per foot, plain ends	Circumference		Transverse area			Length of pipe per square foot		Length of pipe containing 1 cu ft
	External	Internal			External	Internal	External	Internal	Metal	External surface	Internal surface	
	Inches	Inches	Inches	Pounds	Inches	Inches	Square inches	Square inches	Square inches	Feet	Feet	Feet
½	0.840	0.252	0.294	1.71	2.639	0.792	0.554	0.050	0.504	4.547	15.158	2,887.2
¾	1.050	0.434	0.308	2.44	3.299	1.363	0.866	0.148	0.718	3.637	8.801	973.40
1	1.315	0.599	0.358	3.60	4.131	1.882	1.358	0.282	1.076	2.904	6.377	511.00
1¼	1.660	0.896	0.382	5.21	5.215	2.815	2.164	0.630	1.534	2.301	4.263	228.38
1½	1.900	1.100	0.400	6.41	5.969	3.456	2.835	0.950	1.885	2.010	3.472	151.53
2	2.375	1.503	0.436	9.03	7.461	4.722	4.430	1.774	2.656	1.608	2.541	81.162
2½	2.875	1.771	0.552	13.70	9.032	5.564	6.492	2.464	4.028	1.328	2.157	58.457
3	3.500	2.300	0.600	18.58	10.996	7.226	9.261	4.155	5.466	1.091	1.660	34.659
4	4.500	3.152	0.674	27.54	14.137	9.902	15.904	7.803	8.101	0.848	1.211	18.454
5	5.563	4.063	0.750	38.55	17.477	12.763	24.306	12.962	11.340	0.686	0.940	11.109
6	6.625	4.897	0.864	53.16	20.813	15.384	34.472	18.834	15.637	0.570	0.780	7.646
8	8.625	6.875	0.875	72.42	27.096	21.598	58.426	37.122	21.304	0.443	0.555	3.879

[1] National Tube Co.

TABLE IV.—DIMENSIONS, WEIGHTS, AND TEST PRESSURES OF LARGE O.D. LAP-WELDED AND SEAMLESS STEEL PIPE[1]

Size O.D., in.	Weight per ft, plain ends, lb	Wall thickness: Fraction, in.	Wall thickness: Decimal, in.	Test pressure, psi: Lap-welded or Grade A seamless	Test pressure, psi: Grade B, seamless
14	36.71	¼	0.250	550	600
14	40.45	....	0.276	600	650
14	45.68	5/16	0.3125	650	750
14	54.57	⅜	0.375	800	900
14	63.37	7/16	0.4375	950	1,100
14	67.74	15/32	0.46875	1,000	1,100
14	72.09	½	0.500	1,100	1,200
16	42.05	¼	0.250	450	550
16	47.22	9/32	0.28125	550	600
16	50.63	....	0.302	550	650
16	52.36	5/16	0.3125	600	650
16	57.48	11/32	0.34375	650	750
16	62.58	⅜	0.375	700	800
16	66.81	....	0.401	750	850
16	72.72	7/16	0.4375	800	950
16	82.77	½	0.500	950	1,100
16	92.74	9/16	0.5625	1,100	1,200
18	47.39	¼	0.250	400	450
18	53.22	9/32	0.28125	450	550
18	59.03	5/16	0.3125	500	600
18	64.82	11/32	0.34375	550	650
18	70.59	⅜	0.375	650	700
18	76.84	....	0.409	700	750
18	82.06	7/16	0.4375	750	850
18	87.77	15/32	0.46875	800	900
18	93.45	½	0.500	850	950
18	104.76	9/16	0.5625	950	1,100
20	59.23	9/32	0.28125	400	500
20	65.71	5/16	0.3125	450	550
20	72.16	11/32	0.34375	500	600
20	78.60	⅜	0.375	550	650
20	85.58	....	0.409	600	700
20	91.41	7/16	0.4375	650	750
20	97.78	15/32	0.46875	700	800
20	104.13	½	0.500	750	850
20	116.77	9/16	0.5625	850	950
22	72.38	5/16	0.3125	450	500
22	79.51	11/32	0.34375	450	550
22	86.61	⅜	0.375	500	600
22	100.75	7/16	0.4375	600	700
22	107.79	15/32	0.46875	650	700
22	114.81	½	0.500	700	750
22	128.79	9/16	0.5625	750	850
24	79.06	5/16	0.3125	400	450
24	86.85	11/32	0.34375	450	500
24	94.62	⅜	0.375	450	550
24	110.10	7/16	0.4375	550	600
24	117.81	15/32	0.46875	600	650
24	125.49	½	0.500	650	700
24	140.80	9/16	0.5625	700	800

[1] The sizes and weights given in this table are listed as those which can be furnished most readily. Other sizes and weights may be obtained by special agreement. See National Tube Co., General Catalogue, 1941, pp. 103–104.

Pipe listed in this table may be purchased to conform to ASTM Specifications A120 or A53 or to API Specification 5L.

TABLE V.—DIMENSIONS OF WELDED AND SEAMLESS STEEL PIPE
(Conforming to ASA B36.10)
(All dimensions in inches)

Nominal pipe size	Outside diameter	Nominal wall thicknesses or schedule numbers									
		Sched. 10	Sched. 20	Sched. 30	Sched. 40	Sched. 60	Sched. 80	Sched. 100	Sched. 120	Sched. 140	Sched. 160
1/8	0.405	....	....	....	**0.068**	....	**0.095**				
1/4	0.540	....	....	....	**0.088**	....	**0.119**				
3/8	0.675	....	....	....	**0.091**	....	**0.126**				
1/2	0.840	....	....	....	**0.019**	....	**0.147**	....	....	....	0.187
3/4	1.050	....	....	....	**0.113**	....	**0.154**	....	....	....	0.218
1	1.315	....	....	....	**0.133**	....	**0.179**	....	....	....	0.250
1 1/4	1.660	....	....	....	**0.140**	....	**0.191**	....	....	....	0.250
1 1/2	1.900	....	....	....	**0.145**	....	**0.200**	....	....	....	0.281
2	2.375	....	....	....	**0.154**	....	**0.218**	....	....	....	0.343
2 1/2	2.875	....	....	....	**0.203**	....	**0.276**	....	....	....	0.375
3	3.5	....	....	....	**0.216**	....	**0.300**	....	....	....	0.437
3 1/2	4.0	....	....	....	**0.226**	....	**0.318**				
4	4.5	....	....	....	**0.237**	....	**0.337**	....	0.437	....	0.531
5	5.563	....	....	....	**0.258**	....	**0.375**	....	0.500	....	0.625
6	6.625	....	....	....	**0.280**	....	**0.432**	....	0.562	....	0.718
8	8.625	....	0.250	**0.277**	**0.322**	0.406	0.500	0.593	0.718	0.812	0.906
10	10.75	....	0.250	**0.307**	**0.365**	**0.500**	0.593	0.718	0.843	1.000	1.125
12	12.75	....	....	....	**0.375**[1]	**0.500**[1]					
12	12.75	....	0.250	**0.330**	0.406	0.562	0.687	0.843	1.000	1.125	1.312
14 O.D.	14.0	0.250	0.312	0.375	0.437	0.593	0.750	0.937	1.062	1.250	1.406
16 O.D.	16.0	0.250	0.312	0.375	0.500	0.656	0.843	1.031	1.218	1.437	1.562
18 O.D.	18.0	0.250	0.312	0.437	0.562	0.718	0.937	1.156	1.343	1.562	1.750
20 O.D.	20.0	0.250	0.375	0.500	0.593	0.812	1.031	1.250	1.500	1.750	1.937
24 O.D.	24.0	0.250	0.375	0.562	0.687	0.937	1.218	1.500	1.750	2.062	2.312
30 O.D.	30.0	0.312	0.500	0.625							

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions and include an allowance for mill tolerance of 12.5 per cent under the nominal thicknesses.

Thicknesses shown in boldface type in Schedules 30 and 40 are identical with thicknesses for standard-weight pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for extra-strong pipe in former lists.

The schedule numbers indicate approximate values of the expression $1{,}000 \times P/S$.

[1] Old "standard-weight" 0.375-in. wall and "extra-strong" 0.500-in. wall pipe may be substituted for Schedule 40 and Schedule 60 pipe, respectively, where agreeable to the purchaser and suitable for the service conditions.

TABLE VI.—NOMINAL WEIGHTS OF WELDED AND SEAMLESS STEEL PIPE,[1] ASA B36.10

Nominal pipe size, inches	Sched. 10, plain ends	Sched. 20, plain ends	Sched. 30		Sched. 40		Sched. 60, plain ends	Sched. 80, plain ends	Sched. 100, plain ends	Sched. 120, plain ends	Sched. 140, plain ends	Sched. 160, plain ends
			Plain ends	Threads[2] and couplings	Plain ends	Threads[2] and couplings						
1/8	....	.....	.....	.....	**0.25**	**0.25**	.....	**0.32**				
1/4	....	.....	.....	.....	**0.43**	**0.43**	.....	**0.54**				
3/8	....	.....	.....	.....	**0.57**	**0.57**	.....	**0.74**				
1/2	....	.....	.....	.....	**0.86**	**0.86**	.....	**1.09**	.....	.....	.....	1.31
3/4	....	.....	.....	.....	**1.14**	**1.14**	.....	**1.48**	.....	.....	.....	1.94
1	....	.....	.....	.....	**1.68**	**1.69**	.....	**2.18**	.....	.....	.....	2.85
1 1/4	....	.....	.....	.....	**2.28**	**2.29**	.....	**3.00**	.....	.....	.....	3.77
1 1/2	....	.....	.....	.....	**2.72**	**2.74**	.....	**3.64**	.....	.....	.....	4.86
2	....	.....	.....	.....	**3.66**	**3.68**	.....	**5.03**	.....	.....	.....	7.45
2 1/2	....	.....	.....	.....	**5.80**	**5.82**	.....	**7.67**	.....	.....	.....	10.0
3	....	.....	.....	.....	**7.58**	**7.62**	.....	**10.3**	.....	.....	.....	14.3
3 1/2	....	.....	.....	.....	**9.11**	**9.21**	.....	**12.5**				
4	....	.....	.....	.....	**10.8**	**10.9**	.....	**15.0**	.....	19.0	.....	22.6
5	....	.....	.....	.....	**14.7**	**14.9**	.....	**20.8**	.....	27.1	.....	33.0
6	....	.....	.....	.....	**19.0**	**19.2**	.....	**28.6**	.....	36.4	.....	45.3
8	....	22.4	**24.7**	**25.0**	**28.6**	**28.8**	35.7	**43.4**	50.9	60.7	67.8	74.7
10	....	28.1	**34.3**	**35.0**	**40.5**	**41.2**	**54.8**	64.4	77.0	89.2	105	116
12	....	.....	.....	.....	49.5	50.7[3]	**65.4**[3]					
12	....	33.4	**43.8**	**45.0**	53.6	55.0	73.2	88.6	108	126	140	161
14 O.D.	36.8	45.7	54.6	....	63.3	.....	85.0	107	131	147	171	190
16 O.D.	42.1	52.3	62.6	....	82.8	.....	108	137	165	193	224	241
18 O.D.	47.4	59.0	82.0	....	105	.....	133	171	208	239	275	304
20 O.D.	52.8	78.6	105	....	123	.....	167	209	251	297	342	374
24 O.D.	63.5	94.7	141	....	171	.....	231	297	361	416	484	536
30 O.D.	99.0	158	197									

[1] Weights are given in pounds per linear foot and are for pipe with plain ends except for sizes which are commercially available with threads and couplings for which both weights are listed.

[2] The weights for line pipe with couplings are slightly greater than shown in Schedules 30 and 40 and may be found in API Specification 5-L.

[3] Old "standard-weight" 0.375-in. wall and "extra-strong" 0.500-in. wall pipe. Weights shown in boldface type in Schedules 30 and 40 are identical with weights for standard-weight pipe in former lists; those in Schedules 60 and 80 are identical with weights for extra-strong pipe in former lists.

The schedule numbers indicate approximate values of the expression $1,000 \times P/S$.

TABLE VII.—INSIDE-DIAMETER FUNCTIONS OF STEEL PIPE[1] FOR USE IN GAS AND AIR FLOW FORMULAS

Nominal pipe size, in.	Outside diameter, D	Schedule number or weight	Wall thickness, in. t	Inside diameter, in. d	$\sqrt{d^5} = d^{5/2} = d^{2.5}$	$\sqrt{d^{5\frac{1}{3}}} = \sqrt{d^{16/3}} = d^{8/3} = d^{2.667}$	d^5	$d^{16/3} = d^{5\frac{1}{3}} = d^{5.333}$	$K = \sqrt{\dfrac{d^5}{1 + \dfrac{3.6}{d} + 0.03d}}$	Internal cross-sectional area, sq ft
⅜	0.675	40	0.091	0.493	0.1707	0.1517	0.02912	0.02301	0.059	0.00133
½	0.840	40	0.109	0.622	0.3051	0.2820	0.09310	0.06312	0.117	0.00211
¾	1.050	40	0.113	0.824	0.6163	0.5968	0.3799	0.3561	0.265	0.00371
1	1.315	40	0.133	1.049	1.127	1.136	1.270	1.291	0.532	0.00600
1¼	1.660	40	0.140	1.380	2.237	2.361	5.005	5.572	1.171	0.01040
1½	1.900	40	0.145	1.610	3.289	3.561	10.82	12.68	1.816	0.01414
2	2.375	40	0.154	2.067	6.143	6.933	37.73	48.06	3.675	0.02330
2½	2.875	40	0.203	2.469	9.579	11.14	91.75	124.0	6.015	0.03322
3	3.500	40	0.216	3.068	16.49	19.87	271.8	395.0	10.94	0.05130
3½	4.000	40	0.226	3.548	23.71	29.28	562.2	857.5	16 23	0.06870
4	4.500	40	0.237	4.026	32.52	41.02	1,058	1,683	22.95	0.08840
5	5.563	40	0.258	5.047	57.22	74.95	3,275	5,627	41.75	0.1390
6	6.625	40	0.280	6.065	90.59	122.3	8,206	14,970	68.00	0.2006
6	6.625	Special	0.250	6.125	92.85	125.6	8,620	15,773	69.80	0.2046
6	6.625	Special	0.188	6.250	97.66	132.5	9,537	17,567	73.60	0.2131
7	7.625	Special	0.301	7.023	130.7	180.9	17,085	32,720	99.50	0.2690
8	8.625	40	0.322	7.981	179.9	254.4	32,380	64,710	138.5	0.3474
8	8.625	30	0.277	8.071	185.1	262.1	34,248	68,700	142.5	0.3553
8	8.625	20	0.250	8.125	188.2	266.8	35,409	71,182	145.0	0.3601
8	8.625	Special	0.220	8.185	191.7	272.1	36,740	74,035	147.7	0.3654
9	9.625	Special	0.342	8.941	239.0	336.5	57,140	118,600	185.0	0.4360
10	10.75	40	0.365	10.020	317.8	466.6	101,000	217,750	246.8	0.5475
10	10.75	30	0.307	10.136	327.1	481.2	106,987	231,540	254.0	0.5603
10	10.75	Special	0.279	10.192	331.6	488.3	109,977	238,440	257.3	0.5666
10	10.75	20	0.250	10.250	336.4	495.7	113,141	245,760	261.5	0.5731
11	11.75	Special	0.375	11.000	401.3	598.5	161,052	358,176	312.0	0.6500
12	12.75	Std. Wt.	0.375	12.000	498.8	754.8	248,831	569,680	387.5	0.7854
12	12.75	30	0.330	12.090	508.2	770.0	258,304	592,870	395.0	0.7972
12	12.75	Special	0.281	12.188	518.5	786.7	268,946	618,920	403.0	0.8102
12	12.75	20	0.250	12.250	525.2	797.5	275,855	635,930	408.0	0.8185
14	14.00	10	0.250	13.500	669.6	1,033	448,403	1,067,695	517.0	0.9940
15	15.00	Special	0.250	14.500	800.6	1,250	640,975	1,563,000	618.0	1.147
16	16.00	30	0.375	15.250	908.2	1,430	824,801	2,045,400	697.0	1.268
16	16.00	20	0.312	15.375	926.9	1,462	859,442	2,136,500	712.0	1.290
16	16.00	10	0.250	15.500	945.9	1,494	894,660	2,230,600	727.0	1.310
18	18.00	Special	0.375	17.250	1,236	1,987	1,526,000	3,946,500	940.0	1.623
18	18.00	20	0.312	17.375	1,258	2,025	1,583,978	4,101,600	956.0	1.647
18	18.00	10	0.250	17.500	1,281	2,064	1,641,307	4,261,400	975.0	1.670
20	20.00	20	0.375	19.250	1,626	2,662	2,643,344	7,084,200	1,225	2.014
20	20.00	Special	0.312	19.375	1,652	2,708	2,730,295	7,333,000	1,245	2.047

(*Continued on page* 364)

TABLE VII.—(*Concluded*)

Nominal pipe size, in.	Outside diameter, D	Schedule number or weight	Wall thickness, in. t	Inside diameter, in. d	$\sqrt{d^5} = d^{5/2} = d^{2.5}$	$\frac{\sqrt{d^{5\frac{1}{3}}}}{} = \sqrt{d^{16/3}} = d^{8/3} = d^{2.667}$	d^5	$d^{16/3} = d^{5\frac{1}{3}} = d^{5.333}$	$K = \sqrt{\frac{d^5}{1 + \frac{3.6}{d} + 0.03d}}$	Internal cross-sectional area, sq ft
20	20.00	..	.250	19.500	1,679	2,755	2,819,520	,589,000	1,262	2.074
22	22.00	Special	.437	21.125	2,051	3,410	4,207,117	,631,000	1,528	2.434
22	22.00	Speical	.312	21.375	2,112	3,519	4,462,515	,384,000	1,570	2.492
22	22.00	Special	.250	21	2,14[illegible]	3,574	4,594,000	,775,000	1,593	2.521
24	24.00	[illegible]	.437	23.125	2,57[illegible]	4,34[illegible]	6,613,167	,843,000	1,895	2.917
24	24.00		.312	23.375	2,64[illegible]	4,467	6,978,46[illegible]	,953,000	1,943	2.980
24	24.00	10	.250	23.500	2,677	4,531	7,167,03[illegible]	,529,00[illegible]	1,965	3.012
26	26.00	Special	.437	25.125	3,164	5,415	10,012,22[illegible]	,326,00[illegible]	2,300	3.443
26	26.0[illegible]	Special	.250	25.500	3,284	5,63[illegible]	10,782,00[illegible]	,736,00[illegible]	2,380	3.547
28	28.0[illegible]	Special	.437	27.125	3,83[illegible]	6,64[illegible]	14,684,07[illegible]	,120,00[illegible]	2,750	4.013
28	28.0[illegible]	Special	.250	27.500	3,96[illegible]	6,89[illegible]	15,727,68[illegible]	,471,00[illegible]	2,840	4.125
30	30.0[illegible]	Special	.437	29.125	4,578	8,03[illegible]	20,957,05[illegible]	,482,00[illegible]	3,240	4.627
	30.0[illegible]	[illegible]	.312	29.375	4,67[illegible]	8,21	21,872,070	,486,10[illegible]	3,310	4.706

[1] Can be used for copper tubing or cast-iron pipe of approximately the same inside diameter. For inside-diameter functions of Type K copper water tubes, see Table VIII.

TABLE VIII.—INSIDE-DIAMETER FUNCTIONS OF TYPE K[1] COPPER WATER TUBES, CONFORMING TO ASTM SPECIFICATION B88[2]

Nominal pipe size	Actual outside diameter D	Nominal wall thickness t	Inside diameter d	d^2	$\sqrt{d^5} = d^{5/2} = d^{2.5}$	$\sqrt{d^{5\frac{1}{3}}} = \sqrt{d^{16/3}} = d^{8/3} = d^{2.667}$	d^3	d^4	d^5	$d^{16/3} = d^{5\frac{1}{3}} = d^{5.333}$	$K = \sqrt{\frac{d^5}{1 + \frac{3.6}{d} + 0.03d}}$	Internal cross-sectional area, Sq in.	Internal cross-sectional area, Sq ft[3]
	0.500	0.049	0.402	0.162	0.103	0.085	0.065	0.026	0.01[illegible]	0.008	0.033	0.1269	0.0009
	0.625	0.049	0.527	0.278	0.203	0.151	0.146	0.077	0.04	0.033	0.072	0.2181	0.0015
	0.875	0.065	0.745	0.555	0.481	0.456	0.41	0.308	0.232	0.208	0.199	0.4359	0.0030
1	1.125	0.065	0.995	0.990	0.98[illegible]	0.9[illegible]	0.985	0.980	0.97[illegible]	0.97[illegible]	0.457	0.7776	0.0065
1¼	1.375	0.065	1.245	1.550	1.730	1.79[illegible]	1.930	2.400	2.990	3.215	0.874	1.2174	0.0085
1½	1.625	0.072	[illegible].481	2.193	2.670	2.855	3.248	4.800	7.130	8.130	1.433	1.7226	0.0120
2	2.125	0.083	1.959	3.842	5.370	6.000	7.530	14.78	28.85	36.0[illegible]	3.155	3.0141	0.0209
2½	2.625	0.095	2.435	5.930	9.260	10.75	14.4[illegible]	35.17	85.70	115.[illegible]	5.800	4.6568	0.0324
3	3.125	0.109	2.907	8.451	14.40	17.20	24.55	71.50	207.[illegible]	295.[illegible]	9.450	6.6462	0.0461
3½	3.625	0.120	3.385	11.45	21.04	25.70	38.80	132.	443.[illegible]	663.[illegible]	14.30	8.9993	0.0625
4	4.125	0.134	3.857	14.85	29.20	36.50	57.30	222.	850.	1,33[illegible]	20.40	11.683	0.0811
5	5.125	0.160	4.805	23.10	50.60	66.00	112.0	535.0	2,570	4,360	36.80	18.133	0.1260
6	6.125	0.192	5.741	32.90	79.00	105.5	189.0	1,085	6,240	11,130	59.00	25.886	0.1799

[1] The inside diameters and carrying capacities of Types L and M copper water tubes are intermediate between Type K and Schedule 40 (standard-weight) steel pipe. Hence their inside diameter functions can be interpolated between Tables VIII and X.

[2] All dimensions are in inches except where otherwise noted. For tolerances, and for dimensions of Types L and M copper water tubes, see Table XXXV, p. 475.

[3] This column also represents cubic feet per foot of length.

TABLE IX.—INSIDE-DIAMETER FUNCTIONS OF SCHEDULES 10, 20, AND 30 STEEL PIPE[1] CONFORMING TO ASA B36.10

Nominal pipe size	Out-side diam-eter D	Wall thick-	diam-eter	Inside diam-	Inside diam-eter cubed	Inside diam-eter, fourth power d^4	Inside diameter, fifth power d^5	Internal cross-sectional area, Square inches	Internal cross-sectional area, Square feet[2]
Schedule 10:									
14 O.D.	14.0	0.250	13.50	182.25	2,460	33,215	448,403	143.14	0.994
16 O.D.	16.0	0.250	15.50	240.25	3,724	57,720	894,660	188.69	1.310
18 O.D.	18.0	0.250	17.50	306.25	5,359	93,789	1,641,307	240.53	1.670
20 O.D.	20.0	0.250	19.50	380.25	7,415	144,590	2,819,505	298.65	2.074
24 O.D.	24.0	0.250	23.50	552.25	12,978	304,980	7,167,030	433.74	3.01[illegible]
30 O.D.	30.0	0.312	29.376	862.95	25,350	744,680	21,875,768	677.76	4.70[illegible]
Schedule 20:									
8	8.625	0.250	8.125	66.02	536.4	4,358	35,409	51.85	0.3601
10	10.75	0.250	10.25	105.06	1,077	11,038	113,141	82.52	0.5731
12	12.75	0.250	12.25	150.06	1,838	22,519	275,855	117.86	0.8185
14 O.D.	14.0	0.312	13.376	178.92	2,393	32,011	428,185	140.52	0.9758
16 O.D.	16.0	0.312	15.376	236.42	3,635	55,895	859,442	185.69	1.290
18 O.D.	18.0	0.312	17.376	301.92	5,246	91,159	1,583,978	237.13	1.647
20 O.D.	20.0	0.375	19.25	370.56	7,133	137,317	2,643,344	290.04	2.014
24 O.D.	24.0	0.375	23.25	540.56	12,568	292,208	6,793,832	424.56	2.948
30 O.D.	30.0	0.500	29.00	841.00	24,389	707,281	20,511,149	660.52	4.587
Schedule 30:									
8	8.625	**0.277**	8.071	65.14	525.7	4,243	34,248	51.16	0.3553
10	10.75	**0.307**	10.136	102.74	1,041	10,555	106,987	80.69	0.5603
12	12.75	**0.330**	12.09	146.17	1,767	21,365	258,304	114.80	0.7972
14 O.D.	14.0	0.375	13.25	175.56	2,326	30,822	408,394	137.88	0.9575
16 O.D.	16.0	0.375	15.25	232.56	3,546	54,085	824,801	182.65	1.268
18 O.D.	18.0	0.437	17.126	293.30	5,023	86,025	1,473,261	230.36	1.600
20 O.D.	20.0	0.500	19.0	361.00	6,859	30,321	2,476,099	283.53	1.969
24 O.D.	24.0	0.562	22.876	523.31	11,971	273,855	6,264,703	411.00	2.854
30 O.D.	30.0	0.625	28.75	826.56	23,764	683,206	9,642,160	649.18	4.508

[1] All dimensions in inches except where otherwise noted. Wall thicknesses shown in boldface type in Schedule 30 are identical with thicknesses for standard-weight pipe in former tables. For other data on steel pipe to Schedules 10, 20, and 30 dimensions, see Tables V and VI, pp. 361 and 362; for outside-diameter functions and moments of inertia, Table X, p. 858; for external surface, p. 697; for weight of water per foot of pipe, Table [illegible], p. 738. The inside-diameter functions of wrought-iron pipe differ somewhat from the computed values for steel pipe given in the above table but the differences are well within the manufacturing tolerances permitted with both kinds of pipe.

[2] This column also represents the contents in cubic feet per foot of length.

TABLE X.—INSIDE-DIAMETER FUNCTIONS OF SCHEDULES 40 AND 60 STEEL PIPE[1] CONFORMING TO ASA B36.10

Nominal pipe size	Outside diameter D	Wall thickness t	Inside diameter d	Inside diameter squared d^2	Inside diameter cubed d^3	Inside diameter fourth power d^4	Inside diameter fifth power d^5	Internal cross-sectional area: Square inches	Internal cross-sectional area: Square feet[2]
Schedule 40:									
1/8	0.405	**0.068**	0.269	0.0724	0.0195	0.00524	0.00141	0.0569	0.00040
1/4	0.540	**0.088**	0.364	0.1325	0.0482	0.01755	0.00639	0.1041	0.00072
3/8	0.675	**0.091**	0.493	0.2430	0.1198	0.05907	0.02912	0.1909	0.00133
1/2	0.840	**0.109**	0.622	0.3869	0.2406	0.1497	0.09310	0.3039	0.00211
3/4	1.050	**0.113**	0.824	0.679	0.5595	0.4610	0.3799	0.5333	0.00371
1	1.315	**0.133**	1.049	1.100	1.154	1.211	1.270	0.8639	0.00600
1 1/4	1.660	**0.140**	1.380	1.904	2.628	3.627	5.005	1.495	0.01040
1 1/2	1.900	**0.145**	1.610	2.592	4.173	6.719	10.82	2.036	0.01414
2	2.375	**0.154**	2.067	4.272	8.831	18.25	37.72	3.356	0.02330
2 1/2	2.875	**0.203**	2.469	6.096	15.05	37.16	91.75	4.788	0.03322
3	3.5	**0.216**	3.068	9.413	28.88	88.60	271.8	7.393	0.05130
3 1/2	4.0	**0.226**	3.548	12.59	44.66	158.5	562.2	9.888	0.06870
4	4.5	**0.237**	4.026	16.21	65.26	262.7	1,058	12.73	0.08840
5	5.563	**0.258**	5.047	25.47	128.6	648.8	3,275	20.01	0.1390
6	6.625	**0.280**	6.065	36.78	223.1	1,353	8,206	28.89	0.2006
8	8.625	**0.322**	7.981	63.70	508.4	4,057	32,380	50.03	0.3474
10	10.75	**0.365**	10.02	100.4	1,006	10,080	101,000	78.85	0.5475
12[3]	12.75	**0.375**	12.00	144.0	1,728	20,736	248,832	113.1	0.7854
12	12.75	0.406	11.938	142.5	1,701	20,311	242,470	111.93	0.7773
14 O.D.	14.0	0.437	13.126	172.3	2,262	29,684	389,638	135.32	0.9397
16 O.D.	16.0	0.500	15.000	225.0	3,375	50,625	759,375	176.72	1.2272
18 O.D.	18.0	0.562	16.876	284.8	4,806	81,110	1,368,820	223.68	1.5533
20 O.D.	20.0	0.593	18.814	354.0	6,660	125,292	2,357,244	278.00	1.9305
24 O.D.	24.0	0.687	22.626	511.9	11,583	262,078	5,929,784	402.07	2.7921
Schedule 60:									
8	8.625	0.406	7.813	61.04	476.9	3,726	29,113	47.94	0.3329
10	10.75	**0.500**	9.750	95.06	926.9	9,037	88,110	74.66	0.5185
12[3]	12.75	**0.500**	11.75	138.1	1,622	19,061	223,966	108.4	0.7527
12	12.75	0.562	11.626	135.16	1,571	18,269	212,399	106.16	0.7372
14 O.D.	14.0	0.593	12.814	164.20	2,104	26,961	345,480	128.96	0.8956
16 O.D.	16.0	0.656	14.688	215.74	3,169	46,543	683,618	169.44	1.1766
18 O.D.	18.0	0.718	16.564	274.37	4,545	75,277	1,246,884	215.49	1.4964
20 O.D.	20.0	0.812	18.376	337.68	6,205	114,026	2,095,342	265.21	1.8417
24 O.D.	24.0	0.937	22.126	489.56	10,832	239,669	5,302,913	384.50	2.6701

[1] All dimensions in inches except where otherwise noted. Wall thicknesses in boldface type are identical with thickness for standard-weight pipe in former tables. For other data on steel pipe of Schedule 40 and 60 dimensions, see Tables V and VI, pp. 361 and 362; for outside-diameter functions and moments of inertia, Table X, p. 858; for external surface, p. 697; for weight of water per foot of pipe, Table III, p. 738. The inside-diameter functions of wrought-iron pipe differ somewhat from the computed values for steel pipe given in above table, but the differences are well within the manufacturing tolerance permitted with both kinds of pipe.

[2] This column also represents contents in cubic feet per foot of length.

[3] Old "standard-weight" 0.375-in. wall and "extra-strong" 0.500-in. wall pipe may be substituted for Schedule 40 and Schedule 60, respectively, where agreeable to the purchaser and suitable for the service conditions.

TABLE XI.—INSIDE-DIAMETER FUNCTIONS OF SCHEDULES 80 AND 100 STEEL PIPE[1] CONFORMING TO ASA B36.10

Nominal pipe size	Outside diameter D	Wall thickness t	Inside diameter d	Inside diameter squared d^2	Inside diameter cubed d^3	Inside diameter, fourth power d^4	Inside diameter, fifth power d^5	Internal cross-sectional area	
								Square inches	Square feet[2]
Schedule 80:									
⅛	0.405	**0.095**	0.215	0.0462	0.0099	0.00214	0.000459	0.0363	0.00025
¼	0.540	**0.119**	0.302	0.0912	0.0275	0.00832	0.002513	0.0716	0.00050
⅜	0.675	**0.126**	0.423	0.1789	0.0757	0.03201	0.01354	0.1405	0.00098
½	0.840	**0.147**	0.546	0.2981	0.1628	0.08887	0.04852	0.2341	0.00163
¾	1.050	**0.154**	0.742	0.3506	0.4085	0.3031	0.2249	0.4324	0.00300
1	1.315	**0.179**	0.957	0.9158	0.8765	0.8388	0.8027	0.7193	0.00499
1¼	1.660	**0.191**	1.278	1.633	2.087	2.667	3.409	1.283	0.00891
1½	1.900	**0.200**	1.500	2.250	3.375	5.062	7.594	1.767	0.01225
2	2.375	**0.218**	1.939	3.760	7.290	14.14	27.41	2.953	0.02050
2½	2.875	**0.276**	2.323	5.396	12.54	29.12	67.64	4.238	0.02942
3	3.5	**0.300**	2.900	8.410	24.39	70.73	205.1	6.605	0.04587
3½	4.0	**0.318**	3.364	11.32	38.07	128.1	430.8	8.891	0.06170
4	4.5	**0.337**	3.826	14.64	56.00	214.3	819.8	11.50	0.07986
5	5.563	**0.375**	4.813	23.16	111.5	536.6	2,583	18.19	0.1263
6	6.625	**0.432**	5.761	33.19	191.2	1,102	6,346	26.07	0.1810
8	8.625	**0.500**	7.625	58.14	443.3	3,380	25,775	45.66	0.3171
10	10.75	0.593	9.564	91.47	874.8	8,367	80,020	71.84	0.4989
12	12.75	0.687	11.376	129.41	1,472	16,784	190,523	101.64	0.7058
14 O.D.	14.0	0.750	12.500	156.25	1,953	24,414	305,176	122.72	0.8522
16 O.D.	16.0	0.843	14.314	204.89	2,933	41,980	600,904	160.92	1.1175
18 O.D.	18.0	0.937	16.126	260.05	4,194	67,625	1,090,518	204.24	1.4183
20 O.D.	20.0	1.031	17.938	321.77	5,772	103,537	1,857,248	252.72	1.7550
24 O.D.	24.0	1.218	21.564	465.01	10,027	216,230	4,662,798	365.22	2.5362
Schedule 100:									
8	8.625	0.593	7.439	55.34	411.7	3,062	22,781	43.46	0.3018
10	10.75	0.718	9.314	86.75	799.5	7,526	69,357	68.13	0.4732
12	12.75	0.843	11.064	122.41	1,354	14,985	165,791	96.14	0.6677
14 O.D.	14.0	0.937	12.126	147.04	1,783	21,621	262,173	115.49	0.8020
16 O.D.	16.0	1.031	13.938	194.27	2,708	37,740	526,020	152.58	1.0596
18 O.D.	18.0	1.156	15.688	246.11	3,861	60,572	950,250	193.30	1.3423
20 O.D.	20.0	1.250	17.50	306.25	5,359	93,789	1,641,309	240.53	1.6703
24 O.D.	24.0	1.500	21.00	441.00	9,261	194,481	4,084,101	346.36	2.4053

[1] All dimensions in inches except where otherwise noted. Wall thicknesses of Schedule 80 pipe in boldface type are identical with thicknesses of extra-strong pipe in former tables. For other data on steel pipe of Schedule 80 and 100 dimensions see Tables V and VI, pp. 361 and 362; for outside-diameter functions and moments of inertia, see Table X, p. 358; for external surface, see p. 697; for weight of water per foot of pipe, Table III, p. 738. The inside-diameter functions of wrought-iron pipe differ somewhat from the computed values for steel pipe given in above table, but the differences are well within the manufacturing tolerance permitted with both kinds of pipe.

[2] This column also represents contents in cubic feet per foot of length.

TABLE XII.—INSIDE-DIAMETER FUNCTIONS OF SCHEDULES 120, 140, AND 160 STEEL PIPE[1] CONFORMING TO ASA B36.10

Nominal pipe size	Outside diameter D	Wall thickness t	Inside diameter d	Inside diameter squared d^2	Inside diameter cubed d^3	Inside diameter, fourth power d^4	Inside diameter, fifth power d^5	Internal cross-sectional area, Square inches	Internal cross-sectional area, Square feet[2]
Schedule 120:									
4	4.5	0.437	3.626	13.15	47.67	172.9	626.8	10.33	.0717
5	5.563	0.500	4.563	20.82	95.00	433.5	1,978	16.35	.1136
6	6.625	0.562	5.501	30.26	66.5	915.7	5,037	23.77	.1650
8	8.625	0.718	7.189	51.68	71.6	2,671	19,202	40.59	.2819
10	10.75	0.843	9.064	82.16	44.7	6,750	61,179	64.53	.4481
12	12.75	1.000	10.750	115.56	1,242	13,355	143,563	90.76	.6303
14 O.D.	14.0	1.062	11.876	141.04	1,675	19,892	236,239	110.77	.7693
16 O.D.	16.0	1.218	13.564	183.98	2,496	33,849	459,133	144.50	.0035
18 O.D.	18.0	1.343	15.31	234.52	3,591	54,999	842,254	184.19	.2791
20 O.D.	20.0	1.500	17.000	289.00	4,913	83,521	,419,85	226.98	,5762
24 O.D.	24.0	1.750	20.500	420.25	8,615	76,610	,620,506	330.06	2.2921
Schedule 140:									
8	8.625	0.812	7.00	49.0	343.	2,402	16,819	38.50	.2673
10	10.75	1.000	8.750	76.56	669.9	5,862	51,291	60.13	.4176
12	12.75	1.125	10.50	110.25	,158	12,155	127,628	86.59	.6013
14 O.D.	14.0	1.250	11.50	132.25	,521	17,490	201,136	103.87	0.7213
16 O.D.	16.0	1.437	13.126	172.29	2,262	29,686	389,670	135.32	.9397
18 O.D.	18.0	1.562	14.87	221.3	,292	48,972	728,502	173.80	,2070
20 O.D.	20.0	1.750	16.50	272.25	4,492	74,12	,222,98	213.82	.4849
24 O.D.	24.0	2.062	19.87	395.06	,852	56,06	,102,022	310.28	2.1547
Schedule 160:									
½	0.84	0.187	0.46	0.217	0.101	0.0471	0.02197	0.1706	0.00118
¾	1.05	0.218	0.61	0.337	0.231	0.142	0.08726	0.296	0.00206
1	1.31	0.250	0.81	0.664	0.541	0.441	0.3596	0.521	0.00362
1¼	1.66	0.250	1.16	1.34	1.56	1.81	2.11	1.05	0.00734
1½	1.90	0.281	1.33	1.790	2.39	3.20	4.28	1.406	0.00976
2		0.343	1.68	2.85	4.81	8.13	13.7	2.24	0.01556
2½	2.	0.375	2.125	4.51	9.596	20.3	43.3	3.54	0.02463
3	3.5	0.437	2.62	6.896	18.1	47.5	124.	5.41	0.03761
4	4.5	0.531	3.43	11.8	40.64	139.	480.	9.28	0.06447
5	5.56	0.625	4.31	18.6	80.23	346.	1,49	14.6	0.1015
6	6.625	0.718	5.18	26.9	139.	725.	3,76	21.1	0.1469
8	8.6	0.906	6.81	46.4	316.	2,15	14,67	36.4	0.2532
10	10.75	1.125	8.50	72.	614.	5,22	44,37	56.7	0.3941
12	12.75	1.312	10.1	102.5	1,03	10,51	106,46	80.5	0.5592
14 O.D.	14.0	1.406	11.18	125.1	1,40	15,66	175,29	98.3	0.6827
16 O.D.	16.0	1.562	12.87	165.	2,13	27,48	353,92	130.2	0.9043
18 O.D.	18.0	1.750	14.50	210.25	3,04	44,20	640,973	165.1	1.1467
20 O.D.	20.0	1.937	16.12	260.0	4,19	67,625	1,090,51	204.2	1.4183
24 O.D.	24.0	2.312	19.37	375.	7,27	140,94	2,730,99	294.8	2.0476

[1] All dimensions are in inches except where otherwise noted. For other data on steel pipe of Schedule 120, 140, and 160 dimensions see Tables V and VI, pp. 361 and 362; for outside-diameter functions and moments of inertia, see Table X, p. 858; for external surface, p. 697.

[2] This column also represents contents in cubic feet per foot of length.

ASTM Standard Specifications for BLACK AND HOT-DIPPED ZINC-COATED (GALVANIZED) WELDED AND SEAMLESS STEEL PIPE FOR ORDINARY USES

Serial Designation A120-44 (ASA G8.7)

Abstracted[1]

1. (*a*) These specifications cover black and hot-dipped galvanized welded and seamless steel pipe. Pipe ordered under these specifications is intended for ordinary uses such as low-pressure service in steam, water, and gas lines and is not intended for close bending or coiling, or high-temperature service. No physical tests are specified on pipe made to these specifications, except hydrostatic tests which shall be made at the mills, as these specifications are intended to cover pipe purchased mainly from jobber's stocks.

(*b*)[2] When used for waterworks service, pipe furnished shall be standard weight, unless otherwise specified. (See also Standard Specifications for Steel Water Pipe of Sizes Up to But Not Including 30 in., AWWA 7A.4, of the American Water Works Association, abstracted on page 406.)

NOTE.—When tension and flattening or bending tests are required, pipe should be ordered in accordance with the Standard Specifications for Welded and Seamless Steel Pipe (ASTM Designation A53) of the American Society for Testing Materials. In the case of galvanized pipe so ordered, the base metal shall be made and tested in accordance with the Standard Specifications A53 and the pipe shall be galvanized and the coating tested in accordance with these Standard Specifications A120. When specified, results of the physical tests on the base material shall be reported to the purchaser.

2. Process.—(*a*) The steel for both welded and seamless pipe shall be made by one or more of the following processes: open-hearth, electric-furnace, or acid-bessemer. The steel for welded pipe shall be of soft weldable quality.

(*b*) Welded pipe 3 in. and under in nominal diameter may be butt-welded, unless otherwise specified. Welded pipe over 3 in. in nominal diameter shall be lap-welded or electric-welded.

3. Galvanized Pipe.—(*a*) Galvanized pipe shall be coated with zinc inside and outside by the hot-dip process.

(*b*) The zinc used for the coating shall be any grade of zinc conforming to the Standard Specifications for Slab Zinc (Spelter) (ASTM Designation: B 6) of the American Society for Testing Materials.

4. Hydrostatic Test.—Each length of pipe shall be tested at the mill to the hydrostatic pressures prescribed in the accompanying table. Welded pipe 2 in. and larger shall be jarred near one end while under test pressure.

[1] Where complete specifications are required, or revisions of some later date, reference may be made to the standards and/or tentative standards of the American Society for Testing Materials. Bound volumes of ASTM specifications and printed copies of individual specifications can be purchased by addressing headquarters at 260 South Broad St., Philadelphia, Pa.

[2] Emergency Alternate Provisions, issued Aug. 24, 1942.

HYDROSTATIC TEST PRESSURES FOR WELDED AND SEAMLESS STEEL PIPE[a]

(Table I of ASTM A120)

(Pressures expressed in pounds per square inch)

Size (nominal) inside diameter), in.	Standard-weight pipe			Extra-strong pipe			Double-extra-strong pipe		
	Butt-welded	Lap-welded, electric-welded, and Grade A	Grade B	Butt-welded	Lap-welded, electric-welded, and Grade A	Grade B	Butt-welded	Lap-welded, electric-welded, and Grade A	Grade B
⅛ to 1, incl.....	700	700[b,c]	700[c]	850	850[b,c]	850[c]	1,000	1,000[b,c]	1,000[c]
1¼ to 3, incl....	800	1,000	1,100	1,100	1,500	1,600	1,200	1,800	1,900
3½ to 6, incl....	1,200[d]	1,200	1,300	1,700[d]	1,700	1,800	.....	2,000	2,100
8..............	...	1,200	1,300	.....	1,700	2,400	.....	2,800	2,800
10 to 12, incl....	...	1,000	1,200	.....	1,600	1,900			

[a] For pipe over 12 in. in nominal pipe size, the test pressures should be calculated by the formula $p = 2St/D$; where p = pressure, psi; S = fiber stress, 60 per cent of the specified yield point, psi; t = thickness of wall, in.; D = outside diameter, in.
[b] Lap-welded pipe is not made below 1¼-in. size.
[c] Seamless pipe in these small sizes will probably need to be cold drawn.
[d] Butt-weld pipe is not made in sizes larger than 4 in. nominal.

5. Weight of Coating.—The weight of zinc coating shall be not less than 2.0 oz per sq ft of total coated surface. The weight of coating expressed in ounces per square foot shall be calculated by dividing the total weight of zinc, inside plus outside, by the total area, inside plus outside, of the surface coated.

6. Weight of Coating Test.—The weight of zinc coating shall be determined by a stripping test in accordance with the Standard Methods of Test for Weight of Coating on Zinc-Coated (Galvanized) Iron or Steel Articles (ASTM Designation: A90) of the American Society for Testing Materials. The total zinc on each specimen shall be determined in a single stripping operation and the average of the results from the two specimens from each pipe shall be the weight of zinc coating.

7. Test Specimens.—Test specimens for determination of weight of coating shall be cut approximately 4 in. in length from opposite ends of the lengths of pipe selected for testing.

8. Number of Tests.—(*a*) Two test specimens for the determination of weight of coating shall be taken, one from each end of one length of galvanized pipe, selected at random from each lot of 500 lengths or fraction thereof of each size.

(*b*) Each length of pipe shall be subjected to the hydrostatic test specified in Section 4.

9. Retests.—If the weight of coating of any lot does not conform to the requirements specified in Section 5, retests of two additional pipes from the same lot shall be made, each of which shall conform to the requirements specified.

10. Standard Weights.—(*a*) The standard weights with the corresponding wall thicknesses for pipe of various nominal inside diameters are prescribed in the accompanying table.

(*b*) Nipples shall be cut from pipe of the same weight and quality described in these specifications.

Standard Weights and Dimensions of Welded and Seamless Steel Pipe[a]
(Table II of ASTM A120)

Size (nominal inside diameter), in.	Outside diameter, in.	Number of threads per inch	Standard-weight pipe				Extra-strong pipe				Double-extra-strong[b] pipe	
			Schedule 30		Schedule 40		Schedule 60		Schedule 80			
			Thickness, in.	Weight of pipe per linear foot, threaded and with couplings, lb	Thickness, in.	Weight of pipe per linear foot, threaded and with couplings, lb	Thickness, in.	Weight of pipe per linear foot, plain ends, lb	Thickness, in.	Weight of pipe per linear foot, plain ends, lb	Thickness, in.	Weight of pipe per linear foot, plain ends, lb
1/8	0.405	27	.....	.....	0.068	0.24	.....	.....	0.095	0.31		
1/4	0.540	18	.....	.....	0.088	0.42	.....	.....	0.119	0.54		
3/8	0.675	18	.....	.....	0.091	0.57	.....	.....	0.126	0.74		
1/2	0.840	14	.....	.....	0.109	0.85	.....	.....	0.147	1.09	0.294	1.71
3/4	1.050	14	.....	.....	0.113	1.13	.....	.....	0.154	1.47	0.308	2.44
1	1.315	11 1/2	.....	.....	0.133	1.68	.....	.....	0.179	2.17	0.358	3.66
1 1/4	1.660	11 1/2	.....	.....	0.140	2.28	.....	.....	0.191	3.00	0.382	5.21
1 1/2	1.900	11 1/2	.....	.....	0.145	2.73	.....	.....	0.200	3.63	0.400	6.41
2	2.375	11 1/2	.....	.....	0.154	3.68	.....	.....	0.218	5.02	0.436	9.03
2 1/2	2.875	8	.....	.....	0.203	5.82	.....	.....	0.276	7.66	0.552	13.70
3	3.500	8	.....	.....	0.216	7.62	.....	.....	0.300	10.25	0.600	18.58
3 1/2	4.000	8	.....	.....	0.226	9.20	.....	.....	0.318	12.51		
4	4.500	8	.....	.....	0.237	10.89	.....	.....	0.337	14.98	0.674	27.54
5	5.563	8	.....	.....	0.258	14.81	.....	.....	0.375	20.78	0.750	38.55
6	6.625	8	.....	.....	0.280	19.18	.....	.....	0.432	28.57	0.864	53.16
8	8.625	8	0.277	25.00	0.322	28.81	.....	.....	0.500	43.39	0.875	72.42
*10	10.750	8	0.307	35.00	0.365	41.13	0.500	54.74				
**12	12.750	8	0.3[illegible]0	45.00	**0.375	50.71	**0.500	65.41				

[a] Sizes larger than those shown in the table are measured by their outside diameter. These larger sizes will be furnished with plain ends, unless otherwise specified. The weights will correspond to the manufacturers' published standards although it is possible to calculate the theoretical weights for any given size and wall thickness on the basis of 1 cu. in. of steel weighing 0.2833 lb.

[b] The American Standard for Wrought-Iron and Wrought-Steel Pipe (ASA No.: B36.10) has assigned no schedule number to Double-extra-strong pipe.

* Standard weight pipe 10 in. in nominal size is also available with 0.279-in. wall thickness, but this wall is not covered by a schedule number.

** Owing to a departure from the "Standard-weight" and "Extra-strong" wall thicknesses for the 12-in. nominal size, Schedules 40 and 60, in Table 2 of the American Standard for Wrought-Iron and Wrought-Steel Pipe (ASA No.: B36.10) the regular "Standard" and "Extra-strong" wall thicknesses (0.375 in. and 0.500 in.) have been substituted.

NOTE.—Where more than one weight is listed under the same size and class, the order should definitely specify the wall thickness desired.

11. Permissible Variations in Weight and Dimensions. (*a*) *Weight.*—The weight of the pipe shall not vary from that prescribed in the accompanying table by more than plus or minus 5 per cent for standard-weight and extra-strong pipe nor more than plus or minus 10 per cent for double-extra-strong pipe.

(b) *Diameter.*—For pipe 1½ in. and under in nominal diameter, the outside diameter at any point shall not vary more than 1/64 in. over nor more than 1/32 in. under the standard specified. For pipe 2 in. and over in nominal diameter, the outside diameter shall not vary more than plus or minus 1 per cent from the standard specified.

(c) *Thickness.*—The minimum wall thickness at any point shall not be more than 12.5 per cent under the nominal wall thickness specified.

12. Lengths.—Unless otherwise specified, pipe lengths shall be in accordance with the following regular practice:

(a) Standard-weight pipe shall be in random lengths of 16 to 22 ft, but not more than 5 per cent of the total number of lengths may be "jointers," which are two pieces coupled together. When ordered with plain ends, 5 per cent may be in lengths of 12 to 16 ft.

(b) Extra-strong and double-extra-strong pipe shall be in random lengths of 12 to 22 ft. Five per cent may be in lengths of 6 to 12 ft.

13. Workmanship.—Unless otherwise specified, pipe shall conform to the following regular practice:

(a) *Ends.*—Each end of standard-weight welded pipe shall be threaded. Extra-strong welded pipe and standard-weight and extra-strong seamless pipe and all double-extra-strong pipe shall be furnished with plain ends.

(b) *Threads.*—All threads shall be in accordance with the American Standard Pipe Thread[1] and cut so as to make a tight joint when the pipe is tested at the mill to the specified internal hydrostatic pressure.

(c) *Couplings.*[2]—Each length of threaded pipe shall be provided with one coupling, having clean-cut threads of such a pitch diameter as to make a tight joint. Couplings may be of wrought iron or steel.

14. Finish.—(a) The finished pipe shall be reasonably straight and free from injurious defects. All burns at the ends of the pipe shall be removed.

(b) The zinc coating on galvanized pipe shall be free from injurious defects or excessive roughness.

(c) *Water Works Service.*[2]—In cases where the water or ground, or both, have been known to be corrosive, due consideration should be given to the protection of steel water pipe. It is recommended that if coatings are used they conform to the Standard Specifications for Coal-Tar Enamel Protective Coatings for Steel Water Pipe (AWWA No. 7A.6) of the American Water Works Association.

15. Marking.—Each length of pipe shall be legibly marked by rolling, stamping, or stenciling to show the name or brand of the manufacturer, the length, and ASTM A120; except that for small diameter pipe which is bundled, this information may be marked on a tag securely attached to each bundle.

16. Inspection.—The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works that concern the manufacture of the material ordered. The manufacturer shall afford the inspector, without charge, all reasonable facilities to satisfy him that the material is being furnished in accordance with these specifications. All tests and inspection shall be made

[1] See abstract of American Standard for Pipe Threads, ASA B2.1, p. 481.

[2] For sizes 2 in. and smaller, it is commercial practice to furnish straight tapped couplings for Schedule 40 (standard weight) pipe. If taper tapped couplings are required, line pipe in accordance with API 5L (see p. 1131) should be ordered, thread lengths to be in accordance with ASA B2.1.

[2] Emergency Alternate Provisions, issued Aug. 24, 1942.

at the place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

17. Rejection.—Each length of pipe that develops injurious defects during shop working or application operations will be rejected, and the manufacturer shall be notified.

ASTM Standard Specifications for WELDED AND SEAMLESS STEEL PIPE

Serial Designation A53-44 (ASA B36.1)

Abstracted[1]

1. (*a*) These specifications cover black and hot-dipped-galvanized welded and seamless steel pipe. Pipe ordered under these specifications is intended for coiling, bending, flanging, and other special purposes, and is suitable for fusion welding. Butt-welded pipe is not intended for flanging. The purposes for which the pipe is intended should be stated in the order. When seamless or electric-resistance-welded pipe is ordered for close coiling, cold bending, or for forge welding, grade A should be specified rather than grade B.

(*b*) *Galvanized Pipe.*—When pipe ordered under these specifications is to be galvanized, the tension, flattening, and bend tests shall be made on the base material before galvanizing, and the pipe shall be galvanized and the coating tested in accordance with ASTM specification A120 (see page 369).

Note.—If impracticable to make the physical tests on the base material before galvanizing, such tests may be made on galvanized samples, and any flaking or cracking of the zinc coating shall not be considered cause for rejection.

2. Process.—(*a*) The steel for both welded and seamless pipe shall be made by one or more of the following processes: open-hearth, electric-furnace, or acid-bessemer, except that electric-resistance-welded pipe ⅛ and ¼ in. in diameter shall be made from open-hearth steel. The steel for furnace-welded pipe shall be of soft weldable quality.

(*b*) Furnace-welded pipe 3 in. and under in nominal diameter may be butt-welded, unless otherwise specified. Furnace-welded pipe over 3 in. in nominal diameter shall be lap-welded.

3. Chemical Composition.—The steel shall conform to the following ladle analysis requirement as to chemical composition:

	Lap-welded	Seamless or electric-resistance-welded
Carbon, max, per cent		[a]
Phosphorus, max, per cent:		
Open-hearth or electric-furnace	0.06	0.045
Acid-bessemer	....	0.11

[a] Electric-resistance welded pipe ⅛ and ¼ in. in diameter is obtainable in Grade A only and the carbon content shall not exceed 0.10 per cent.

[1] For complete specification, reference may be made to ASTM A53 (see note, p. 369).

4. Check Analysis.—An analysis of two pipes from each lot of 500 lengths or fraction thereof may be made by the purchaser. Drillings for analysis shall be taken from several points around each pipe selected for analysis. The phosphorus content of open-hearth or electric-furnace steel thus determined shall not exceed that specified in Section 3 by more than 25 per cent. For acid-bessemer seamless or electric-resistance-welded steel pipe, the phosphorus content shall not exceed the maximum specified in Section 3. For electric resistance-welded pipe 1/8 and 1/4 in. in diameter, the carbon content shall not exceed 0.12 per cent.

5. Tensile Properties.—(*a*) The material shall conform to the requirements as to tensile properties prescribed in the accompanying table.

(*b*) The yield point shall be determined by the drop of the beam or halt in the gage of the testing machine, or other approved method.

TENSILE REQUIREMENTS
(Table I of ASTM A-53)

	Furnace-welded		Seamless or electric-resistance-welded[d]	
	Acid-bessemer	Open-hearth or electric-furnace	Grade A	Grade B
Tensile strength, min, psi	50,000	45,000	48,000	60,000
Yield point, min, psi	30,000	25,000	30,000	35,000
Elongation in 8 in., min, per cent	18[a]	20[a]		
Elongation in 2 in., min, per cent	......	30[b]		
Elongation, in 2 in., min, per cent				
Basic minimum elongation for walls 5/16 in. and over in thickness, longitudinal strip tests, and for all small sizes tested in full section	......	......	35	30
When standard round 2-in. gage length test specimen is used	......	......	28	22
For longitudinal strip tests a deduction for each 1/32 in. decrease in wall thickness below 5/16 in. from the basic minimum elongation of the following percentage	......	......	1.75[c]	1.50[c]

[a] Gage distances for measuring elongation on furnace-welded pipe of nominal sizes 3/4 in. and smaller shall be as follows:

Nominal Size, in.	Gage Length, in.
3/4 and 1/2	6
3/8 and 1/4	4
1/8	2

[b] When standard round 2-in.-gage-length test specimen is used.

[c] The following table gives the computed minimum values:

Wall thickness, in.	Elongation, min per cent	
	Grade A	Grade B
5/16 (0.312)	35.00	30.00
9/32 (0.281)	33.25	28.50
1/4 (0.250)	31.50	27.00
7/32 (0.219)	29.75	25.50
3/16 (0.188)	28.00	24.00
5/32 (0.156)	26.25	22.50
1/8 (0.125)	24.50	21.60
3/32 (0.094)	22.75	19.50
1/16 (0.062)	21.00	18.00

[d] Tension tests shall not be required for electric-resistance-welded pipe 1/8 and 1/4 in. in diameter.

6. Bending Properties.—For pipe 2 in. and under in nominal diameter, a sufficient length of pipe shall stand being bent cold through 90 deg around a cylindrical mandrel the diameter of which is 12 times the nominal diameter of the pipe without developing cracks at any portion and without opening the weld. When ordered for close coiling, the pipe shall stand being bent cold through 180 deg around a cylindrical mandrel the diameter of which is eight times the nominal diameter of the pipe, without failure. Double-extra-strong pipe over 1¼ in. in diameter need not be subjected to the bend test.

7. Flattening Test.—(*a*) The flattening test shall be made on standard-weight and extra-strong pipe over 2 in. in nominal diameter. It shall not be required for double-extra-strong pipe or Grade C pipe.[1]

(*b*) For lap-welded and butt-welded pipe the test section shall be 4 to 6 in. in length and the weld shall be located 45 deg from the line of direction of the applied force.

(*c*) For electric-resistance-welded pipe, the test section shall be 4 to 6 in. in length and the weld shall be located 90 deg from the line of direction of the applied force.

(*d*) For seamless pipe the test section shall not be less than 2½ in. in length.

(*e*) The test shall consist in flattening a section of pipe between parallel plates until the opposite walls meet.

For welded pipe, no opening in the weld shall take place until the distance between the plates is less than three-fourths of the original outside diameter for butt-weld, or two-thirds the outside diameter for lap-weld and electric-resistance-weld, and no cracks or breaks in the metal elsewhere than in the weld shall occur until the distance between the plates is less than shown below. For seamless pipe no breaks or cracks in the metal, shall occur until the distance between the plates is less than that shown below:

Kind of Pipe	Distance between Plates, H
For butt-welded pipe	60 per cent of outside diameter
For lap-welded pipe	One-third the outside diameter
For electric-resistance-welded pipe, Grades A and B	One-third the outside diameter
For seamless pipe, Grades A and B	To the distance H developed by the following formula:

$$H = \frac{(1 + e)t}{e + t/D}$$

where H = distance between flattening plates, in.[2]
t = nominal wall thickness of pipe, in.
D = actual outside diameter of pipe, in.
e = deformation per unit length (constant for a given grade of steel, 0.09 for Grade A and 0.07 for Grade B).

8. Hydrostatic Test.—Each length of pipe shall be tested at the mill to the hydrostatic pressures described in Table II. Welded pipe 2 in. and larger shall be jarred near one end while under test pressure.

NOTE.—"Table II" is identical with "Table I" of ASTM Specification A 120 reproduced on page 370. Electric-resistance-welded pipe receives the same hydrostatic test as lap-welded and Grade A seamless pipe.

[1] Emergency Alternate Provisions, issued Jan. 30, 1943.

[2] Calculated values of H are given in Table V of Appendix II of ASTM A53 not reproduced here.

9. Test Specimens.—(*a*) Tension test specimens shall be cut longitudinally from the pipe and not flattened between gage marks. The sides of specimens shall be parallel between gage marks. If desired, the tension test specimen may consist of a full section of the pipe. When impracticable to pull a test specimen in full thickness the ASTM standard 2-in. gage length tension test specimen may be used.[1]

(*b*) Test specimens for the bend and flattening tests shall consist of sections cut from a pipe. Specimens for flattening tests shall be smooth on the ends and free from burrs, except when made on crop ends taken from welded pipe.

(*c*) All specimens shall be tested at room temperature.

10. Number of Tests.—(*a*) One of each of the tests specified in Sections 5, 6, and 7 shall be made on one length of pipe from each lot of 500 lengths or fraction thereof of each size.

(*b*) In the case of welded pipe ordered for "flanging" the crop ends cut from each length shall stand the flattening test specified in Section 7.

(*c*) Each length of pipe shall be subjected to the hydrostatic test specified in Section 8.

11. Retests.—(*a*) If the results of the physical tests of any lot do not conform to the requirements specified in Sections 5, 6, and 7, retests may be made on additional pipe of double the original number from the same lot, each of which shall conform to the requirements specified.

(*b*) If any section fails when flattening tests are made on the crop ends of each length of welded pipe, other pieces from the length may be cut until satisfactory tests are obtained, otherwise the length shall be rejected.

12. Standard Weights.—(*a*) The standard weights with the corresponding wall thicknesses for pipe of various nominal inside diameters are prescribed in Table III of the specification. ["Table III" is identical with "Table II" of ASTM Specification A 120 reproduced on page 371.]

NOTE.—For dimensions and weights of welded and seamless steel pipe conforming to the schedules of wall thickness of ASA B36.10, see Tables V and VI, pages 361 and 362.

(*b*) Nipples shall be cut from pipe of the same weight and quality described in these specifications.

13. Permissible Variations in Weights and Dimensions.—(*a*) *Weight.*—The weight of the pipe shall not vary from that prescribed in "Table III" by more than plus or minus 5 per cent for standard-weight and extra-strong pipe nor more than plus or minus 10 per cent for double-extra-strong pipe.

(*b*) *Diameter.*—For pipe 1½ in. and under in nominal diameter, the outside diameter at any point shall not vary more than $\frac{1}{64}$ in. over nor more than $\frac{1}{32}$ in. under the standard specified. For pipe 2 in. and over in nominal diameter, the outside diameter shall not vary more than plus or minus 1 per cent from the standard specified.

(*c*) *Thickness.*—The minimum wall thickness at any point shall be not more than 12.5 per cent under the nominal wall thickness specified.

14. Lengths.—Unless otherwise specified, pipe lengths shall be in accordance with the following regular practice:

(*a*) Standard-weight pipe shall be in random lengths of 16 to 22 ft., but not more than 5 per cent of the total number of lengths may be "jointers," which are

[1] See Fig. 3 p. 309.

two pieces coupled together. When ordered with plain ends, 5 per cent may be in lengths of 12 to 16 ft.

(b) Extra-strong and double-extra-strong pipe shall be in random lengths of 12 to 22 ft. Five per cent may be in lengths of 6 to 12 ft.

15. Workmanship.—Unless otherwise specified, pipe shall conform to the following regular practice:

(a) *Ends.*—Each end of standard-weight welded pipe shall be threaded. Extra-strong welded pipe and standard-weight and extra-strong seamless pipe and all double-extra-strong pipe shall be furnished with plain ends.

(b) *Threads.*—All threads shall be in accordance with the American Standard Pipe Thread[1] and cut so as to make a tight joint when the pipe is tested at the mill to the specified internal hydrostatic pressure. The variation from the standard, when tested with the standard working gage, shall not exceed one and one-half turns either way.

(c) *Couplings.*[2]—Each length of threaded pipe shall be provided with one coupling, having clean-cut threads of such a pitch diameter as to make a tight joint. Couplings may be of wrought iron or steel.

16. Finish.—The finished pipe shall be reasonably straight and free from injurious defects. All burrs at the ends of the pipe shall be removed.

17. Marking.—Each length of pipe shall be legibly marked by rolling, stamping, or stenciling to show the name or brand of the manufacturer; the type of pipe (that is, lap-welded, electric-resistance-welded A, electric-resistance-welded B, seamless A, or seamless B; where acid-bessemer steel is used in seamless or electric-resistance-welded pipe, the word "bessemer" shall be added before the letter A or B); XS for extra strong, XXS for double extra strong; ASTM A53; and the length; except that for small diameter pipe which is bundled, this information may be marked on a tag securely attached to each bundle.

18. Inspection.—The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works that concern the manufacture of the material ordered. The manufacturer shall afford the inspector, without charge, all reasonable facilities to satisfy him that the material is being furnished in accordance with these specifications. All tests (except check analysis) and inspection shall be made at the place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

19. Rejection.—Each length of pipe that develops injurious defects during shop working or application operations will be rejected, and the manufacturer shall be notified.

Appendix.—The schedule pipe thicknesses of ASA Standard B36.10 are reproduced in a table [Table V, page 361] in appendix to this specification.

[1] A complete description of the American Standard Pipe Threads applicable to pipe, valves, and fittings is contained in the American Standard for Pipe Threads (ASA B2.1). See abstract, p. 481.

[2] For sizes 2 in. and smaller, it is commercial practice to furnish straight-tapped couplings for Schedule 40 (standard weight) pipe. If taper-tapped couplings are required, line pipe in accordance with API 5L (see p. 483) should be ordered, thread lengths to be in accordance with ASA B2.1.

ASTM Tentative Specifications for LAP-WELDED AND SEAMLESS STEEL PIPE FOR HIGH-TEMPERATURE SERVICE

Serial Designation A106-44T (ASA B36.3)

Abstracted[1]

These specifications cover lap-welded and seamless steel pipe for high temperatures. In particular this contemplates temperatures up to 850 F for power piping and 1000 F for oil piping. Higher temperatures may be used at the discretion of the designing engineer, provided suitable reduction in working stresses is made. Pipe ordered under these specifications shall be suitable for bending, flanging, and similar forming operations. Low-carbon Grade A pipe should be used for close coiling, cold bending, or for forge welding. For use in central stations having steam-service pressures of 400 psi or over and for other applications where a superior grade of pipe is required, additional tests of an optional nature are provided.

Process.—The steel for lap-welded pipe shall be of good weldable quality made by the open-hearth process. The steel for seamless pipe Grade A, and Grade B (silicon-killed) shall be killed steel made by one or more of the following processes: open-hearth or electric-furnace. The steel for seamless pipe Grade B (bessemer) shall be killed steel made by the deoxidized acid-bessemer process.

Welded pipe 2 in. or over in diameter shall be made by the lap-weld process. All pipe 1½ in. or under in nominal diameter shall be made by the seamless process, and may be either hot-finished or cold drawn and annealed. Unless otherwise specified, seamless pipe 2 in. or over in nominal diameter shall be furnished hot-finished.

Chemical Composition.—The steel for welded and seamless pipe shall conform to the following ladle and check analysis requirements as to chemical composition:

	Welded open-hearth	Seamless		
		Grade A	Grade B, silicon-killed	Grade B acid-bessemer killed
Carbon, max, per cent	....	0.25	0.35	0.25
Manganese, per cent	0.30 to 0.60	0.30 to 0.90	0.35 to 1.00	0.35 to 1.00
Phosphorus, max, per cent	0.045	0.04	0.04	0.11
Sulfur, max, per cent	0.06	0.06	0.06	0.06
Silicon, min, per cent			0.10	0.10

Tensile Properties.—The material shall conform to the requirements as to tensile properties given in the accompanying table.

[1] For complete specification, reference may be made to ASTM A106 (see note p. 369).

TENSILE REQUIREMENTS
(Table II of ASTM A-106)

	Welded open-hearth	Seamless			
		Grade A		Grade B	
Tensile strength, min, psi	45,000	48,000		60,000	
Yield point, min, psi	25,000	30,000		35,000	
Elongation in 8 in., min, per cent	22				
		Longitudinal	Transverse	Longitudinal	Transverse
Elongation in 2 in., min, per cent					
Basic minimum elongation for walls 5/16 in. and over in thickness, strip tests, and for all small sizes tested in full section	......	35	25	30	16.5
When standard round 2-in. gage length test specimen is used	......	28	20	22	12
For strip tests, a deduction for each 1/32 in. decrease in wall thickness below 5/16 in. from the basic minimum elongation of the following percentage	......	1.75[a]	1.25[a]	1.50[a]	1.00[a]

[a] The following table gives the computed minimum values:

Wall thickness, in.	Elongation in 2 in., min per cent			
	Grade A		Grade B	
	Longitudinal	Transverse	Longitudinal	Transverse
5/16 (0.312)	35.00	25.00	30.00	16.50
9/32 (0.281)	33.25	23.75	28.50	15.50
1/4 (0.250)	31.50	22.50	27.00	14.50
7/32 (0.219)	29.75	.....	25.50	
3/16 (0.188)	28.00	.....	24.00	
5/32 (0.156)	26.25	.....	22.50	
1/8 (0.125)	24.50	.....	21.00	
3/32 (0.094)	22.75	.....	19.50	
1/16 (0.062)	21.00	.....	18.00	

Bending Properties.—For Grades A and B seamless pipe 2 in. or under in nominal diameter, a sufficient length of pipe shall stand being bent cold through 90 deg around a cylindrical mandrel, the diameter of which is twelve times the nominal diameter of the pipe, without developing cracks. When ordered for close coiling (Grade A), the pipe shall stand being bent cold through 180 deg around a cylindrical mandrel, the diameter of which is eight times the nominal diameter of the pipe without failure.

Flattening Tests.—For welded pipe, the crop end cut from each end of each length of pipe shall be flattened cold between parallel plates with the weld

located 90 deg from the line of direction of the applied force, until opposite walls of the pipe meet. No opening in the weld shall take place until the distance between the plates is less than two-thirds of the original outside diameter of the pipe. No cracks or breaks in the metal elsewhere than in the lap-weld shall occur until the distance between the plates is less than one-third of the original outside diameter of the pipe. Evidence of laminations or burnt material shall not develop during the entire flattening process. Precautions shall be taken so that crop ends can be identified with respect to the length from which they are cut.

For Grades A and B seamless pipe over 2 in. in nominal diameter a section of pipe not less than 2½ in. in length shall be flattened cold between parallel plates until the opposite walls of the pipe meet. No cracks or breaks in the metal shall occur until the distance between the plates is less than that calculated for the value of H by the following formula:

$$H = \frac{(1 + e)t}{e + t/D}$$

where H = distance between flattening plates, in.[1]
t = nominal wall thickness of pipe, in.
D = actual outside diameter of pipe, in.
e = deformation per unit length (constant for a given grade of steel, 0.09 for Grade A and 0.07 for Grade B).

Evidence of laminations or burnt material shall not develop during the entire flattening process.

Hydrostatic Tests.—Unless otherwise agreed upon between the manufacturer and purchaser, each length of pipe shall be tested by the manufacturer to a hydrostatic pressure which will produce in the pipe wall a stress of 50 per cent of the minimum specified yield point at room temperature. This pressure shall be determined by the formula:

$$p = \frac{2St}{D}$$

where p = the minimum hydrostatic test pressure, psi.
S = 0.50 times the minimum specified yield point at room temperature, psi.
t = the nominal wall thickness, in.
D = the outside diameter, in.

The maximum hydrostatic test pressure shall not exceed 2,500 psi for nominal sizes 3 in. and under, or 2,800 psi for all nominal sizes over 3 in. The hydrostatic pressure shall be maintained for not less than 5 sec.

NOTE.—When requested by the purchaser and so stated in the order, pipe in sizes 14 in. in nominal diameter and smaller shall be tested to one and one-half times the specified working pressure, provided the fiber stress corresponding to those test pressures does not exceed one-half the minimum specified yield point of the material, as determined by the above formula. When one and one-half times the working pressure exceeds 2,800 psi, the hydrostatic test pressure shall be a matter of agreement between the purchaser and the manufacturer.

Test Specimens.—Specimens cut longitudinally or in the case of seamless pipe, either longitudinally or transversely, shall be acceptable for the tension test.

[1] The H values calculated for sizes from 2½ to 24 in., incl., are shown in Table IV of these specifications, not reproduced here.

Tension specimens from welded pipe shall be taken at a point approximately 90 deg from the weld.

The longitudinal tension test may be made in full section of the pipe up to the capacity of the testing machine. For larger sizes, the tension test specimens shall consist of strips cut from the pipe. The width of these specimens shall be 1½ in. and shall have a gage length of 8 in. for welded pipe and 2 in. for seamless pipe. For seamless pipe, when the pipe wall thickness is ¾ in. and over, the ASTM standard 2-in. gage length tension test specimen[1] may be used. Longitudinal tension specimens shall not be flattened between gage marks. The sides of specimens shall be parallel between gage marks.

The transverse tension test may be made on seamless pipe 8 in. and over in nominal diameter. Specimens may be taken from a ring cut from the pipe or from sections resulting from the flattening tests. The specimen shall consist of a strip cut transversely from the pipe; the width of the specimen shall be 1½ in. and its gage length 2 in. When the pipe wall thickness is ¾ in. and over, the ASTM standard 2-in. gage length tension test specimen may be used. Specimens cut from the ring section shall be flattened cold and shall be parallel between gage marks. At the option of the manufacturer, the transverse tension test specimen may be machined off on either surface provided not over 15 per cent of the nominal thickness is removed from either side.

Test specimens for the bend and flattening tests shall consist of sections cut from a pipe. Specimens for flattening tests shall be smooth on the ends and free from burrs, except when made on crop ends.

All routine check tests shall be made at room temperature.

Number of Tests. (*a*) *Welded.*—The longitudinal tension test shall be made on one length of pipe from each lot[2] of 400 lengths or fraction thereof of each size under 6 in., and from each lot of 200 lengths or fraction thereof of each size 6 in. and over.

Each length of welded pipe shall be subjected to the hydrostatic test specified.

The flattening test shall be made on both crop ends cut from each length of welded pipe.

(*b*) *Seamless.*—One of either of the tests specified shall be made on one length of pipe from each lot of 400 lengths or fraction thereof of each size under 6 in., and from each lot of 200 lengths or fraction thereof of each size 6 in. and over.

Each length of seamless pipe shall be subjected to the hydrostatic test specified.

The flattening tests specified shall be made on one length of pipe from each lot of 400 lengths or fraction thereof of each size over 2 in., up to but not including 6 in., and from each lot of 200 lengths or fraction thereof, of each size 6 in. and over.

For Grades A and B seamless pipe, 2 in. and under in nominal diameter, the bend test specified herein shall be made on one pipe from each lot of 400 lengths or fraction thereof of each size.

If any test specimen shows defective machining or develops flaws, it may be discarded and another specimen substituted.

If the percentage of elongation of any tension test specimen is less than that prescribed in the table of tensile requirements and any part of the fracture is more than ¾ in. from the center of the gage length of a 2-in. specimen, or is

[1] See Fig. 3, p. 309.

[2] A lot shall consist of all the pipe bought by a purchaser of the same size and wall thickness from any one melt.

outside the middle third of the gage length of an 8-in. specimen, as indicated by scribe scratches marked on the specimen before testing, a retest shall be allowed. If a specimen breaks in an inside or outside surface flaw, a retest shall be allowed.

Retests.—If the results of any chemical or physical tests of any lot do not conform to the requirements specified herein, retests shall be made on additional pipes of double the original number from the same lot, each of which shall conform to the requirements specified.

Should a crop end of a finished welded or seamless pipe fail in the flattening test, one retest may be made from the failed end. Pipe may be normalized either before or after the first test, but pipe shall be subjected to only two normalizing treatments.

Dimensions.—The nominal wall thicknesses for pipe of various diameters are as prescribed in Table V, page 361, except for the omission of certain sizes which are not commercially available, *viz.*, the 20- and 24-in. sizes in Schedule 160, the 24-in. size in Schedule 140, and all thicknesses of the 30-in. size.

Nipples shall be cut from pipe of the same dimensions and quality described in these specifications.

Permissible Variations in Weight and Dimensions. *Weight.*—The weight of any length of pipe shall not vary more than 6.5 per cent over and 3.5 per cent under that specified for pipe of Schedules 10 to 120, inclusive, nor more than 10 per cent over and 3.5 per cent under that specified for pipe heavier than Schedule 120. Unless otherwise agreed upon between the manufacturer and the purchaser, pipe in sizes smaller than 4 in. may be weighed in convenient lots; pipe in sizes 4 in. and larger shall be weighed separately.

The nominal weights for pipe of various diameters are given in Table VI, page 362.

Diameter.—Variations in outside diameter shall not exceed the following:

Nominal pipe size, in.	Permissible variations in outside diameter, in.	
	Over	Under
1/8 to 1 1/2, incl.	1/64 (0.015)	1/32 (0.031)
2 to 4, incl.	1/32 (0.031)	1/32 (0.031)
5 to 8, incl.	1/16 (0.062)	1/32 (0.031)
10 to 18, incl.	3/32 (0.093)	1/32 (0.031)
20 to 24, incl.	1/8 (0.125)	1/32 (0.031)

Thickness.—The minimum wall thickness at any point for lap-welded and seamless pipe shall not be more than 12.5 per cent under the nominal wall thickness specified.

Lengths.—Pipe lengths shall be in accordance with the following regular practice:

The lengths required shall be specified in the order, and

No jointers are permitted unless otherwise specified.

Ends.—Unless otherwise specified, pipe shall be furnished with plain ends. All burrs at the ends of the pipe shall be removed.

Finish.—The finished pipe shall be reasonably straight and free from injurious defects, and shall have a workmanlike finish. At the discretion of the inspector

representing the purchaser, finished pipe shall be subject to rejection if surface defects classified below are not scattered, but appear over a large area in excess of what is considered "workmanlike finish."

Depth of Injurious Defects.—All defects shall be explored for depth. When the depth of defects of either welded or seamless pipe encroaches on the minimum wall thickness (87.5 per cent of the nominal thickness), or is in excess of 12.5 per cent of the nominal wall thickness, such defects shall be considered injurious.

Machining or Grinding Defects.—Surface defects shall be classified and treated as follows:

U-bottom mechanical marks and abrasions[1] excepting pits, tears, scabs, slivers, etc., shall be acceptable without grinding or machining, if not deeper than 12.5 per cent of the nominal wall thickness nor in any case deeper than 1/16 in. All V-bottom marks and abrasions and U-bottom marks and abrasions deeper than 1/16 in. but not deeper than 12.5 per cent of the nominal wall thickness shall be removed by grinding or machining to sound metal. Where the width of any mark or abrasion is less than the depth it shall be classified as V-bottomed.

Pipe showing inside or outside surface checks (fish scale) 1/64 in. or less in depth need not have these defects removed. Such defects over 1/64 in. but not more than 1/32 in. in depth shall be removed by machining or grinding. Pipe on which these defects are more than 1/32 in. in depth shall be rejected, unless the manufacturer can demonstrate to the purchaser that the defects are not injurious as defined under Depth of Injurious Defects above.

Pipe showing scabs, seams, laps, tears, or slivers not deeper than 5 per cent of the nominal wall thickness need not have these defects removed. If deeper than 5 per cent such defects shall be removed by machining or grinding providing they are not injurious as defined above.

Pits which do not fall under the classification of injurious defects as defined above need not be removed.

When grinding or machining is permitted, the outside diameter at the point of grinding or machining may be reduced by the amount so removed. Should it be impracticable to secure a direct measurement, the wall thickness at the point of grinding, or at a defect not required to be removed, shall be determined by deducting the amount removed in grinding, or the depth of the defect, from the minimum measured wall thickness at the ends of the pipe, and the remainder shall not be less than 87.5 per cent of the nominal wall thickness.

Machining or grinding shall follow the mill's inspection of the pipe as rolled and shall be followed by supplementary visual inspection.

Repair by Welding.—Repair of injurious defects shall be permitted only subject to the approval of the purchaser. Welding of injurious defects in no case shall be permitted when the depth of defect exceeds 33⅓ per cent of the nominal pipe wall thickness or the length of repair of seamless pipe exceeds 25 per cent of the nominal diameter of the pipe. Defects shall be thoroughly chipped out before welding. Each length of repaired pipe shall be retested hydrostatically.

Marking.—Each length of pipe manufactured in accordance with these specifications shall be legibly marked beginning within 12 in. from the end, either by stenciling, stamping, or rolling, with the manufacturer's private identifying mark, together with the designation A 106-A (or B depending on the grade of seamless

[1] Marks and abrasions are defined as cable marks, dinges, guide marks, roll marks, ball scratches, scores, etc.

steel used, or no letter if lap-welded; where acid-bessemer steel is used, the word bessemer shall be added, that is, seamless bessemer A or seamless bessemer B), and an additional *S* if the pipe conforms in any case to the supplementary requirements specified in Paragraphs S1 to S6. The length and schedule number of the pipe also shall be included. Marking shall begin within 12 in. of one end of each length. On pipe sizes 4 in. and larger, the weight shall be given. On small diameter pipe which is bundled, the above information may be legibly stamped on a metal tag securely attached to each bundle. When pipe marked as specified in this section is rejected, the designation A 106 shall be canceled.

Inspection.—The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works that concern the manufacture of the material ordered. The manufacturer shall afford the inspector, without charge, all reasonable facilities to satisfy him that the material is being furnished in accordance with these specifications. All tests and inspection shall be made at the place of manufacture prior to shipment and at the manufacturer's expense, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

Rejection.—Each length of pipe that develops injurious defects during shop working or application operations will be rejected, and the manufacturer shall be notified. No rejections under these or any other specifications shall be marked as specified in the section on Marking for sale under these specifications except where such pipe fails to comply with the weight requirements alone, in which case it may be sold under the weight specifications with which it does comply.

Supplementary Requirements for ASTM A 106 SEAMLESS PIPE FOR USE IN CENTRAL STATIONS At Pressures of 400 Psi, or Over, and Temperatures of 850 F, or Other Applications Where a Superior Grade of Pipe Is Required in Nominal Sizes 8 In. and Larger

S1. These requirements shall not be considered unless specified in the order, in which event the supplementary tests specified in Paragraphs S2 to S6 shall be made at the mill at the purchaser's expense and witnessed by his inspector before shipment of the material. If these tests are required for smaller size seamless pipe, they shall be made the subject of agreement between the manufacturer and the purchaser.

S2. Check Analysis.—Check analysis may be made on any length of pipe. Individual lengths failing to conform to the chemical requirements prescribed shall be rejected.

S3. Transverse Tension Test.—Transverse tension tests may be made on specimens from both ends of each length of pipe. If the specimen from either end of any length fails to conform to the physical properties prescribed, that length shall be rejected.

S4. Flattening Test.—The flattening tests specified in this specification may be made on specimens from both ends of each length of pipe. Crop ends may be used. If the specimen from either end of any length fails to conform to the specified requirements, that length shall be rejected.

S5. Finish and Metal Structure.—The steel shall have a homogeneous structure as shown by the etching test described in Paragraph S6.

S6. Etching Tests.—Etching tests[1] may be made on sections from any pipe, and shall show sound and reasonably uniform material, free from injurious laminations, cracks, and similar objectionable defects. If the specimen from any length shows objectionable defects, that length may be rejected.

ASTM Tentative Specifications for SEAMLESS CARBON-MOLYBDENUM ALLOY-STEEL PIPE FOR SERVICE AT TEMPERATURES FROM 750 TO 1000 F

Serial Designation A206-44T

Abstracted[2]

These specifications cover seamless carbon-molybdenum alloy-steel pipe intended for service at metal temperatures from 750 to 1000 F. (Other varieties of ferritic alloy steels and austenitic alloy steels for service at temperatures from 750 to 1100 F are covered by ASTM Specifications A158 abstracted on page 389. Carbon-steel pipe for more moderate high-temperature service is covered by ASTM Specification A106 abstracted on page 378.) Choice of steels from the respective specifications, or from additional compositions which may be proposed, should be made on the basis of high-temperature properties, weldability, and general stability under the particular service conditions.

Pending the incorporation of high-temperature data in an appendix, the high-temperature properties and tests are left as a matter of agreement between the manufacturer and purchaser. For use in central stations having steam-service pressures of 400 psi or over and temperatures up to 1000 F, or other applications requiring a superior grade of pipe, supplementary tests of an optional nature are provided.

Process.—The steel shall be made by either or both the open-hearth or the electric-furnace process. When requested by the purchaser and so stated in the order, hot-finished pipe in the as-finished condition, in nominal sizes 3 in. and over made to Schedules 140 and 160, shall have a structural grain size[3] which is predominantly 3 to 6 as measured on an ASTM grain size chart.

Heat Treatment.—The hot-rolled pipe after finishing shall be given a stress-relief anneal by heating in a furnace to approximately 1200 F, followed by furnace cooling at a rate not to exceed 50 F per hr until below 1000 F.

Unless otherwise specified, cold-drawn pipe after finishing shall be given a process anneal by heating in a furnace from 1200 to 1300 F, followed by cooling in the furnace at a rate not to exceed 50 deg. F per hr until below 1000 F.

Chemical Composition.—The steel shall conform to the following requirements as to chemical composition:

[1] Pending development of etching methods applicable to the product covered by these specifications, it is recommended that the Recommended Practice for a Macro-etch Test for Steel, described in the *Metals Handbook*, ASM, 1939 ed., p. 730, be followed.

[2] For complete specification, reference may be made to ASTM A206 (see note, p. 369).

[3] The term "structural grain size" refers to the predominant grain as evidenced by ferrite grains, Widmanstätten areas, or pearlite patches.

Identification symbol[a]	P 1
Grade	Carbon-molybdenum
Carbon, per cent	0.10 to 0.20
Manganese, per cent	0.30 to 0.60
Phosphorus max, per cent	0.04
Sulfur, max, per cent	0.05
Silicon, per cent	0.10 to 0.50
Molybdenum, per cent	0.45 to 0.65

[a] The similar alloy in the corresponding specifications for alloy-steel forgings and castings for high-temperature service bears the number F 1 and C 1, respectively.

Tensile Properties.—The material shall conform to the requirements of tensile properties at room temperature tabulated herewith.

Tensile strength, min, psi	55,000	
Yield point, min, psi	30,000	
	Longitudinal	Transverse
Elongation in 2 in., min, per cent:		
Basic minimum elongation for walls, 5/16 in. and over in thickness, strip tests, and for all small sizes tested in full section	30	20
When standard round 2-in. gage length test specimen is used	22	14
For strip tests, a deduction for each 1/32 in. decrease in wall thickness below 5/16 in. from the basic minimum elongation of the following percentage	1.50[a]	1.00[a]

[a] The following table gives the computed minimum values:

Wall thickness, in.	Longitudinal	Transverse
5/16 (0.312)	30.00	20.00
9/32 (0.281)	28.50	19.00
1/4 (0.250)	27.00	18.00
7/32 (0.219)	25.50	
3/16 (0.188)	24.00	
5/32 (0.156)	22.50	
1/8 (0.125)	21.00	
3/32 (0.094)	19.50	
1/16 (0.062)	18.00	

NOTE.—Where a definite yield point is not exhibited, the yield strength, corresponding to a limiting permanent set of 0.2 per cent of the gage length of the specimen, shall be reported.

Bending Properties.—Pipe 2 in. and smaller shall stand being bent cold 90 deg around a cylindrical mandrel 12 times the nominal diameter of the pipe without developing cracks. When ordered for close coiling, the pipe shall stand being bent 180 deg around a mandrel 8 times the nominal pipe diameter without failure.

Flattening Test.—For pipe over 2 in., a section of pipe not less than 2½ in. in length shall be flattened cold between parallel plates until opposite walls of the pipe meet. No cracks or breaks shall appear until the distance between the plates is less than that calculated for the value of H determined by the formula given on page 380 using a value of $e = 0.08$. Evidence of laminations or burned material shall not develop during the entire flattening process.

Hydrostatic Tests.—Unless otherwise agreed upon between the manufacturer and purchaser, each length of pipe shall be tested by the manufacturer to a hydrostatic pressure which shall produce in the pipe wall a stress of 50 per cent of the minimum specified yield point at room temperature. This pressure shall be determined by the formula given in abstract of ASTM A106 on page 380.

Photomicrographs.—When requested by the purchaser and so stated in the order, the manufacturer shall furnish one photomicrograph at 100 diameters from a specimen of pipe in the as-finished condition for each individual size and wall thickness from each melt, for pipe in nominal sizes 3 in. and over made to Schedules 140 and 160, to demonstrate that such pipe conforms to the requirements given under Process, when the order states that those requirements are to be met. Such photomicrographs shall be suitably identified as to pipe size, wall thickness, and melt. No photomicrographs for the individual pieces purchased shall be required except as specified in Paragraph S7 of supplementary requirements.

Test Specimens.—Specimens cut either longitudinally or transversely shall be acceptable for the tension test.

The longitudinal tension test may be made in full section of the pipe up to the capacity of the testing machine. For larger sizes, the tension test specimens shall consist of strips cut from the pipe. The width of these specimens shall be 1½ in. and shall have a gage length of 2 in. When the pipe wall thickness is ¾ in. and over, the ASTM standard 2-in. gage length tension test specimen[1] may be used. Longitudinal tension specimens shall not be flattened between gage marks. The sides of specimens shall be parallel between gage marks.

The transverse tension test may be made on pipe 8 in. and over in nominal diameter. Specimens may be taken from a ring cut from the pipe or from sections resulting from the flattening tests. The specimen shall consist of a strip cut transversely from the pipe; the width of the specimen shall be 1½ in. and its gage length 2 in. When the pipe wall thickness is ¾ in. and over, the ASTM standard 2-in. gage length tension test specimen[1] may be used. Specimens shall be flattened cold and heat-treated in the same manner as the pipe.

Number of Tests.—Tests shall be made as follows on one pipe from each heat-treated lot, but in no case on less than 5 per cent of the pipe ordered. Results of these tests shall be reported to the purchaser or his representative:

One transverse or longitudinal tension test.

One flattening test for pipe over 2 in. in nominal diameter.

One bend test for pipe 2 in. or under in nominal diameter.

For material heat-treated by the continuous process, the tests shall be made on each pipe in a lot constituting 5 per cent of the pipe ordered, but on not less than two pipes.

Each length of pipe shall be subjected to the hydrostatic test specified.

Dimensions.—The nominal wall thicknesses for pipe of various diameters are as prescribed in Table V, page 361, except for the omission of certain sizes which

[1] See Fig. 3, p. 309.

are not commercially available, *viz.*, the 20 and 24-in. sizes in Schedule 160, the 24-in. size in Schedule 140, and all thicknesses of the 30-in. size. The size and wall thicknesses shown in Appendix I of A206 were agreed on as standard for stock purposes and included as information only.

Permissible Variations in Weight and Dimensions.—These variations are identical with those of ASTM A106 abstracted on page 382, except that ranges in which weight tolerances apply are based on pipe size rather than on schedule thickness.

Lengths.—No pipe shall be under the specified length and not more than ¼ in. over that specified. No jointers are permitted, unless otherwise specified.

Ends.—Pipe shall be furnished with plain ends unless otherwise specified.

Finish.—The finished pipe shall be free from injurious defects. See abstract of ASTM A106, page 383, for definition of injurious defects.

Marking.—Each length of pipe manufactured in accordance with these specifications shall be legibly marked, either by stenciling, stamping, or rolling, with the manufacturer's private identifying mark, with the designation A206, and an additional symbol *S* if the pipe conforms to the supplementary requirements specified in Paragraphs S1 to S7, the length, the ASA schedule number, and the heat number or manufacturer's number by which the heat can be identified. For further details see abstract of ASTM A106, page 383.

Inspection.—In addition to provisions given in ASTM A106 (see page 384), when specified in the order, the manufacturer shall notify the purchaser in time so that he may have his inspector present to witness any part of the manufacture or tests that may be desired.

Certification.—When agreed upon in writing between the manufacturer and the purchaser, a certification that the material conforms to the requirements of these specifications shall be the basis of acceptance of the material. Otherwise the manufacturer shall report the results of the chemical analyses and physical tests made in accordance with these specifications.

Supplementary Requirements for ASTM A206
SEAMLESS PIPE FOR USE IN CENTRAL STATIONS
At Pressures of 400 Psi, or Over, and Temperatures of 750 to 1000 F or Other Applications Where a Superior Grade of Pipe Is Required

These requirements shall not be considered unless specified in the order, in which event the supplementary tests shall be made at the mill, unless otherwise agreed upon, at the purchaser's expense and witnessed by his inspector before shipment of the material. For description of supplementary tests, S1 to S6 inclusive, refer to ASTM Specification A106, abstracted on page 384. A further requirement S7 of A206 provides for photomicrographs, where specified, on each piece of pipe.

Appendix I of A206

The following sizes and wall thicknesses of carbon-molybdenum pipe as provided for in these specifications were agreed on as standard for stock purposes by the pipe manufacturers and the Prime Movers Committee of the Edison Electric Institute. These data are included in A206 as information only.

Appendix I of A206

Nominal pipe size, in.	Schedule No.	Thickness, in.	Schedule No.	Thickness, in.
½	160	0.187	160	0.187
¾	160	0.218	160	0.218
1	160	0.250	160	0.250
1¼	160	0.250	160	0.250
2	160	0.343	160	0.343
2½	160	0.375	160	0.375
3	160	0.437	160	0.437
4	160	0.531	120	0.437
6	160	0.718	120	0.562
8	160	0.906	100	0.593
10	160	1.125	100	0.718
12	160	1.312	100	0.843
14	160	1.406	100	0.937
16	160	1.562	100	1.031

ASTM Tentative Specifications for SEAMLESS ALLOY-STEEL PIPE FOR SERVICE AT TEMPERATURES FROM 750 TO 1100 F

Serial Designation A158-44T

Abstracted[1]

These specifications cover several varieties of ferritic and austenitic seamless pipe which have been used rather extensively for service at metal temperatures from 750 to 1100 F. (Carbon-molybdenum alloy-steel pipe for service at temperatures from 750 to 1000 F is covered by ASTM Specification A206 abstracted on page 385. Carbon-steel pipe for more moderate high-temperature service is covered by ASTM Specification A106, abstracted on page 378.) Choice from the respective steels, or from additional compositions which may be proposed, should be made on the basis of high-temperature properties, weldability, and general stability at the service temperature.

Pending the incorporation of high-temperature data in an appendix, the high-temperature properties and tests are left as a matter of agreement between the manufacturer and purchaser. For use in central stations having steam-service pressures of 400 psi or over, and for other applications where a superior grade of pipe is required, additional tests of an optional nature are provided.

Process.—The steel shall be made by either or both of the following processes: open-hearth or electric-furnace. Unless otherwise specified, pipe 2 in. and over in nominal diameter shall be furnished hot-finished followed by the treatment specified herein. Unless otherwise specified, pipe under 2 in. in nominal diameter may be furnished either hot-finished or cold-drawn, with, where necessary, a suitable manufacturer's finishing treatment.

Heat Treatment.—All ferritic steels covered under Process other than Grade P 5c shall be reheated and furnished in a full-annealed or normalized and drawn condition as stated in the order. If furnished in the normalized and drawn con-

[1] For complete specification, reference may be made to ASTM A158 (see note, p. 369). Abstract includes Emergency Alternate Provisions, issued June 1, 1943.

dition, the temperature for drawing shall be at least 200 F above the service temperature as given in the order (Note 1).

Grade P5c steel, unless otherwise specified, shall be treated by heating to approximately 1350 F for a proper time, followed by air or furnace cooling.

All austenitic steels shall be heat-treated (Note 2) according to present accepted practice.

Note 1.—Certain of the ferritic steels covered by these specifications will harden if cooled rapidly from above their critical temperature. Some will air-harden, that is, become hardened to an undesirable degree when cooled in air from high temperatures, particularly the 4 to 6 per cent chromium steels, P5. Therefore, operations involving heating such steels above their critical temperatures, such as welding, flanging, and hot bending, should be followed by suitable heat treatment.

Note 2.—Fabricating operations involving upsetting, flanging, bending, and forming often make retreating desirable to insure adequate corrosion resistance, depending upon service conditions.

Tensile Properties.—The material shall conform to the following minimum requirements as to tensile properties at room temperature:

	Ferritic		Austenitic	
Tensile strength, min, psi	60,000		75,000	
Yield point, min, psi	30,000		30,000	
	Longitudinal	Transverse	Longitudinal	Transverse
Elongation in 2 in., min, per cent:				
Basic minimum elongation for walls 5/16 in. and over in thickness, strip tests, and for all small sizes tested in full section	30	20	50	40
When standard round 2-in. gage length test specimen is used	22	14	35	25
For strip tests, a deduction for each 1/32-in. decrease in wall thickness below 5/16 in. from the basic minimum elongation of the following percentage	1.50[a]	1.00[a]	2.50[a]	2.00[a]

[a] The table of calculated minimum values given in ASTM A158 is not reproduced here.

Bending Properties.—For pipe 2 in. and under in nominal diameter, a sufficient length of pipe shall stand being bent cold through 90 deg around a cylindrical mandrel, the diameter of which is twelve times the nominal diameter of the pipe, without developing cracks. When ordered for close coiling, the pipe shall stand being bent cold through 180 deg around a cylindrical mandrel, the diameter of which is eight times the nominal diameter of the pipe, without failure.

Flattening Test.—For pipe over 2 in. in nominal diameter, a section of pipe not less than 2½ in. in length shall be flattened cold between parallel plates until the opposite walls of the pipe meet. No cracks or breaks in the metal shall occur until the distance between the plates is less than that calculated for the value of H determined by the formula given on page 380, using a value of $e = 0.08$ for ferritic steels and 0.09 for austenitic steels. Evidence of laminations or burnt material shall not develop during the entire flattening process.

Hydrostatic Tests.—Unless otherwise agreed upon between the manufacturer and the purchaser, each length of pipe shall be tested by the manufacturer to a hydrostatic pressure which will produce in the pipe wall a stress of 50 per cent of the minimum specified yield point at room temperature. This pressure shall be determined by the formula given in the abstract of ASTM A106, page 380.

Test Specimens.—Specimens cut either longitudinally or transversely shall be acceptable for tension tests. Longitudinal tension tests may be made in full section of the pipe up to the capacity of the testing machine. For larger sizes, the tension test specimens shall consist of strips cut from the pipe. The width of these specimens shall be 1½ in. and shall have a gage length of 2 in. When the pipe wall thickness exceeds ¾ in., the ASTM standard 0.505-in. round specimen may be used (Note below). Longitudinal-tension specimens shall not be flattened between gage marks. The sides of specimens shall be parallel between gage marks.

The transverse tension test may be made on pipe 8 in. and over in nominal diameter. Specimens may be taken from a ring cut from the pipe or from sections resulting from the flattening tests. The specimen shall consist of a strip cut transversely from the pipe; the width of the specimen shall be 1½ in., and its gage length 2 in. When the pipe wall thickness exceeds ¾ in., the ASTM standard 0.505-in. round specimen may be used (see Note below). Specimens shall be flattened cold and heat-treated in the same manner as the pipe, and shall be parallel between gage marks. At the option of the manufacturer, the transverse tension test specimen may be machined off on either surface provided not over 15 per cent of the nominal thickness is removed from either side.

Note.—The standard tension test specimens are described in the Standard Methods of Tension Testing of Metallic Materials (ASTM Designation: E8) of the American Society for Testing Materials (see Fig. 3, page 309).

Number of Tests.—Tests shall be made as follows on one pipe from each heat-treated lot, but in no case on less than 5 per cent of the pipe ordered. Results of these tests shall be reported to the purchaser or his representative:

One transverse or longitudinal tension test.

One flattening test for pipe over 2 in. in nominal diameter.

One bend test for pipe 2 in. or under in nominal diameter.

For material heat treated by the continuous process, tests shall be made on each pipe in a lot constituting 5 per cent of the pipe ordered, but on not less than two pipes.

Each length of pipe shall be subjected to the hydrostatic test specified.

Dimensions.—The nominal wall thicknesses for pipe of various diameters are as prescribed in Table V, page 361, except for the omission of certain sizes which are not commercially available, *viz.*, the 20- and 24-in. sizes in Schedule 160. the 24-in. size in Schedule 140, and all thicknesses of the 30-in. size.

Permissible Variations in Weight and Dimensions.—The weight of any length of pipe in nominal sizes of 12 in. and under shall not vary more than 6.5 per cent over and 3.5 per cent under that specified. For nominal sizes over 2 in. the weight of any length shall not vary more than 10 per cent over and 5 per cent under that specified. Variations in diameter and wall thickness are identical with ASTM A106, abstracted on page 382.

Lengths.—No pipe shall be under the specified length and not more than ¼ in. over. No jointers are permitted unless otherwise specified.

Chemical Composition.*—Each alloy shall conform to the following chemical requirements, or to other chemical requirements as specified in the purchase order:

CHEMICAL COMPOSITION ASTM A158-44T PIPE

	Ferritic steels					
Identification symbol	P3a[1]	P3b[1]	P5a	P5b[1]	P5c	P6[1]
Grade	Chromium molybdenum	Chromium molybdenum	4 to 6 per cent chromium	4 to 6 per cent chromium	4 to 6 per cent chromium[2]	13 per cent chromium
Carbon, per cent	0.15 max	0.15 max	0.15 max	0.15 max	0.12 max	0.12 max
Manganese, per cent	0.40 to 0.60	0.30 to 0.60	0.50 max	0.50 max	0.50 max	0.50 max
Phosphorus, max, per cent	0.04	0.03	0.03	0.03	0.03	0.03
Sulphur, max, per cent	0.05	0.03	0.03	0.03	0.03	0.03
Silicon, per cent	0.45 to 0.75	0.50 max	0.50 max	1.00 to 2.00	0.50 max	0.50 max
Chromium, per cent	1.50 to 2.00	1.75 to 2.25	4.00 to 6.00	4.00 to 6.00	4.00 to 6.00	12.00 to 15.00
Molybdenum, per cent	0.60 to 0.80	0.45 to 0.65	0.45 to 0.65[3]	0.45 to 0.65	0.45 to 0.65	
Tungsten, per cent	....	....	0.75 to 1.25[3]	....	....	2.50 to 3.50
Titanium or columbium, per cent	....	....	....	....	See footnote [2]	

[1] Compositions not available for duration of war.

[2] Stabilized with titanium or columbium. Where titanium is used, the titanium content shall be not less than 4 times the carbon content and not more than 0.70 per cent. Where columbium is used, the columbium content shall be not less than 8 to 10 times the carbon content.

[3] Either molybdenum or tungsten shall be used.

* When material is ordered for welding fabrication, chemical and physical requirements shall be a matter of agreement between the purchaser and the manufacturer.

Chemical Composition ASTM A158-44T Pipe.—(*Concluded*)

	Ferritic steels				Austenitic steels		
Identification symbol	P11	P15[1]	EP16[2]	EP17[2]	P8a[2]	P8b	P8d
Grade	Chromium molybdenum	Silicon molybdenum	7 per cent chromium	9 per cent chromium	18 chromium 8 nickel	18 chromium 10 nickel stabilized with titanium	18 chromium 10 nickel stabilized with columbium
Carbon, max, per cent	0.15	0.15	0.15	0.15	0.08	0.08	0.08
Manganese, per cent	0.30 to 0.60	0.30 to 0.60	0.30 to 0.60	0.30 to 0.60	2.00 max	2.00 max	2.00 max
Phosphorus, max, per cent	0.04	0.04	0.04	0.04	0.03	0.03	0.03
Sulphur, max, per cent	0.05	0.045	0.04	0.04	0.03	0.03	0.03
Silicon, per cent	0.50 to 1.00	0.15 to 1.65	0.50 to 1.00	0.50 to 1.00	0.75 max	0.75 max	0.75 max
Nickel, per cent	....	....	....	....	8.00 to 11.00	9.00 to 13.00	9.00 to 13.00
Chromium, per cent	1.00 to 1.50	....	6.00 to 7.50	8.00 to 9.50	18.00 to 20.00	17.00 to 20.00	17.00 to 20.00
Molybdenum, per cent	0.45 to 0.65	0.45 to 0.65	0.40 to 0.55	0.80 to 0.95			
Titanium	....	....	....	....	....	5 times carbon, max 0.60	
Columbium	....	....	....	....	....	....	10 times carbon, max 1.00

[1] Compositions not available for duration of war.

[2] Additional grades, Emergency Alternate Provisions, issued June 1, 1943.

Finish.—The finished pipe shall be free from injurious defects. See abstract of ASTM A106, page 383, for definition of injurious defects.

Marking.—Each length of pipe manufactured in accordance with these specifications shall be legibly marked, either by stenciling, stamping, or rolling, with the manufacturer's private identifying mark, with the designation A158, the length, the type composition, and an additional symbol *S* if the pipe conforms to the supplementary requirements specified in Paragraphs S1 to S6. For further details see abstract of ASTM A 106, page 383.

Inspection.—In addition to provisions given in ASTM A106, see page 384 when specified in the order, the manufacturer shall notify the purchaser in time so that he may have his inspector present to witness any part of the manufacture or tests that may be desired.

Certification.—When agreed upon in writing between the manufacturer and the purchaser, a certification that the material conforms to the requirements of these specifications shall be the basis of acceptance of the material in lieu of test results.

Supplementary Requirements for ASTM A158 SEAMLESS PIPE FOR USE IN CENTRAL STATIONS At Pressures of 400 Psi or Over and Temperatures of 750 to 1100 F, or Other Applications Where a Superior Grade of Pipe Is Required

These requirements shall not be considered unless specified in the order, in which event the supplementary tests shall be made at the mill, unless otherwise agreed upon, at the purchaser's expense and witnessed by his inspector before shipment of the material. (For supplementary tests, refer to ASTM Specification A106, abstracted on page 384.)

ASTM Standard Specifications for ELECTRIC-RESISTANCE-WELDED STEEL PIPE

Serial Designation A135-44 (ASA B36.5)

Abstracted[1]

These specifications cover two grades of electric-resistance-welded steel pipe in sizes up to and including 30 in. diameter. Grade A is adapted for flanging and bending. The plate from which the pipe is made shall be either open-hearth or electric-furnace steel having the following chemical composition and physical properties:

Chemical Composition

Phosphorus, max, per cent	0.045
Sulphur, max, per cent	0.060

[1] For complete specification, reference may be made to ASTM A135 (see note, p. 369).

TENSILE REQUIREMENTS
(Table I of ASTM A135)

	Grade A	Grade B
Tensile strength, min, psi	48,000	60,000
Yield point, min, psi	30,000	35,000
Elongation in 2 in., min, per cent		
Basic minimum elongation for walls 5/16 in. and over in thickness, strip tests, and for all small sizes tested in full section	35	30
For strip tests, a deduction for each 1/32 in. decrease in wall thickness below 5/16 in. from the basic minimum elongation of the following percentage	1.75[a]	1.50[a]

[a] For table of values see ASTM A53, page 374.

The tensile strength across the weld shall be not less than the minimum tensile strength of the grade of pipe ordered. This test is not required for pipe under 6 in. outside diameter.

Flattening Test.—Both crop ends from each length of pipe shall be flattened between parallel plates with the weld located at the point of maximum bending until opposite walls of the pipe meet. No opening in the weld shall take place until the distance between the plates is less than two-thirds the original outside diameter of the pipe. No cracks or breaks shall occur in the metal elsewhere than in the weld until the distance between the plates is less than one-third the original outside diameter of the pipe, but in no case less than five times the thickness of the pipe wall. Evidence of laminations or burned material shall not develop during the entire flattening operation, and the weld shall not show injurious defects.

Hydrostatic Test. Each length of pipe shall be subjected to a hydrostatic test, which will produce a stress in the pipe wall of Grade A pipe of 16,000 to 18,000 psi and in Grade B pipe of 20,000 to 22,000 psi, but in no case greater than 80 per cent of the yield point. The test pressure in no case shall exceed 2,500 psi (see formula given in abstract of ASTM A106, page 380).

The test pressure shall be maintained for not less than 5 sec during which the pipe shall be jarred near both ends with a 2-lb hammer or its equivalent.

Number of Tests.—One longitudinal tension test shall be made on one length from each lot of 400 lengths or fraction thereof of each size under 6 in.

One longitudinal, and if required by purchaser one transverse tension, test across the weld shall be made on one length from each lot of 200 lengths or fraction thereof of each size 6 to 20 in. and from each lot of 100 lengths or fraction thereof of each size over 20 in. and up to and including 30 in.

Dimensions.—The nominal wall thicknesses for pipe of various diameters are as prescribed in Table V, page 361, except for the omission of certain sizes which are not commercially available, *viz.*, the 20- and 24-in. sizes in Schedule 160. the 24-in. size in Schedule 140, and all thicknesses of the 30-in. size.

Permissible Variations in Weight and Dimensions.—The weight of any length of pipe shall not vary more than 3.5 per cent under or 10 per cent over that specified, but the carload weight shall be not more than 1.75 per cent under the nominal weight. The outside diameter shall not vary more than plus or minus 1 per cent from the nominal size specified. The minimum wall thickness at any

point shall be not more than 12.5 per cent under the nominal wall thickness specified.

Lengths.—Unless otherwise specified, pipe shall be furnished in lengths averaging 38 ft or over, with a minimum length of 20 ft, but not more than 5 per cent may be under 32 ft. Jointers made by welding are permissible. When threaded pipe is ordered, jointers shall be made by threading and shall not exceed 5 per cent of the order.

Ends, Plain End Pipe.—Unless otherwise specified, plain end pipe for use with the Dresser or Dayton type coupling shall be reamed both outside and inside sufficiently to remove all burrs. Plain end pipe for welding shall be beveled on the outside to an angle of 35 deg with a width of flat at the end of the pipe of $1/16 \pm 1/32$ in.

Pipe shall be sufficiently free from indentations, projections, or roll marks for a distance of 8 in. from the end of the pipe to make a tight joint with the rubber gasket type of coupling. All plain end pipe intended for Dresser or Dayton type joints or for welding, sizes $10\frac{3}{4}$ in. and smaller in outside diameter, shall be not more than $1/32$ in. smaller than the nominal outside diameter for a distance of 8 in. from the end of the pipe and shall permit the passing for a distance of 8 in. of a ring gage which has a bore $1/16$ in. larger than the nominal outside diameter of the pipe. For sizes larger than $10\frac{3}{4}$ in. in outside diameter, the bore of the ring shall be $3/32$ in. larger than the nominal diameter of the pipe.

Ends, Threaded Pipe.—All threads shall be in accordance with American Standard Pipe Threads, ASA B2.1 (see abstract, page 481). Each length of threaded pipe shall be provided with a coupling.

Finish.—The finished pipe shall be reasonably straight and free from injurious defects. Defects in excess of 12.5 per cent of the nominal wall thickness shall be considered injurious.

Repair by Welding.—Injurious defects in the pipe wall, provided their depth does not exceed one-third the specified wall thickness, shall be repaired by electric welding. Defects in the welds such as sweats or leaks, unless otherwise specified shall be repaired or the piece rejected at the option of the manufacturer. Repairs of this nature shall be made by completely removing the defect, cleaning the cavity, and then electric welding.

All repaired pipe shall be retested hydrostatically, as specified in the section on hydrostatic tests.

Marking.—Each length of pipe shall be legibly marked with appropriate symbols by stenciling, stamping, or rolling to show by whom manufactured, the grade of pipe, and that it conforms to these specifications.

Protective Coating.—After the pipe has been subjected to the hydrostatic test, and if required by the purchaser, it shall be thoroughly cleaned of all dirt, oil, grease, loose scale, and rust; then dried, and given a protective coating of the kind and in the manner specified by the purchaser.

Water Works Service.[1]—Pipe thicknesses should be selected, not only with respect to the stresses to which the pipe will be subjected, but also with respect to the degree of corrosiveness of the soil and of the water to be carried, with due regard to the probable required life of the pipe and the cost of replacement. In cases where the water or ground, or both, have been known to be corrosive, due consideration should be given to the protection of steel water pipe, particularly that with thin walls. It is recommended that if coatings are used they conform

[1] Emergency Alternate Provision, issued Aug. 24, 1942.

to the Standard Specifications for Coal-tar Enamel Protective Coatings for Steel Water Pipe of Sizes 30 in. and Over (AWWA 7A.5) and for Coal-tar Enamel Protective Coatings for Steel Water Pipe of Sizes of 4½ in. Outside Diameter up to but not Including 30 in. (AWWA 7A.6) of the American Water Works Association.

Inspection.—For inspection provisions, see abstract of ASTM A106, page 384.

Rejection.—Each length of pipe which develops injurious defects during shop working or application operations will be rejected, and the manufacturer shall be notified.

ASTM Standard Specifications for ELECTRIC-FUSION-WELDED STEEL PIPE FOR HIGH-TEMPERATURE AND HIGH-PRESSURE SERVICE

Serial Designation A155-42 (ASA B36.11)

Abstracted[1]

These specifications cover electric-fusion-welded steel pipe 18 in. outside diameter and over, fabricated from plate and suitable for high-temperature and high-pressure service. In particular this contemplates pressures above 250 psi and temperatures from 450 to 850 F. Higher temperatures with appropriate reduction in working stresses may be used at the discretion of the designing engineer. Pipe ordered under these specifications shall be suitable for bending, flanging, corrugating, and similar forming operations. For use in central stations or other applications where a superior grade of pipe is required, additional tests of an optional nature are provided.

The welding, radiographic examinations, and stress relieving shall be in accordance with the requirements for Par. U68 of the ASME Rules for Construction of Unfired Pressure Vessels.[2] Stress relieving shall precede any radiographic examination of the weld.

The steel shall be a killed steel, made by either the open-hearth or electric-furnace process, and shall conform to the following requirements as to chemical composition and physical properties.

CHEMICAL COMPOSITION

	Grade A	Grade B	Grade C
Carbon, max, per cent, plate ¾ in. or less	0.15	0.20	0.25
Plate over ¾ in.	0.17	0.22	0.30
Manganese, per cent	0.35 to 0.60	0.35 to 0.60	0.30 to 0.60
Phosphorus, max, per cent, acid	0.05	0.05	0.05
Basic	0.04	0.04	0.04
Sulphur, max, per cent	0.05	0.05	0.05
Silicon, min, per cent	0.10	0.10	0.10

[1] For complete specification, reference may be made to ASTM A155 (see note, p. 369).

[2] Boiler Construction Code, ASME, Sec. VIII, latest edition.

Tensile Properties

	Grade A	Grade B	Grade C
Tensile strength, min, psi........	45,000	50,000	55,000
Yield point, min, psi.............	0.5 ten str	0.5 ten str	0.5 ten str
But in no case less than........	24,000	27,000	
Elongation in 8 in., min, per cent	$\frac{1,500,000}{\text{ten str}}$	$\frac{1,500,000}{\text{ten str}}$	$\frac{1,500,000}{\text{ten str}}$

Note.—A reduction of 0.125 per cent shall be made for each increase of 1/32 in. of the specified thickness of over 3/4 in. to a minimum elongation of 20 per cent. For transverse-tension tests, an elongation of 5 per cent less than specified will be acceptable.

Bending Properties.—One transverse bend test shall be made on a specimen cut from the plate material of the finished pipe from each lot of 500 ft of pipe or fraction thereof. This bend specimen shall stand being bent cold in the opposite direction to its initial curvature through 180 deg without cracking on the outside of the bent portion. For material 1 in. or under in nominal thickness the bend specimen shall be bent around a pin the diameter of which is equal to the thickness of the specimen. For material over 1 in. in nominal thickness, the diameter of the pin shall be equal to twice the thickness of the specimen.

Hydrostatic Tests.—Each length of finished pipe shall be subjected to a hydrostatic test designed to stress the pipe to 75 per cent of the minimum specified yield point. While under pressure, the pipe shall be struck at 6-in. intervals on both sides of the weld with a hammer.

Tension Tests.—One longitudinal, or if desired one transverse, tension test shall be made on one pipe length from each lot of 500 ft of pipe or fraction thereof.

Weld Test.—One welded test plate shall be prepared and tested for each 200 ft of weld or fraction thereof in accordance with the requirements governing welding in Par. U68 of the ASME Rules for Construction of Unfired Pressure Vessels.

Thicknesses and Weights.—Variations in wall thickness and weight of pipe are governed by the specifications to which the plate is ordered, except that permissible overweights of plates of different thicknesses and widths are given. The permissible overweights for plate 48 in. or under in width range from 6 per cent for plate 1/4 in. thick to 2.5 per cent for plate over 3/4 in. thick. The permissible overweight also varies with width of plate. For example, plate 1 in. thick and under 48 in. in width may vary 2.5 per cent over the nominal weight. while the same thickness of plate 168 in. or over in width may vary 9 per cent, The weights of individual plates ordered to thickness shall not exceed the nominal weight by more than 1 1/3 times the amount given.

Permissible Variations in Dimensions.—Permissible variations in dimensions at any point in a length of pipe shall not exceed: (1) for outside diameter, based on circumferential measurement, ± 0.5 per cent of the nominal outside diameter; (2) for out-of-roundness, that is, difference between major and minor outside diameters, 1 per cent; (3) for alignment, using a 10-ft straightedge placed so that both ends are in contact with the pipe, 1/8 in.

Thickness.—The minimum wall thickness at any point in the pipe wall shall not be more than 0.01 in. under the nominal thickness.

Lengths.—The lengths required shall be specified in the order; circumferentially welded joints of the same quality as the longitudinal joints shall be allowed by mutual agreement between the manufacturer and the purchaser.

Finish.—Provision is made for machining or grinding out defects, provided the wall thickness is not reduced below the specified minimum. Repair of injurious defects is subject to approval of the purchaser. Repair welds shall be radiographed and the pipe stress relieved again unless the welds are small and the second stress relief is not considered necessary by the purchaser's inspector. Hydrostatic tests shall be made on the finished pipe.

Marking.—Each length of pipe shall be marked within 12 in. of one end with the manufacturer's private identifying mark, the symbols A155A (B or C), and an additional *S* if pipe conforms to supplementary requirements. For further details, see abstract of ASTM A106, page 383.

Inspection.—For inspection provisions, see abstract of ASTM A106, page 384.

Rejection.—Where pipe is rejected because of failure to comply with weight requirements alone, it may be sold under the weight specifications with which it does comply.

Supplementary Requirements.—Check analysis of individual lengths, tension, and bend tests of the plate and of the welded joint from both ends of each length, chipping of the weld flush with the inside of the pipe, and etching tests to show freedom from injurious laminations, cracks, etc., shall be made if specified in the order. Such tests shall be made at the mill at the purchaser's expense and witnessed by his inspector.

ASTM Standard Specifications for ELECTRIC-FUSION-WELDED STEEL PIPE (SIZES 30 IN. AND OVER)

Serial Designation A134-42 (ASA B36.4)

Abstracted[1]

These specifications cover steel pipe 30 in. in diameter and over, with a wall thickness up to ¾ in., inclusive, manufactured by the electric-fusion-welding process. (This pipe is suitable for ordinary uses, such as waterworks service, steam lines below 250 psi, and air and gas lines at temperatures below 450 F.[2])

When pipe made in accordance with these specifications is to be used for water works service, the steel plate to be selected by the manufacturer shall conform to one of the following specifications of the ASTM, except that all references to manganese limitations shall be omitted: Standard Specifications for Steel for Bridges and Buildings (ASTM Designation: A7), or for Mild Steel Plates (ASTM Designation: A10), or for Low Tensile Strength Carbon-steel Plates of Structural Quality for Welding (ASTM Designation: A78), Grade B.

Tentative Specifications for Light Gage Structural Quality Flat Hot-rolled Carbon Steel (0.2499 and 0.1874 in. to 0.0478 in. in thickness) (ASTM Designation: A245) Grades A, B, or C.

[1] For complete specification, reference may be made to ASTM A134 (see note, p. 369). Abstract includes Emergency Alternate Provisions, issued Aug. 18, 1942.

[2] Requirements of ASA Code for Pressure Piping, B31.1.

The longitudinal welds shall be made by automatic means and shall be of reasonably uniform width and height for the entire length of pipe.

The tensile strength across the weld shall be not less than the minimum tensile strength specified for the plate.

Hydrostatic Test.—Unless otherwise specified, each length of pipe shall be tested at the mill to a hydrostatic pressure equal to 150 per cent of the working pressure but in no case shall the test stress exceed 36 per cent of the minimum tensile strength specified. The working pressure shall be calculated by Barlow's formula.[1] While under test pressure, the pipe shall be jarred with a 2-lb hammer or its equivalent.

Tension Test.—If required by the purchaser, a transverse tension test specimen shall be taken across the weld from the end of the pipe or at any point along the pipe from one length of each lot of 100 lengths or fraction thereof. The tension test specimen shall be flattened and the sides machined.

Bend Test.—If required by the purchaser, a transverse bend specimen shall be cut from the same length of pipe selected for the tension test. The bend-test specimen shall have the protruding portions of the weld removed from both the inside and outside of the pipe. The specimen shall stand being bent cold through 180 deg around a pin the diameter of which is equal to 4½ times the thickness of the plate, without developing cracks. In making the bend test, the side of the specimen representing the inside of the pipe shall be placed next the pin.

Diameter Variations.—The outside diameter of the pipe shall be within tolerances consistent with the requirements of field joints as arranged by agreement with the purchaser.

Lengths.—Standard lengths of pipe shall be approximately 30 to 40 ft. Not more than 2 per cent of the total number of lengths may be shorter than 30 ft., except in the case of specials, such as reducers, etc. On pipe of larger diameter one or more circumferential-joints may be made by welding or riveting as specified.

ASTM Standard Specifications for ELECTRIC-FUSION-WELDED STEEL PIPE (SIZES 4 IN. TO, BUT NOT INCLUDING, 30 IN.)

Serial Designation A139-42 (ASA B36.9)

Abstracted[2]

These specifications cover two grades of electric-fusion-welded steel pipe in sizes 4 in. to but not including 30 in. in diameter, and wall thicknesses up to ⅝ in., inclusive. This pipe is intended for conveying liquid, gas, or vapor; only Grade A is adapted for flanging and bending. Unless otherwise specified, the steel shall be made by either or both the open-hearth or electric-furnace

[1] See p. 41.

[2] For complete specification, reference may be made to ASTM A139 (see note, p. 369). Abstract includes Emergency Alternate Provisions, issued Aug. 18, 1942.

processes. The steel shall conform to the following requirements as to chemical composition and physical properties.

Chemical Composition

Phosphorus, max, per cent	0.045
Sulphur, max, per cent	0.060

Minimum Physical Properties

	Grade A	Grade B
Tensile strength, psi	48,000	60,000
Yield point, psi	30,000	35,000

Wall thickness, in.	Elongation in 2 in., min, per cent	
	Grade A	Grade B
5/16(0.3125) and over	35.00	30.00
9/32(0.281)	33.25	28.50
1/4(0.250)	31.50	27.00
7/32(0.2187)	29.75	25.50
3/16(0.1875)	28.00	24.00
5/32(0.156)	26.25	22.50
1/8(0.125)	24.50	21.00
3/32(0.0937)	22.75	19.50
1/16(0.0625)	21.00	18.00

The tensile strength across the weld shall be not less than the minimum tensile strength of the grade of steel ordered.

Hydrostatic Tests.—Each length of pipe shall be subjected to a hydrostatic test designed to give a stress of approximately 36 per cent of the specified tensile strength but not to exceed 80 per cent of the specified yield point. The hydrostatic-test pressure shall be maintained for not less than 5 sec and, while under pressure, shall be jarred near both ends with a 2-lb hammer or its equivalent.

Tension Tests.—One longitudinal test of the pipe or plate and one transverse tension test across the weld shall be made on one length from each lot of pipe or fraction thereof of a given size.

Bend Test.—One bend test specimen taken across the weld shall, if required by purchaser, stand being bent cold through 180 deg around a pin the diameter of which is equal to 4½ times the thickness of the plate without developing cracks.

Diameter Variations.—The outside diameter shall not vary more than 1 per cent over or under the nominal size specified.

Lengths.—Unless otherwise specified, pipe shall be furnished in lengths averaging 38 ft or over, with a minimum length of 20 ft, but not more than 5 per cent may be under 32 ft. Jointers made by welding are permissible. Where threaded pipe is ordered, jointers shall be made by threading and shall not exceed 5 per cent of the order.

AWWA Tentative Standard Specifications for LOCK-BAR STEEL PIPE

Serial Designation 7A.2

Abstracted[1]

These specifications cover pipe made from steel plates rolled or formed into a circle, having the longitudinal edges planed and upset to a dovetail form which will engage in the groove of an H-shaped steel lock bar to form the longitudinal joint of the pipe. After assembly of the plates and lock bars, the latter are closed by cold-pressing, thus making a tight fit over the dovetail edges of the plates.

Materials.—Steel for plates, rivets, and lock bars shall be made by the basic open-hearth process. The steel plate shall conform to requirements of ASTM Specification A30.

The lock bars shall be extra-dead-soft steel of 40,000 to 50,000 psi tensile strength and capable of being bent cold through 180 deg and flattened without showing sign of fracture on outside of bent portion.

Rivets for assembling sections of pipe shall be of the buttonhead or other approved type and shall conform to requirements of ASTM Specification A31.

Fabrication.—Lock-bar steel pipe is made in sections approximately 30 ft long between centers of end rivet holes, except for closures. Sections may be tapered or may consist of alternate in-and-out sections. The nominal diameter is the minimum inside diameter measured to the plate for pipes with tapered courses and to the inside of the inner course for pipe made with alternate in-and-out courses.

A specimen cut transversely across the longitudinal joint shall develop not less than 90 per cent of the ultimate tensile strength of the plate.

Each section of pipe, except specials which in the judgment of the engineer cannot be so tested, shall be tested and made tight under a hydrostatic pressure which will develop in the plates a stress specified by the purchaser.

Each section of pipes and each special shall be marked after coating with a serial number painted in white on the inside at each end.

ASTM Standard Specifications for SPIRAL-WELDED STEEL OR IRON PIPE

Serial Designation A211-40 (ASA B36.16)

Abstracted[2]

These specifications cover spiral-welded steel or iron pipe 4 to 48 in. inclusive in diameter and $\frac{1}{16}$ to $\frac{11}{64}$ in. (No. 16 to No. 8 U. S. Standard gage) in wall

[1] For complete specifications, reference may be made to AWWA 7A.2. Copies of this tentative standard can be purchased from the American Water Works Association, 500 Fifth Ave., New York 18, N. Y.

[2] For complete specification, reference may be made to ASTM A211 (see note, p. 369). Abstract includes Emergency Alternate Provisions, issued Aug. 18, 1942.

thickness. When used for waterworks service, the wall thicknesses of 4-, 6-, and 8-in. pipe shall be either 0.105 in. or 0.135 in. and the wall thickness of 10- to 20-in. inclusive shall be 0.135 in., unless heavier thicknesses are produced for other purposes. Spiral welded pipe also may be purchased to AWWA Specifications 7A.3 and 7A.4.[1]

Material.—The material shall conform to one of the following ASTM Specifications for plate, A78, A89, A70, A10, or A129 as specified.

Manufacture.—Pipe shall be fabricated from sheet or strip wound in a spiral and having seams joined by the electric-fusion-welding process. The spiral seam may be lapped, locked, or butted together for welding.

Hydrostatic Test.—Unless otherwise specified, each length shall be tested at the mill to a hydrostatic pressure equal to 150 per cent of the working pressure, but in no case shall the maximum stress produced exceed 80 per cent of the yield point of the material.

Weights and Dimensions.—Permissible variations on the theoretical weights of the pipe shall be governed by permissible variations for the plates. Variations in outside diameter shall be consistent with the requirements of field joints as agreed upon by the manufacturer and the purchaser.

Lengths.—Pipe shall be furnished in standard lengths of 20, 30, or 40 ft as specified on the purchase order, except that closure pieces and specials shall be furnished as specified.

Protective Coating.—If required by the purchaser, the pipe after testing shall be cleaned and given a protective coating of the kind and in the manner specified by the purchaser. For waterworks service it is recommended that coatings conform to the Standard Specifications for Coal Tar Enamel Protective Coatings for Steel Water Pipe of Sizes 30 in. and Over, AWWA No. 7A5, of the American Water Works Association.

Marking.—Each length shall be legibly marked to show the name or brand of the manufacturer, the size (diameter and wall thickness), the hydrostatic test pressure, and the specification number: A211.

ASTM Tentative Specifications for WELDED ALLOYED OPEN-HEARTH IRON PIPE

Serial Designation A253-44

Abstracted[2]

These specifications cover black and galvanized furnace butt-welded and electric-resistance-welded alloyed open-hearth iron pipe. Pipe is intended for coiling, bending, flanging, and other special purposes. Pipe 2 in. and under may be specified for close coiling. Furnace butt-welded pipe is not intended for flanging. Galvanizing, when specified, shall be in accordance with ASTM Specification A120 (see page 369). Pipe 1½ in. and under shall be furnace butt-welded. Sizes 2, 2½, and 3 in. may be either furnace butt-welded or electric-resistance-welded. Pipe over 3 in. shall be electric-resistance-welded.

[1] Copies may be obtained from the American Water Works Association, 500 Fifth Ave., New York 18, N. Y.

[2] For complete specification, reference may be made to ASTM A253 (see note, p. 369).

The pipe shall be made from alloyed basic open-hearth iron conforming to the following chemical and physical requirements.

Chemical Composition

Carbon, max, per cent	0.05
Phosphorus, max, per cent	0.015
Sulphur, max, per cent	0.040
Copper, min, per cent	0.40
Molybdenum, min, per cent	0.05
Total carbon, manganese, phosphorus, sulphur, and silicon, max, per cent	0.25[1]

Tensile Properties

Tensile strength, min, psi	46,000
Yield point, min, psi	30,000
Elongation in 2 in. min per cent	30

Tension Tests.—One longitudinal tension test shall be made on one pipe length from each lot of 500 lengths or fraction thereof of each size. The tension test specimen shall be cut longitudinally from the pipe and not flattened between gage marks, or if desired may consist of a full section of pipe.

Bend Tests.—For pipe 2 in. and under, a bend test shall be made on one pipe length from each lot of 500 lengths or fraction thereof of each size. A sufficient length of pipe shall stand being bent cold through 90 deg around a cylindrical mandrel the diameter of which is twelve times the nominal pipe diameter. When ordered for close coiling, the angle of bend shall be 180 deg and the diameter of the mandrel eight times the pipe diameter.

Flattening Test.—For pipe over 2 in., a flattening test shall be made on a section 4 to 6 in. long from one pipe length from each lot of 500 lengths or fraction thereof of each size. When electric-resistance-welded pipe is ordered for flanging, the crop ends from each length shall stand the flattening test. For furnace butt-welded pipe the weld shall be located 45 deg from the line of direction of the applied force. For electric-resistance-welded pipe, the weld shall be located 90 deg from the direction of the force.

No cracks or breaks shall develop in furnace butt-welded pipe until the distance between the parallel plates is 60 per cent of the original outside diameter. No opening in the weld shall take place in electric-resistance-welded pipe until the distance between the plates is less than two-thirds of the original outside diameter. No cracks or breaks in the metal elsewhere than in the weld shall occur until the distance between the plates is less than one-third of the original outside diameter of the pipe, but in no case less than five times the thickness of the pipe wall.

Hydrostatic Test.—Each length of pipe shall be tested at the mill to the hydrostatic pressures as shown in the table on page 405. Pipe 2 in. and larger shall be jarred near one end, while under test pressure, with a 2-lb hammer or its equivalent.

Standard Weights.—The standard weights with the corresponding wall thicknesses for "standard" and "extra-strong" pipe of various nominal inside diameters are given in Table II of the specification which, with the omission of the ⅛-in. nominal size, is identical with Table II of A120 reproduced on page 371.

[1] Permitted up to 0.31 per cent on check analysis by purchaser.

HYDROSTATIC TEST PRESSURES FOR WELDED ALLOYED OPEN-HEARTH IRON PIPE[a]

(Pressures expressed in pounds per square inch)

Size (nominal inside diameter), in.	Standard-weight pipe		Extra-strong pipe	
	Furnace butt-welded	Electric welded	Furnace butt-welded	Electric welded
1/8 to 1, incl.	700	.....	850	
1¼ to 3, incl.	800	1,000	1,100	1,500
3½ to 6, incl.	...	1,200	.....	1,700
8	...	1,200	.....	1,700
10 to 12, incl.	...	1,000	.....	1,600

[a] For pipe over 12 in. in nominal pipe size, the test pressures should be calculated by the formula $p = \frac{2St}{D}$; where p = pressure, psi, S = fiber stress, 60 per cent of the specified yield point, t = thickness of wall, in., and D = outside diameter, in.

Variations in Weights and Dimensions.—The weights of pipe shall not vary more than plus or minus 5 per cent from the weights given in Table II, page 371. The outside diameter at any point for pipe 1½ in. and under in normal diameter shall not be more than 1/64 in. over nor more than 1/32 in. under the standard specified. For pipe 2 in. and over the outside diameter shall not vary more than ±1 per cent from the standard specified. The minimum wall thickness at any point shall be not more than 12.5 per cent under the normal wall thickness specified.

Lengths.—Standard-weight pipe shall be in random lengths of 16 to 22 ft, unless otherwise specified. Not more than 5 per cent of the total number of lengths may be jointers, which are two pieces coupled together. When ordered with plain ends, 5 per cent may be in lengths of 12 to 16 ft. Extra-strong pipe is commonly furnished with plain ends in random lengths of 12 to 22 ft. Five per cent may be in lengths of 6 to 12 ft unless otherwise specified.

Ends.—Standard-weight pipe is commonly furnished with threads and couplings. Extra-strong pipe is commonly finished with plain ends square cut but may be ordered with plain ends beveled or with threaded ends.

Threads.—Threads shall be in accordance with the American Standard Pipe Thread, ASA B2.1 (see abstract, page 481).

Couplings.—Each length of threaded pipe shall be provided with one coupling. Taper-tapped couplings shall be furnished on all weights of threaded pipe 2½ in. and larger. For sizes 2 in. and smaller, it is commercial practice to furnish straight-tapped couplings for standard-weight (Schedule 40) pipe and taper-tapped couplings for extra-strong (Schedule 80) and double-extra-strong pipe. If taper-tapped couplings are required for sizes 2 in. and smaller on standard-weight (Schedule 40) pipe, line pipe in accordance with API Specification 5L should be ordered, thread lengths to be in accordance with the American Standard for Pipe Threads (ASA B2.1). The taper-tapped couplings provided on line pipe in these sizes may be used on mill-threaded standard-weight pipe of the same size.

Marking.—Each length of pipe shall be legibly marked by rolling, stamping, or stenciling to show the name or brand of the manufacturer; the type of pipe (that is, furnace-welded or electric-resistance-welded); XS for extra strong; ASTM A253, and the length; except that for small diameter pipe which is bundled, this information may be marked on a tag securely attached to each bundle.

AWWA Standard Specifications for STEEL WATER PIPE OF SIZES UP TO BUT NOT INCLUDING 30 IN.

Serial Designation 7A.4-1941

Abstracted[1]

These specifications cover steel pipe in random, double-random, or specified laying lengths for the conveyance of water.

The pipe covered by these specifications is manufactured by the following processes and in the nominal size ranges indicated in wall thicknesses and weights as covered by Tables XIII and XIV.

Mill Pipe:

Seamless in sizes ⅛ to 24 in. inclusive.

Lap-welded in sizes 1¼ to 20 in. inclusive.

Butt-welded in sizes ⅛ to 3½ in. O.D. inclusive.

Electric-welded:

Resistance-welded in sizes ⅛ in. to but not including 30 in.

Fusion-welded:

Straight seam in sizes up to but not including 30 in.

Spiral seam in sizes up to but not including 30 in.

Fabricated Pipe:

Electric-fusion-welded:

Straight seam in sizes up to but not including 30 in.

Spiral seam in sizes up to but not including 30 in.

Chemical Composition.—The steel for *mill* pipe shall be of good welding quality and may be made by the acid-bessemer process or the open-hearth process and shall conform to the following chemical composition:

CHEMICAL PROPERTIES

	Bessemer	Open-hearth
Manganese, not under, per cent	0.30	0.30
Phosphorus, not over, per cent	0.11	0.045
Sulphur, not over, per cent	0.065	0.060

Steel plate for *fabricated* pipe shall conform to the chemical requirements of Grade B plates of ASTM A78, or as otherwise specified by the purchaser.

[1] Abstract is in accordance with tentative revision as published in the *AWWA Jour*, April, 1943. For complete specification, reference may be made to latest revision of AWWA 7A.4. Copies may be obtained from the American Water Works Association, 500 Fifth Ave., New York 18, N. Y.

Steel sheets or coils used in *fabricated* pipe shall conform to the chemical requirements of Grade A material of ASTM A245, or as otherwise specified by the purchaser.

The inspector may witness the manufacture and test of plates, sheets, or coils at the rolling mill and/or he may require that the manufacturer furnish mill test reports on each heat of steel. The inspector may require specimen plates or sheets of steel that will be used in the manufacture of the pipe. The inspector's approval of plates, sheets, or coils shall be obtained before manufacture of any pipe under these specifications.

Physical Properties.—The steel for *mill* pipe shall conform to the minimum longitudinal tensile properties in the following table:

PHYSICAL PROPERTIES

Property	Bessemer lap-welded or butt-welded	Open-hearth	
		Lap-welded or butt-welded	Grade A seamless or electric-welded
Tensile strength, psi	50,000	45,000	48,000
Yield point, psi	30,000	25,000	30,000
Elongation in 8 in., per cent	18	22	
Elongation in 2 in., per cent	......	......	30

Steel plate for *fabricated* pipe shall conform to the physical properties specified for Grade B plates of ASTM A78, or as otherwise specified by the purchaser.

Steel sheets or coils used in *fabricated* pipe shall conform to the physical properties of Grade A material of ASTM A245, or as otherwise specified by the purchaser.

Ends of Sections.—Ends of pipe sections and special sections shall be any of the following types as specified by the purchaser: (1) ends for mechanical-coupled field joints, (2) ends with slip joints for field welding, (3) plain ends or beveled ends for field butt welding, (4) plain ends fitted with butt straps for field welding, (5) beveled ends fitted with butt straps for field welding (this applies to sizes 24 in. and over), (6) ends for riveted field joints, (7) bell-and-spigot ends for calked joints, (8) plain ends fitted with flanges, or (9) taper ends for driven field joints.

Fabrication.—The longitudinal seams of electric-resistance-welded and fusion-welded straight-seam and spiral-seam *mill* pipe and *fabricated* pipe shall be butt-welded. Shop girth seams shall be butt-welded or lap-joint welded. The purchaser may specify the maximum number of girth seams to be permitted in each section of *fabricated* pipe. Requirements for circumferential welding of straight pipe lengths are given in an appendix Sec. A4.2 of these specifications not reproduced here.

All lap breaking, rolling, and cleaning of plate surfaces to be welded and all fitting-up operations as well as the qualification of the welding operators, welding procedure, etc., for *fabricated* pipe shall be in accordance with requirements of AWWA 7A.3 (see abstract page 416).

Tolerances.—The outside diameter of *mill* pipe shall not vary more than 1 per cent over or under the size specified, except that the ends of pipe on sizes

10¾ in. outside diameter and under intended for either mechanical compression couplings or for field welding shall be not more than 1/64 in. smaller than the specified outside pipe diameter for a distance of 8 in. from the ends as determined by a steel tape circumferentially applied and shall permit the passing for a distance of 8 in. of a ring gage which has a bore 1/16 in. larger than the specified diameter. Sizes 12 in. and larger in outside diameter shall not be more than 1/32 in. smaller for this same distance and the ring gage shall be 3/32 in. larger. Ends for other types of mechanical couplings shall have tolerance within the limits specified by the manufacture of the type of coupling to be used.

Each section of *fabricated* pipe shall be accurately measured with a steel tape and shall be within tolerances of −1/16 and +1/8 in. of the outside circumference computed from the specified normal inside diameter and plate thickness for (*a*) plain ends of slip-joint pipe for field welding, (*b*) plain ends with butt strap for field welding, (*c*) beveled end fitted with butt straps for field welding, (*d*) both ends of inside section of inside and outside pipe sections and the smaller end of tapered section for lap-riveted field joints, (*e*) plain ends fitted with flanges. Ends for mechanical couplings shall be within the limits required by the coupling manufacturers.

The inside circumference of *fabricated* pipe, for a distance of not less than 8 in. from the ends, shall be with tolerances of −1/16 in. and +1/8 in. of the inside circumference computed from the normal inside diameter for the following types of ends: (*a*) plain ends or beveled for field butt welding, (*b*) bumped ends for riveted field joints, (*c*) plain end of bell-and-spigot pipe for calked joints, (*d*) ends for riveted field joints (tolerance for tapered pipe to apply at the smaller end only).

The inside diameter of the bell of slip-joint pipe for field welding shall be between 1/32 and 1/16 in. greater than the outside diameter of the plain end. The inside diameter of the outside section of inside and outside pipe sections and the inside diameter of the larger end of tapered sections of lap-riveted field joints shall be between 1/16 and 1/8 in. greater than the outside diameter of the inside section or the smaller end of the tapered section; such outside diameter shall be determined from the nominal inside diameter and the specified plate thickness.

The wall thickness of steel plate shall be ordered to thickness with a maximum under-thickness variation of 0.01 in. Sheets or coils shall be ordered to thickness and tolerances given in ASTM A245.

The length of straight sections of fabricated pipe shall not vary more than ±2 in. from the specified length. Special sections shall not vary more than ±1/8 in.

When finished pipe is to be spun-enamel lined by the centrifugal casting process and the spinning equipment does not hold the pipe in axial alignment, any free pipe sections with faulty alignment exceeding 1/8 in. for each 10 ft. in length from a line parallel to the axis of the pipe shall be straightened. Pipe not to be spun-enamel lined shall be commercially straight.

Tests.—Transverse weld tests required on electric-welded pipe are for the purpose of determining minimum tensile strength only and shall not be required on sizes under 8 in. in outside diameter. For transverse weld tests, the test specimen shall consist of a strip cut transversely from the end of the pipe with a width of 1½ in. with the weld located about the middle of its length and machined parallel for a length of 2 in. Specimens may be flattened hot or may be flattened

cold and normalized before testing. The tensile strength across the weld shall not be less than the minimum tensile strength of the grade of steel ordered.

For fabricated pipe, the transverse weld test specimens for electric fusion-welded pipe may be taken from a welded test plate (or coupon). The welded test plate (or coupon) shall be cut from steel of the same specifications and thickness as the pipe, have edges prepared for welding, and then be attached at one end of the longitudinal joint of the pipe so that the edges to be welded in the test plate form a continuation of, and duplication of, the corresponding edges of the longitudinal joint. In this case, the weld metal shall be deposited in the plate continuously with the weld metal deposited in the longitudinal joint.

Where specimens by agreement between the inspector and the manufacturer are taken from points other than the ends of the pipe, the openings resulting from removal of test specimens shall be patched in a manner approved by the inspector.

When *fabricated* pipe is furnished for working pressure exceeding 200 psi, additional tests of welds and welded joints as required in AWWA 7A.3 (see abstract page 416) will be made if specifically requested by the purchaser in the call for bids.

Crop ends or rings 3 in. in width shall be cut from each end of lap-welded or electric-resistance-welded pipe and flattened between parallel plates with the weld at the point of maximum bending until opposite walls of the pipe meet. No opening in the weld shall take place before the distance between the plates is less than two-thirds of the original outside diameter of the pipe. No cracks or breaks elsewhere in the pipe metal shall occur before the distance between the plates is less than one-third of the outside pipe diameter.

For butt-welded pipe over 2 in. in nominal diameter, a section of pipe 4 to 6 in. long cut from each lot of 100 lengths or less of each size shall be flattened between parallel plates with the weld located 45 deg from the point of maximum bending until the distance between the plates is 60 per cent of the outside diameter of the pipe, without developing cracks.

A bend test specimen 1 in. wide by 6 in. long which has been cut across the weld from each test length of electric-fusion-welded pipe of sizes 8 in. and over in outside diameter shall stand being bent cold through 180 deg around a pin the diameter of which is 4½ times the plate thickness without developing cracks. The side of the specimen representing the inside of the pipe shall be placed next to the pin.

One longitudinal tension test shall be made on one length from each lot of 300 lengths or fraction thereof, of each size 6 in. (nominal size) and under, from each lot of 200 lengths, or fraction thereof of each size 8 in. (nominal size) to 20 in. inclusive, and from each lot of 100 lengths or fraction thereof of each size 22 in. and larger. In addition to the above, one transverse weld test on electric-welded pipe and one bend test on electric-fusion-weld pipe may be made on one length from each lot of 200 lengths or fraction thereof, of sizes 8 to 20 in. inclusive, and on one length from each lot of 100 lengths or fraction thereof of each size 22 in. and larger.

All straight pipe shall be tested at the mill or shop to the hydrostatic pressures specified in Table XIII for mill pipe and in Table XIV for fabricated pipe. Pressures shall be maintained for not less than 5 sec. No hammering is required. Test pressures for diameters and wall thicknesses not listed in the tables shall be made proportional to the pressures given therein.

Bends, branch connections, and special sections made by certified welding operators from tested pipe shall not be subjected to hydrostatic test pressure.

TABLE XIII.—PHYSICAL DATA ON STEEL PIPE FOR UNDERGROUND WATER SERVICE—AWWA 7A.4 MILL PIPE

Nominal size, in.	Outside diameter, in.	Wall thickness (decimal), in.	Weight per linear foot, lb	Working pressure (safety factor of 4.0): 40,000-psi fiber stress, psi	Working pressure (safety factor of 4.0): 50,000-psi fiber stress, psi	Approximate ultimate bursting pressure: 40,000-psi fiber stress, psi	Approximate ultimate bursting pressure: 50,000-psi fiber stress, psi	Test pressure: Butt weld—plain or threaded ends, psi	Test pressure: Lap weld—Grade A seamless or electric-welded, psi: Plain ends	Test pressure: Lap weld—Grade A seamless or electric-welded, psi: Threaded ends
1/8	0.405	0.068	0.24	3,360	.....	13,430	.....	700	700	700
1/4	0.540	0.088	0.42	3,260	.....	13,040	.....	700	700	700
3/8	0.675	0.091	0.57	2,700	.....	10,780	.....	700	700	700
1/2	0.840	0.109	0.85	2,600	.....	10,380	.....	700	700	700
3/4	1.050	0.113	1.13	2,150	.....	8,610	.....	700	700	700
1	1.315	0.133	1.68	2,020	.....	8,090	.....	700	700	700
1 1/4	1.660	0.140	2.27	1,690	.....	6,750	.....	800	1,000	1,000
1 1/2	1.900	0.145	2.72	1,530	.....	6,100	.....	800	1,000	1,000
2	2.375	0.154	3.65	1,300	1,620	5,190	6,480	800	1,000	1,000
2 1/2	2.875	0.203	5.79	1,410	1,770	5,650	7,060	800	1,000	1,000
Note.—Sizes above shall be considered as service pipe.										
3	3.500	0.125	4.51	710	890	2,860	3,570	800	1,300	
		0.156	5.58	890	1,110	3,560	4,460	800	1,600	
		0.188	6.63	1,070	1,340	4,300	5,370	800	1,900	
		0.216	7.58	1,230	1,540	4,940	6,170	800	2,200	1,000
		0.250	8.68	1,430	1,790	5,720	7,140	800	2,500	
		0.281	9.67	1,610	2,010	6,420	8,030	800	2,500	
		0.300	10.25	1,710	2,140	6,860	8,570	800	2,500	
3 1/2	4.000	0.125	5.17	.....	780	......	3,120	...	1,100	
		0.156	6.41	.....	980	......	3,900	...	1,400	
		0.188	7.63	.....	1,180	......	4,700	...	1,700	
		0.226	9.11	.....	1,410	......	5,650	...	2,000	1,200
		0.250	10.01	.....	1,560	......	6,250	...	2,200	
		0.281	11.17	.....	1,760	......	7,020	...	2,500	
		0.318	12.51	.....	1,990	......	7,950	...	2,500	
4	4.500	0.125	5.84	.....	690	......	2,780	...	1,000	
		0.156	7.25	.....	870	......	3,470	...	1,200	
		0.188	8.64	.....	1,040	......	4,180	...	1,500	
		0.219	10.00	.....	1,220	......	4,870	...	1,700	
		0.237	10.79	.....	1,320	......	5,270	...	1,900	1,200
		0.250	11.35	.....	1,390	......	5,560	...	2,000	
		0.281	12.67		1,560	......	6,240	.....	2,200	
		0.312	13.98	.....	1,730	......	6,930	...	2,500	
		0.337	14.98	.....	1,870	......	7,490	...	2,500	
5	5.563	0.156	9.02	.....	700	......	2,800	...	1,000	
		0.188	10.76	.....	850	.....	3,380	...	1,200	
		0.219	12.49	.....	980	......	3,940	...	1,400	
		0.258	14.62	.....	1,160	......	4,640	...	1,700	1,200
		0.281	15.87	.....	1,260	......	5,050	...	1,800	
		0.312	17.52	.....	1,400	......	5,610	...	2,000	
		0.344	19.16	.....	1,550	......	6,180	...	2,200	
		0.375	20.78	.....	1,690	......	6,740	...	2,400	

TABLE XIII.—(*Continued*)

Nominal size, in.	Outside diameter, in.	Wall thickness (decimal), in.	Weight per linear foot, lb	Working pressure (safety factor of 4.0)		Approximate ultimate bursting pressure		Test pressure		
				40,000-psi fiber stress, psi	50,000-psi fiber stress, psi	40,000-psi fiber stress, psi	50,000-psi fiber stress, psi	Butt weld—plain or threaded ends, psi	Lap weld—Grade A seamless or electric-welded, psi: Plain ends	Lap weld—Grade A seamless or electric-welded, psi: Threaded ends
6	6.625	0.188	12.89	.....	710	......	2,840	...	1,000	
		0.219	14.97	.....	830	......	3,300	...	1,200	
		0.250	17.02	.....	940	......	3,780	...	1,400	
		0.280	18.97	.....	1,060	......	4,220	...	1,500	1,200
		0.312	21.07	.....	1,180	......	4,710	...	1,700	
		0.344	23.06	.....	1,300	......	5,190	...	1,900	
		0.375	25.03	.....	1,420	......	5,660	...	2,000	
		0.432	28.57	.....	1,630	......	6,520	...	2,300	
8	8.625	0.188	16.90	.....	550	.....	2,180	...	800	
		0.219	19.64	.....	640	......	2,540	...	900	
		0.250	22.36	.....	730	.....	2,900	...	1,000	
		0.277	24.70	.....	800	......	3,210	...	1,200	1,200
		0.312	27.74	.....	900	......	3,620	...	1,300	
		0.322	28.55	.....	930	......	3,730	...	1,300	1,200
		0.344	30.40	.....	1,000	......	3,990	...	1,400	
		0.375	33.04	.....	1,090	......	4,350	...	1,600	
		0.438	38.26	.....	1,270	......	5,080	...	1,800	
		0.500	43.39	.....	1,450	......	5,800	...	2,100	
10	10.750	0.188	21.15	.....	440	......	1,750	...	650	
		0.219	24.60	.....	510	......	2,040	...	750	
		0.250	28.04	.....	580	......	2,330	...	850	
		0.279	31.20	.....	650	......	2,600	...	1,000	1,000
		0.307	34.24	.....	710	......	2,860	...	1,000	1,000
		0 344	38.20	.....	800	......	3,200	...	1,100	
		0.365	40.48	.....	850	......	3,400	...	1,200	1,000
		0.438	48.19	.....	1,020	......	4,070	...	1,500	
		0.500	54.74	.....	1,160	......	4,650	...	1,700	
12	12.750	0.188	25.15	.....	370	......	1,480	...	500	
		0.219	29.28	.....	430	......	1,720	...	600	
		0.250	33.38	.....	490	......	1,960	...	700	
		0.281	37.45	.....	550	......	2,200	...	800	
		0.312	41.51	.....	610	......	2,450	...	900	
		0.330	43.77	.....	650	......	2,590	...	1,000	1,000
		0.344	45.55	.....	670	......	2,700	...	1,000	
		0.375	49.56	.....	740	......	2,940	...	1,100	1,000
		0.438	57.53	.....	860	......	3,440	...	1,200	
		0.500	65.42	.....	980	......	3,920	...	1,400	
14	14.000	0.188	27.65	.....	340	......	1,340	...	450	
		0.219	32.19	.....	390	......	1,560	...	550	
		0.250	36.71	.....	450	......	1,780	...	650	
		0.281	41.21	.....	500	......	2,010	...	700	
		0.312	45.68	.....	560	......	2,230	...	800	
		0.344	50.14	.....	610	......	2,460	...	900	
		0.375	54.57	.....	670	......	2,680	...	950	950
		0.438	63.37	.....	780	......	3,130	...	1,100	
		0.500	72.09	.....	890	......	3,570	...	1,300	

TABLE XIII.—(*Continued*)

Nominal size, in.	Outside diameter, in.	Wall thickness (decimal), in.	Weight per linear foot, lb	Working pressure safety factor of 4.0: 40,000-psi fiber stress, psi	Working pressure safety factor of 4.0: 50,000-psi fiber stress, psi	Approximate ultimate bursting pressure: 40,000-psi fiber stress, psi	Approximate ultimate bursting pressure: 50,000-psi fiber stress, psi	Test pressure: Butt weld—plain or threaded ends, psi	Test pressure: Lap weld—Grade A seamless or electric-welded, psi: Plain ends	Test pressure: Lap weld—Grade A seamless or electric-welded, psi: Threaded ends
16	16.000	0.188	31.66	.....	290	......	1,180	...	400	
		0.219	36.86	.....	340	......	1,370	...	500	
		0.250	42.05	.....	390	......	1,560	...	550	
		0.281	47.22	.....	440	......	1,760	...	650	
		0.312	52.36	.....	490	......	1,950	...	700	
		0.344	57.48	.....	540	......	2,150	...	750	
		0.375	62.58	.....	590	......	2,340	.....	850	850
		0.438	72.72	.....	680	......	2,740	...	1,000	
		0.500	82.77	.....	780	......	3,120	...	1,100	
18	18.000	0.188	35.67	.....	260	......	1,040	...	350	
		0.219	41.54	.....	300	......	1,220	...	400	
		0.250	47.39	.....	350	......	1,390	...	500	
		0.281	53.22	.....	390	......	1,560	...	550	
		0.312	59.03	.....	430	......	1,730	...	600	
		0.344	64.82	.....	480	......	1,910	...	700	
		0.375	70.59	.....	520	.....	2,080	...	750	
		0.438	82.06	.....	600	......	2,430	...	900	
		0.500	93.45	.....	690	......	2,780	...	1,000	
20	20.000	0.188	39.67	.....	240	......	940	...	350	
		0.219	46.21	.....	270	......	1,100	...	400	
		0.250	52.73	.....	310	......	1,250	...	450	
		0.281	59.23	.....	350	......	1,400	...	500	
		0.312	65.71	.....	390	......	1,560	...	550	
		0.344	72.16	.....	430	......	1,720	...	600	
		0.375	78.60	.....	470	......	1,880	...	700	
		0.438	91.41	.....	550	......	2,190	...	800	
		0.500	104.13	.....	630	.....	2,500	...	900	
22	22.000	0.188	43.68	.....	210	......	860	...	300	
		0.219	50.88	.....	250	......	1,000	...	350	
		0.250	58.07	.....	280	......	1,140	...	400	
		0.312	72.38	.....	350	......	1,420	...	500	
		0.344	79.51	.....	390	......	1,560	...	550	
		0.375	86.61	.....	430	......	1,700	...	600	
		0.438	100.75	.....	500	......	1,990	...	700	
		0.500	114.81	.....	570	......	2,270	...	800	
24	24.000	0.188	47.68	.....	200	......	780	...	300	
		0.219	55.56	.....	230	......	910	...	300	
		0.250	63.41	.....	260	......	1,040	.....	350	
		0.312	79.06	.....	330	......	1,300	...	450	
		0.344	86.85	.....	360	......	1,430	...	500	
		0.375	94.62	.....	390	......	1,560	...	550	
		0.438	110.10	.....	460	......	1,820	.....	650	
		0.500	125.49	.....	520	....	2,080	.....	750	

TABLE XIII.—(*Concluded*)

Nominal size, in.	Outside diameter, in.	Wall thickness (decimal), in.	Weight per linear foot, lb	Working pressure (safety factor of 4.0)		Approximate ultimate bursting pressure		Test pressure		
				40,000-psi fiber stress, psi	50,000-psi fiber stress, psi	40,000-psi fiber stress, psi	50,000-psi fiber stress, psi	Butt weld—plain or threaded ends, psi	Lap weld—Grade A seamless or electric-welded, psi: Plain ends	Lap weld—Grade A seamless or electric-welded, psi: Threaded ends
26	26.000	0.188	51.69	.....	180	......	720	...	250	
		0.219	60.23	.....	210	......	840	.....	300	
		0.250	68.75	.....	240	......	960	.....	350	
		0.281	77.25	.....	270	......	1,080	.....	400	
		0.312	85.73	.....	300	......	1,200	.....	400	
		0.375	102.62	.....	360	......	1,440	.....	500	
		0.438	119.44	.....	420	......	1,680	.....	600	
		0.500	136.17	.....	480	......	1,920	...	700	
28	28.000	0.188	55.69	.....	170	......	670	...	250	
		0.219	64.90	.....	200	......	780	...	250	
		0.250	74.09	.....	220	......	890	.....	300	
		0.281	83.26	.....	250	......	1,000	.....	350	
		0.312	92.40	.....	280	......	1,110	...	400	
		0.375	110.63	.....	340	......	1,340	.....	450	
		0.438	128.78	.....	390	......	1,560	.....	550	
		0.500	146.85	.....	450	......	1,780	.....	650	

When not fabricated from tested pipe, all bends, branch connections, and special sections shall be subjected to a hydrostatic pressure specified by the purchaser. It is recommended that such test pressure shall not exceed 1½ times the working water pressure.

Cleaning and Coating Pipe Surfaces.—If not otherwise specified by the purchaser, each section of pipe shall be thoroughly cleaned both inside and outside. It is recommended that all pipe be properly coated in accordance with purchaser's supplementary specification which shall preferably be based on AWWA 7A.6.

Delivery.—Each length of uncoated mill pipe shall be legibly marked by stenciling, stamping, or rolling to show by whom manufactured, the class of pipe, the outside diameter, nominal weight per foot or wall thickness, the over-all length, the test pressure, and any additional marks or symbols specified by purchaser. In the case of coatings applied to mill pipe at the mill the above markings shall be applied by stenciling after coating.

When so specified, pipe sections and special sections shall be delivered in the order required and shall be loaded as required by the purchaser.

TABLE XIV.—PHYSICAL DATA ON STEEL FABRICATED PIPE, AWWA 7A.4[1]

Outside diameter, in.	Wall thickness[2]		Weight per linear foot, lb	Working pressure (safety factor of 4.0; 50,000-psi fiber stress), psi	Approx. ultimate bursting pressure 50,000-psi fiber stress, psi	Test pressure, psi
	Mfr.'s std. gage No.	Decimal, in.				
4	12	0.105	4.46	660	2,620	1,000
	10	0.135	5.69	840	3,380	1,200
6	12	0.105	6.75	440	1,750	600
	10	0.135	8.64	560	2,250	800
		0.188	11.63	780	3,130	1,100
		0.219	13.50	910	3,650	1,300
		0.250	15.35	1,040	4,170	1,500
8	12	0.105	9.04	330	1,310	450
	10	0.135	11.58	420	1,690	600
	7	0.179	15.36	560	2,240	800
		0.188	15.64	590	2,350	850
	3	0.239	20.32	750	2,990	1,100
10	10	0.135	14.53	340	1,350	450
	7	0.179	19.28	450	1,790	650
		0.188	19.64	470	1,880	700
	3	0.239	25.55	600	2,390	850
12	10	0.135	17.47	280	1,120	400
	7	0.179	23.21	370	1,490	550
		0.188	23.65	390	1,570	550
	3	0.239	30.79	500	1,990	700
14	10	0.135	20.42	240	960	350
	7	0.179	27.14	320	1,280	450
		0.188	27.65	340	1,340	450
	3	0.239	36.03	430	1,710	600
16	10	0.135	23.36	210	840	300
	7	0.179	31.06	280	1,120	400
		0.188	31.66	290	1,180	400
	3	0.239	41.26	370	1,500	550
		0.250	42.05	390	1,560	550
		0.312	52.36	490	1,950	700
18	10	0.135	26.31	190	750	250
	7	0.179	34.99	250	990	350
		0.188	35.67	260	1,040	350
	3	0.239	46.50	330	1,330	450
		0.250	47.39	350	1,390	500
		0.312	59.03	430	1,730	600

[1] Pipe thicknesses shown above the single hairline crossrule (shown thus— —) in each pipe size group are those used in the Pacific Coast and Rocky Mountain areas.

[2] Other diameters and other wall thicknesses can be furnished if required.

TABLE XIV.—(*Concluded*)

Outside diameter, in.	Wall thickness[2] Mfr.'s std. gage No.	Wall thickness[2] Decimal, in.	Weight per linear foot, lb	Working pressure (safety factor of 4.0) 50,000-psi fiber stress, psi	Approx. ultimate bursting pressure 50,000-psi fiber stress, psi	Test pressure, psi
20	10	0.135	29.25	170	680	250
	7	0.179	38.92	220	900	300
		0.188	39.67	240	940	350
	3	0.239	51.73	300	1,200	400
		0.250	52.73	310	1,250	450
		0.312	65.71	390	1,560	550
		0.375	78.60	470	1,880	700
22	7	0.179	42.84	200	820	300
		0.188	43.68	210	860	300
	3	0.239	56.97	270	1,090	400
		0.250	58.07	280	1,140	400
		0.312	72.38	350	1,420	500
		0.375	86.61	430	1,700	600
24	7	0.179	46.77	190	740	250
		0.188	47.68	200	780	300
	3	0.239	62.21	250	1,000	350
		0.250	63.41	260	1,040	350
		0.312	79.06	330	1,300	450
		0.375	94.62	390	1,560	550
		0.438	110.10	460	1,820	650
		0.500	125.49	520	2,080	750
26	7	0.179	50.70	170	690	250
		0.188	51.69	180	720	250
	3	0.239	67.44	230	920	300
		0.250	68.75	240	960	350
		0.312	85.73	300	1,200	400
		0.375	102.62	360	1,440	500
		0.438	119.44	420	1,680	600
		0.500	136.17	480	1,920	700
28	7	0.179	54.62	160	640	250
		0.188	55.69	170	670	250
	3	0.239	72.68	210	860	300
		0.250	74.09	220	890	300
		0.312	92.40	280	1,110	400
		0.375	110.63	340	1,340	450
		0.438	128.78	390	1,560	550
		0.500	146.85	450	1,780	650

AWWA Standard Specifications for ELECTRIC-FUSION-WELDED STEEL WATER PIPE OF SIZES 30 IN. AND OVER

Serial Designation 7A.3-1940

Abstracted[1]

These specifications cover the manufacture of straight-seam or spiral-seam electric-fusion-welded steel pipe for the conveyance of water.

The purchaser is required to specify the inside diameter, thickness of plate, laying length, details of ends, etc., which are desired. Provision is made for inspection of the plate and of all phases of manufacture of the pipe.

All welds shall be made by an electric shielded arc method, using either automatic welding or manual welding. All longitudinal, spiral, and girth seams of straight pipe sections, and special sections where practicable, shall be welded with an automatic welding machine. The penetration, extent of reinforcement, and correction of weld defects are covered in detail. Weld test specimen plates shall be furnished for every 300 lin ft of pipe, or oftener if the inspector requires. Two etch specimens, one reduced tensile, one free bend, one reverse bend, and one nick-break test specimen are required from each test specimen plate. The test results shall show sound welds with tensile and yield strength equal to the minimum of the specified range of the plate used.

Straight pipe sections and tapered sections shall be hydrostatically tested at twice the specified working water pressure (including allowance for water hammer) but not to exceed 80 per cent of the yield point of the specified plate or 22,000 psi as computed by Barlow's formula (see page 41).

Special requirements for cleaning and coating pipe, marking and delivering pipe, and optional requirements for stress relieving and radiographic inspection are included.

WROUGHT-IRON PIPE AND BOLTS

The essential feature of wrought iron is the presence, in a finely divided and uniformly distributed state, of a quantity of siliceous slag. In the original hand-puddling process a charge of pig iron and scrap is melted in a puddling furnace and most of the carbon driven off. The silicon, phosphorus, and manganese become oxidized and unite into a slag. The molten metal is vigorously stirred by the operator and presently reaches a pasty state so that it can be removed in spongy balls for rolling into muck bars. Squeezing and rolling cause the included slag to be drawn out into thin layers and veins in the iron to produce the fibrous structure of wrought iron. Following this initial rolling the muck bars are heated in piles for rolling again into plate or pipe skelp.

[1] For complete specification, reference may be made to the latest revision of AWWA 7A.3. Copies may be obtained from the American Water Works Association, 500 Fifth Ave., New York 18, N. Y.

Standard definitions of terms relating to wrought iron are given in ASTM Specification A81.

Superficially there is little difference in the appearance of wrought-iron and steel pipe or bolts, but there are several tests which readily identify the materials. By hammering a piece of pipe flat, the fracture can be observed easily. The fracture of wrought iron is ragged, dull gray, and fibrous, whereas that of steel is even, bright and crystalline. A microscopic examination of polished and etched specimens is an infallible test, since the slag inclusions of wrought iron are unmistakable, whereas the steel is almost without slag. Wrought iron, when threads are cut on it, gives a crumbling chip, owing to the fibrous structure, whereas steel gives a long spiral chip. Genuine wrought iron is identified chemically by its low-carbon and -manganese contents. It is generally conceded that freedom from contamination by steel or poor-quality wrought iron can be shown if the manganese content is below 0.07 per cent. A carbon content over 0.10 per cent usually is construed to mean imperfect refining or to indicate that the iron has been diluted with steel scrap.

In addition to the hand-puddling process, wrought iron is made commercially by the Roe process of mechanical puddling and by the Ashton, or Byers, process of adding slag to molten iron from a cupola, following which it is refined in a Bessemer converter. The corrosion resistance of these three types of wrought iron, as determined by 2-year exposure to various mediums in a laboratory did not disclose any significant differences.[1] Typical compositions for these three types of wrought iron are given in the accompanying table. Because of the greater difficulty of manufacture and limited use, wrought iron is sold at a

TYPICAL COMPOSITION OF WROUGHT IRONS
(From "Metals Handbook," 1936 ed.)

Method of manufacture	C	Mn	Si	P	S	Slag by weight, per cent
Hand puddled	0.06	0.045	0.101	0.068	0.009	1.97
Mechanical, Roe	0.08	0.029	0.183	0.115	0.015	2.85
Ashton or Byers, No. 1	0.08	0.015	0.158	0.062	0.010	1.20

[1] "A Comparison of the Corrosion Resistance of Several Wrought-Irons Made by Different Processes," by O. A. Knight and J. R. Benner, *Trans. ASM*, September, 1935, p. 693.

considerable premium over carbon steel. Despite its higher cost, wrought iron is used still to some extent, especially in hot-water lines, underground piping, and plumbing work where it is claimed to be more corrosion-resistant than steel pipe.

The addition of a small amount of copper, usually 0.10 to 0.25 per cent, to either wrought iron or carbon steel has been found to increase greatly the resistance of the metal to atmospheric corrosion when exposed to the weather. Under atmospheric conditions, the beneficial effect of copper is said to be more pronounced than oxides, slag, sulphur, or other foreign matter in iron or steel.[1] The superior resistance to atmospheric corrosion of copper-bearing steel or iron is attributed to the dense, tightly adherent character of the rust coating formed. Tests by the Bureau of Standards and by ASTM Committee A5 on Corrosion of Iron and Steel failed to disclose any marked superiority between carbon steel, wrought iron, or copper-bearing steel or iron in so far as resistance to soil corrosion or liquid immersion was concerned. Under such conditions the characteristics of the soil or liquid were found to have decidedly more effect than the differences in character of these metals.[2]

Minimum tensile properties for commercial wrought-iron bolts are: tensile strength, 45,000 psi; yield point, 25,000 psi; elongation in 8 in., 25 per cent; reduction of area, 40 per cent.

ASTM Standard Specifications for WELDED WROUGHT-IRON PIPE

Serial Designation A72-39 (ASA B36.2)

Abstracted[3]

These specifications cover butt-welded and lap-welded wrought-iron pipe intended for coiling, bending, flanging, and other special purposes. Butt-welded pipe is not intended for flanging and is not recommended for bending or coiling in sizes 1¼ in. or over. Reference to "standard-weight," "extra-strong," and "double-extra-strong" pipe has been omitted in this abstract since these specifications also apply to the schedule wall thicknesses of ASA B36.10 as well as to these former designations.

[1] "Corrosion Causes and Prevention," by F. N. Speller, McGraw-Hill Book Company, Inc., New York, 2d ed., 1936.

[2] Proc. ASTM for 1920 and subsequent years, also "Studies in Corrosion Control," by H. P. Stockwell, Jr., *Jour. AWWA*, Vol. 33, No. 8, p. 1409, August, 1941.

[3] For complete specification, reference may be made to ASTM A72 (see note, p. 369).

The pipe shall be made from all pig-puddled or processed wrought iron which is defined as "a ferrous material, aggregated from a solidifying mass of pasty particles of highly refined metallic iron with which, without subsequent fusion, is incorporated a minutely and uniformly distributed quantity of slag."

Pipe 2 in. or under in mominal size may be butt-welded, unless otherwise specified. All pipe over 2 in. nominal size shall be lap-welded.

The wrought iron shall conform to the following requirements as to chemical composition and physical properties.

CHEMICAL COMPOSITION

Manganese, max, per cent	0.05
Minimum physical properties:	
Tensile strength, psi	40,000
Yield point, psi	24,000
Elongation in 8 in., per cent	12

Hydrostatic Test.—Each length of pipe shall be tested at the mill to specified hydrostatic pressures (Table I of ASTM A120, reproduced on page 370). Pipe 2 in. and larger shall be jarred near one end while under test pressure.

Tension Tests.—One tension test shall be made on one length in each lot of 500 lengths or fraction thereof of each size.

Fracture Tests.—A section of pipe 6 in. long from one length in each lot of 500 lengths or fraction thereof of each size shall be flattened until broken by repeated light blows of a hammer or by pressure. The fracture shall have a fibrous appearance characteristic of wrought iron.

Bend Tests.—A bend test shall be made on a length of pipe from each lot of 500 lengths or fraction thereof of pipe 2 in. or under in diameter. Double-extra-strong pipe over 1¼-in. in nominal size need not be subjected to this test.

Dimensions and Weights.—The standard contains a table of standard weights and nominal dimensions of welded wrought-iron pipe identical with Table II of ASTM A120 for steel pipe (reproduced on page 371), except for slight differences in the nominal wall thicknesses which are well within the manufacturing tolerances permitted for both kinds of pipe. Additional weights and dimensions of wrought-iron pipe conforming to schedule numbers are given in ASA Standard B36.10.

Weight Variations.—The weight of the pipe shall not vary more than 5 per cent for standard-weight (Schedule 40) and extra-strong (Schedule 80) pipe nor more than 10 per cent for double-extra-strong pipe from that specified.

Diameter Variations.—The outside diameter of pipe 1½ in. or under in nominal size shall not be more than $\frac{1}{64}$ in. over nor more than $\frac{1}{32}$ in. under the standard outside diameter. For pipe 2 in. or over in nominal size, the outside diameter shall not be more than 1 per cent over or under the standard outside diameter.

Lengths.—Standard-weight (Schedule 40) threaded wrought-iron pipe shall be furnished in random lengths of 16 to 22 ft, unless otherwise specified. Not more than 5 per cent of the total number of lengths may be "jointers," which are two pieces coupled together. Plain-end pipe may have 5 per cent in lengths of 12 to 16 ft.

Extra-strong (Schedule 80) plain-end wrought-iron pipe shall be furnished in random lengths of 12 to 22 ft with not more than 5 per cent in lengths of 6 to 12 ft.

CAST-IRON PIPE AND FITTINGS

Cast-iron pipe has a relatively long life in underground service owing to its heavy wall and inherently good resistance to internal, as well as soil, corrosion. Hence it is used extensively for water and gas distribution systems in cities and towns, particularly under paved streets where it is important to get a material having a long life in order to avoid tearing up and repairing pavements to fix leaks or renew pipe. Bell-and-spigot pipe and fittings with the AWWA water bell or the AGA gas bell comprise the bulk of the underground cast-iron pipe business, but with mechanical bell joints finding increasing use, especially in distribution systems for natural gas and dry manufactured gas. Universal joint pipe is used to some extent in underground water systems.

Flanged cast-iron pipe and mechanical bell-joint pipe of various types are used to a considerable extent above ground in gas works and oil refineries, and in process industries for carrying certain fluids where cast iron will withstand corrosion better than steel. Cast-iron pipe is made also to the outside diameter of wrought pipe for making up with American Standard taper pipe threads and screwed couplings, and it can be obtained with plain ends for compression couplings of the Dayton or Dresser type, or with grooved ends for Victaulic couplings.

Extensive changes have taken place in the cast-iron pipe industry within the past few years through the introduction of improved manufacturing methods, mechanical (stuffing-box) joints, and a new system of ratings which is based on earth loadings as well as on fluid pressure. These developments are reviewed briefly in succeeding paragraphs.

Manufacturing Methods.—*Cast-iron pipe* is manufactured by four distinct processes: (1) pit-cast vertically in dry-sand molds, (2) horizontally cast in green-sand molds, (3) centrifugally cast in sand-lined molds, and (4) centrifugally cast in permanent metal molds. At the present time over 75 per cent of all the pipe manufactured is produced by either the centrifugal or horizontally cast process. The pipe is available with bell-and-spigot joints ordinarily used for underground water-supply installations, with flanged joints for construction work above ground, and with mechanical (gland type) joints for underground gas construction as well as for waterworks and industrial installations.

The pit-cast process consists essentially in forming a mold by ramming sand around a pattern, drying this mold in an oven,

inserting a previously made core, and pouring the iron in the space between the core and the mold. After the iron is cooled and the core bar removed, the pipe is shaken out of the flask, cleaned, dipped in a tar bath, subjected to an internal test pressure, and is then ready for shipping. Pit-cast pipe for water-supply systems usually is ordered to American Standard Specifications for Cast Iron Pit Cast Pipe for Water or Other Liquids (ASA 21.2),[1] to ASTM Specification A44, or to Federal Specifications WW-P-421. In the gas industry pit-cast pipe is ordered to AGA Standard Specifications for Cast Iron Pipe and Special Castings. Pit-cast culvert pipe can be bought to ASTM Designation A-142. These specifications are abstracted in succeeding pages.

In the horizontal method of making cast-iron pipe, molds are placed on ramming machines filled with properly prepared sand and mechanically rammed. Cores are made by building up tempered sand around the core bar which is perforated to allow for the escape of steam and gas. The cores are set, gauged, and located in the proper position in the mold and clamped into position to prevent movement. The top of the mold is then set in place and metal poured from a multiple-lipped ladle that is designed to draw the iron from the bottom. In this manner the introduction of impurities into the mold is eliminated and the molten iron travels a minimum distance in the mold. After the pipes are cooled, they are cleaned, dipped, and tested. Horizontally cast pipe usually is ordered to Federal Specifications WW-P-421.

In making pipe by the centrifugal process in sand-lined molds, a mold is made by ramming sand about the proper sized pattern. This mold is then placed horizontally in a centrifugal casting machine and, while it is revolving, the exact quantity of metal required to make the pipe is introduced. The speed of rotation sets up sufficient centrifugal force to distribute the molten metal on the wall of the mold, producing a completely formed pipe within a few seconds. The spinning liquid metal solidifies gradually—in from 2 to 10 min. Within 30 to 60 min the pipe is removed from the flask, the spinning and cooling time being determined by the diameter and thickness of the pipe.

In making centrifugally cast pipe in permanent molds, the molten metal is poured into a rapidly rotating water-cooled metal mold to the surface of which a thin pulverulent refractory coating is applied to control the cooling of the iron. A sand core is used

[1] This standard also is reproduced as ASTM Specification A-44.

for forming the inside contour of the bell. The mold is mounted on rollers in a water jacket so that it can be rotated at high speeds, and the water jacket is on wheels so that the entire assembly can be moved by a hydraulic cylinder in the direction of the axis of the mold. The molten iron is fed into the mold through a trough that has a spout curved toward a side wall. The tube used to spray the pulverulent material is built into the trough with the nozzle set just behind the pouring spout, and iron is poured from a tilting ladle at a uniform rate. As the assembly moves over the spout, the iron is distributed from one end of the mold to the other and is held against the inner wall by centrifugal force. After completion of the pour, the mold is kept rotating until the pipe has cooled below the critical temperature. It is then removed and placed in a thermostatically controlled heat-treating furnace and the temperature gradually lowered. After leaving the furnace the pipe is cleaned, dipped, and hydraulically tested.

Pending formulation of an American Standard for this product, centrifugally cast pipe made in either sand-lined or permanent molds usually is ordered to Federal Specification WW-P-221. Centrifugally cast culvert pipe can be ordered to ASTM Designation A-142.

Cast-iron fittings are made by conventional foundry methods for a variety of joints including bell-and-spigot, flanged, and mechanical (gland type), or other special proprietary designs. Bell-and-spigot fittings usually are bought to the AGA or AWWA specifications abstracted on pages 437 to 443. Flanged fittings may be bought to the AGA specifications or to the usual American Standards for flanged cast-iron fittings abstracted on pages 592 to 624. Mechanical-joint fittings may be ordered to AGA or AWWA specifications where applicable, or bought according to usual manufacturing practice.

Coating and Lining.—The usual commercial practices for coating and lining cast-iron pipe and fittings are described on pages 1270 to 1271.

Nominal Sizes.—According to trade practice the nominal sizes of cast-iron pipe and fittings are designated according to *inside* diameter. Where a cement lining is used, however, allowance should be made for the fact that the lining encroaches on the nominal inside diameter of the pipe.

Types of Joints.—The principal types of joints for cast-iron pipe and fittings are described in succeeding paragraphs with a brief

account of where each is commonly used. In some cases illustrations are given, and in others page numbers are referenced where illustrations are available elsewhere in this handbook. For further information reference should be made to manufacturers' catalogues. In the absence of national standards for the various types of mechanical joints, it is necessary to buy to the proprietary designs of competing manufacturers.

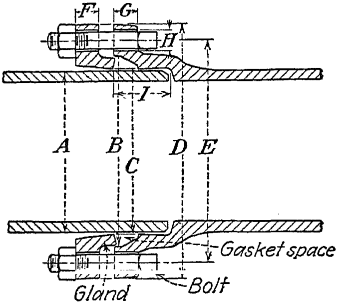
Bell-and-spigot Joint.—This is the old conventional underground joint developed in 1785 which is made up with lead and oakum, sulphur compounds, or cement (see pages 451 to 455). Lead and oakum constitute the prevailing joint for water-supply systems, and oakum with lead or cement backing is used to a considerable extent for the low-pressure distribution of wet manufactured gas (see pages 1192 to 1196). In order to have such joints hold tight, however, it is necessary to keep the oakum wet, either with moisture from the fluid conveyed or, in the case of dry gas systems, by introducing an impregnating compound into the lines. Whereas bell-and-spigot joints have proved satisfactory in water-supply systems operating at pressures of 200 psi or more, they are not regarded with favor for gas pressures in excess of 10 psi. Regular cast-iron pipe, as well as soil pipe, usually has bell-and-spigot joints where used in sewer systems, although threaded cast-iron pipe is used to some extent with screwed drainage fittings for waste and vent lines of plumbing systems inside buildings.

The AWWA and AGA each have their own designs of bell-and-spigot joints which differ principally in the grooving of the bells (compare the illustrations connected with Tables XV and XIX). When so ordered bell-and-spigot pipe and fittings for fire lines and other high-pressure water service can be furnished with bolting lugs as noted on page 438 and 441.

Mechanical (Gland-type) Joint.—This modification of the bell-and-spigot joint has come into common use for low- and intermediate-pressure gas distribution systems, particularly those conveying natural gas or dry manufactured gas. Mechanical joints are used also for water lines where excessive vibration may be expected, and for water, gas, sewage, and process piping above ground within plants. In the mechanical (gland-type) joint shown in Fig. 3 the lead and oakum of the conventional bell-and-spigot joint are supplanted by a stuffing box in which a rubber or composition packing ring, with or without a metal or canvas tip or canvas backing, is compressed by a cast-iron follower ring drawn

up with bolts. In addition to making an inherently tight joint even under considerable pressure, this arrangement possesses the further advantage of permitting relatively large lateral deflections (3½ to 7 deg) as well as longitudinal expansion or contraction. Manufacturers' standard dimensions for mechanical-joint cast-iron pipe are given in the accompanying table.

MANUFACTURERS' STANDARD DIMENSIONS FOR MECHANICAL-JOINT CAST-IRON PIPE[1]



(All dimensions in inches)

Nominal pipe size	A	B	C	D	E	F	G	H	I	Bolts		
										No.	Size	Length
3	3.96	4.94	4.06	7.50	6.19	0.63	0.94	¾	2.50	4	⅝	3
4	4.80	6.02	4.90	8.88	7.50	0.75	1.00	⅞	2.50	4	¾	3½
6	6.90	8.12	7.00	10.88	9.50	0.87	1.06	⅞	2.50	6	¾	3½
8	9.05	10.27	9.15	13.13	11.75	1.00	1.13	⅞	2.50	6	¾	4
10	11.10	12.34	11.20	15.50	14.00	1.00	1.19	⅞	2.50	8	¾	4
12	13.20	14.44	13.30	17.75	16.25	1.00	1.25	⅞	2.50	8	¾	4
14	15.30	16.54	15.44	20.25	18.75	1.25	1.31	⅞	3.50	10	¾	4
16	17.40	18.64	17.54	22.50	21.00	1.31	1.38	⅞	3.50	12	¾	4½
18	19.50	20.74	19.64	24.75	23.25	1.38	1.44	⅞	3.50	12	¾	4½
20	21.60	22.84	21.74	27.00	25.50	1.44	1.50	⅞	3.50	14	¾	4½
24	25.80	27.04	25.94	31.50	30.00	1.56	1.63	⅞	3.50	16	¾	5
30	32.00	33.46	32.17	38.75	36.50	2.00	1.81	1⅛	4.00	20	1	6
36	38.30	39.76	38.47	45.50	43.25	2.00	2.00	1⅛	4.00	24	1	6
42	44.50	45.96	44.67	52.50	50.00	2.00	2.00	1⅜	4.00	28	1¼	6
48	50.80	52.26	50.97	59.00	56.50	2.00	2.00	1⅜	4.00	32	1¼	6

[1] Data furnished by Thomas F. Wolfe of the Cast Iron Pipe Research Assoc., April, 1944.

Mechanical (Lock-type) Joint.—For installations where the joints may tend to come apart owing to sag or lateral thrust in the

pipe line, a mechanical joint having a self-locking feature is used to resist end pull. This joint is similar to the gland-type mechanical joint except that in the locked joint the spigot end of the pipe is grooved, or has a recess, to grip the gasket. Although only slight expansion or contraction can be accommodated in this type of joint, it does allow the usual $3\frac{1}{2}$ to 7 deg angular deflection. The lock-type joint finds application above ground in the process industries, and in river crossings on bridges or trestles, as well as in submarine crossings, or in unusually loose or marshy soils. Where the locking feature is on the spigot rather than the bell, this type of pipe can be used with the regular line of mechanical-joint fittings.

Mechanical (Roll-on Type) Joint.—Where a low-cost mechanical joint is desired the roll-on type can be used. In this joint a round rubber gasket is placed over the spigot end which is pulled into the bell by mechanical means, thus rolling the ring into place in the bottom of the bell. Outboard of the rubber gasket, braided jute is wedged behind a projecting ridge in the bell which serves to confine the gasket under pressure in the joint. A bituminous compound is used to seal the mouth of the bell and to aid in retaining the hemp and rubber gasket. In addition, when pipe is laid in cold climates where electrical thawing of mains and services is sometimes necessary, a cold lead strip about $\frac{1}{4}$ in. sq can be calked in between the hemp and bitumastic to provide an electrical circuit through the joint. Either bell-and-spigot or mechanical (gland-type) fittings are used with this line of pipe.

Mechanical (Screw-gland Type) Joint.—This type of mechanical joint for cast-iron pipe (see Fig. 10*b*, page 1196) makes use of a coarse-threaded screw gland drawn up by means of a spanner wrench to compress a standard rubber or composition packing gasket. The joint allows from 2 to 7 deg angular deflection as well as expansion or contraction without danger of leaks. A lead ring inserted in the bell ahead of the gasket seals off the contents of the line from the gasket and at the same time provides an electrical circuit through the joint for thawing out frozen underground mains and service lines by the electrical method. The screw-gland joint is suitable for use with water, gas, oil, and other fluids at considerable pressure. The gaskets and lead rings are interchangeable with those used in equivalent lines of mechanical joints of the bolted-gland type. A full line of fittings is available for use with screw-gland pipe.

Ball-and-socket Joints.—For river crossings, submarine lines, or other places where great flexibility is necessary, cast-iron pipe can be obtained with ball-and-socket joints of either the lead-and-oakum or mechanical-gland types. The latter is shown in Fig. 4. Provision is made for longitudinal expansion and contraction in at least one mechanical-joint type, and a positive stop against disengagement of the joint is a feature of all designs. Up to 15 deg angular deflection can be accommodated without leakage. This pipe is heavy enough to remain under water where laid without requiring river clamps or anchorage devices. The pipe may be pulled across streams with a cable, as the joints are positively locked against separating, or it may be laid direct from a barge, bridge, or pontoons without the service of a diver. The mechanical ball-and-socket joint is suitable for use with water, sewage, air, gas, oil, and other fluids at considerable pressure. Either bell-and-spigot or mechanical (gland-type) fittings can be used with this line of pipe, although the integral ball present on the spigot end of some designs has to be cut off before the pipe can be inserted in a regular bell.

Universal Pipe Joint.—This type of cast-iron pipe joint, shown in Fig. 5, has a machined taper seat which obviates the need for calking or a compression gasket. The joint is pulled up snugly with two bolts, after which the nuts are backed off slightly thus enabling the lock washers to give enough to avoid overstressing the socket or lugs. Pipe is made in 6- and 12-ft lengths to the usual pressure classes and can be bought as Type IV under Federal Specification WW-P-421 (see page 443). Universal-joint fittings are available for use with the pipe. This type of joint is used to some extent in pipe diameters of 4 to 24 in. for underground water-supply systems, but it is not considered suitable for gas service and it does not permit much angular displacement or expansive movement.

Compression Sleeve Coupling.—The type of joint shown in Fig. 6 is used with plain-end pipe of either cast iron or steel. It is well known under the trade names of Dresser Coupling or Dayton Coupling and is used extensively for air, gas, oil, water, and other services above- or underground. With a joint of this type it is necessary to anchor or brace solidly at dead ends or turns to prevent the line from pulling apart. Compression couplings and fittings with screwed packing glands are available for use with small-size cast-iron or steel pipe (see Fig. 2, page 1178). In welded

transmission lines for oil or gas where any significant change in temperature is expected, a certain percentage of the joints may be made up with compression couplings instead of welding in order to take care of expansion.

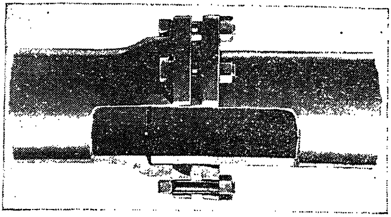


FIG. 3.—Mechanical (gland-type) joint for cast-iron pipe.

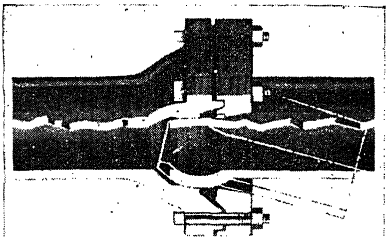


FIG. 4.—Ball-and-socket mechanical joint for cast iron pipe.

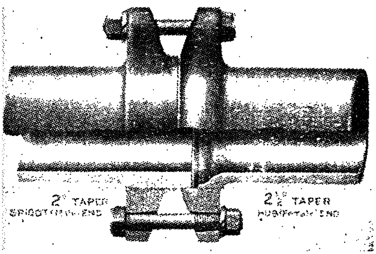


FIG. 5.—Universal cast-iron pipe joint.

Victaulic Coupling.—The type of split coupling shown in Fig. 7 is used with either cast-iron or steel pipe having grooves near the ends which enable the coupling to grip the pipe in order to prevent disengagement of the joint. Victaulic couplings come in two

segments in sizes from ¾ to 14 in. and in four or six segments in larger sizes. Grooved-end fittings are available to go with the couplings. With proper choice of gasket material, the Victaulic joint is suitable for use above- or underground with nearly any fluid or gas. Among its advantages are provision for considerable angular displacement and some expansive travel; pressure action in a direction that tends to seat the gasket; a positive stop against disengagement of the joint; and a design suited to rapid erection

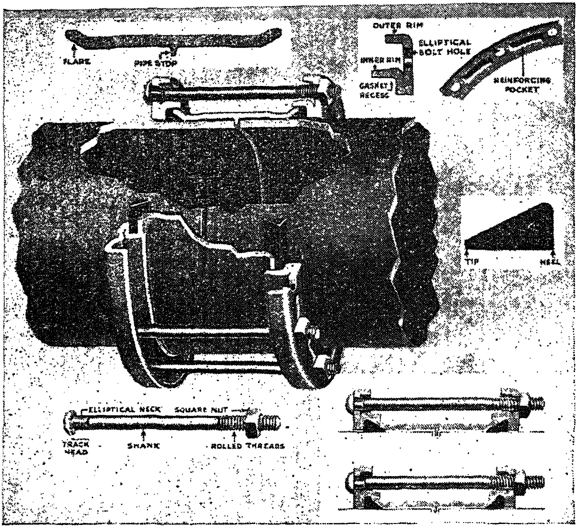


FIG. 6.—Compression sleeve (Dresser) coupling for plain-end cast-iron or steel pipe.

or tearing down of a line as might be desired for temporary construction purposes. One of the disadvantages is the necessity for grooving the pipe ends nearly as deep as taper threads. Where this is objectionable and in the case of light-weight pipe, the ends can be upset, banded, or have heavier grooved nipples attached by welding.

Flange-type Joint.—Flanged cast-iron pipe is used aboveground for low and intermediate pressures in water-pumping stations, gas works, power and industrial plants, oil refineries, and booster stations on water, gas, and oil transmission lines. Flanges usually

are faced and drilled according to the American Standard for Class 125 (steam) Cast-iron Pipe Flanges and Flanged Fittings ASA-B16a which is used with both Classes 150 and 250 pipe, although Classes 25 and 250 (steam) flanges are furnished where required. There is an AGA standard for flanged cast-iron pipe and fittings (see pages 441 to 443) which has been largely supplanted by Class 125 (steam) of ASA-B16a (see pages 597 to 616). Cast-iron pipe is made both with integrally cast flanges and with threaded companion flanges for screwing onto the pipe. In the latter case the outside diameter of the pipe conforms to "iron-pipe-size" (IPS) dimensions on account of the threads.

Screwed-type Joint.—Small-size cast-iron pipe also is made to "iron-pipe-size" (IPS) dimensions for use with screwed couplings or fittings similar to those used with threaded steel pipe. Pressure pipe for water, gas, steam, and other services is supplied in standard, strong, and extra-strong weights and made to meet the requirements of Federal Specifications WW-P-421. A lightweight pipe for drain lines inside buildings and elsewhere is available which conforms to American Standard Cast-iron Pipe for Drainage, Vent, and Waste Services, ASA A40.5, abstracted on page 456. It is used with American Standard Cast-iron Screwed Drainage Fittings, ASA B16.12 (see page 457), or with other screwed fittings suiting particular applications.

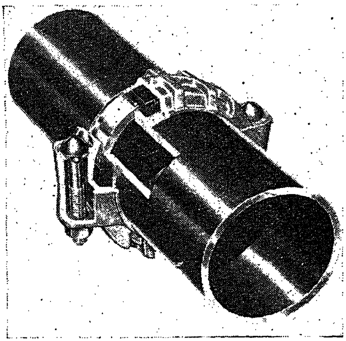


FIG. 7.—Victaulic coupling for grooved-end cast-iron or steel pipe.

American Standards for Cast-iron Pipe.—Since 1926 ASA Sectional Committee A21 has been engaged in unifying specifications for cast-iron pipe, including materials, types of joints, dimensions, pressure ratings, and manufacturing methods in so far as necessary to eliminate little-used sizes and varieties and to develop a coordinated scheme of pipe and fittings applicable to all common mediums. This program has been sponsored by the American Gas Association, the American Society for Testing Materials, the American Water Works Association, and the New England Water

Works Association. Up to the close of 1944 the following American Standard Specifications, Recommended Practices, and Bulletins have been issued under the auspices of Sectional Committee A21:

ASA—A21.1-1939 Manual for the Computation of Strength and Thickness of Cast-iron Pipe.

ASA—A21.2-1939 Specifications for Cast-iron Pit Cast Pipe for Water or Other Liquids. (ASTM Specification A44 is identical.)

ASA—A21.4-1939 Specifications for Cement Mortar Lining for Cast-iron Pipe and Fittings.

Iowa State College, *Bull.* 146 (1940), Supporting Strengths of Cast-iron Pipe for Water and Gas Service.

The foregoing design methods and specifications have been adopted in their particular fields where they have largely supplanted former practices.

Committee A21 has in preparation specifications on Cast Iron Pit Cast Pipe for Gas; on pipes other than pit-cast, including pipe cast centrifugally; on Cast Iron Bell and Spigot Fittings; on Cast Iron Flanged Pipe and Fittings; and on Coal Tar Dip Coating for Cast Iron Pipe and Fittings. Until these specifications are issued, the corresponding AGA, AWWA, and Federal specifications will continue in use in their respective fields.

The following abstracts of available American Standard recommended practices and specifications will serve for limited reference and to indicate the nature of their contents.

American Recommended Practice

MANUAL FOR THE COMPUTATION OF STRENGTH AND THICKNESS OF CAST-IRON PIPE

ASA A21.1-1939

Abstracted[1]

This recommended practice covers a new method of design for computing pipe wall thickness which takes into account not only static pressure and water hammer, but also earth pressure and load from trucks for several different

[1] This abstract is based on ASA A21.1-1939. Where complete standards are required, or revisions of some later date, reference may be made to the publications of the ASA. Printed copies can be purchased by addressing American Standards Association, 29 West 39th St., New York 18, N.Y.

methods of laying. Four common and two uncommon field conditions encountered in laying cast-iron pipe were selected as meeting conditions to be expected in trenches having approximately vertical sides:

Field Condition	Explanation
A	Flat bottom trench, backfilling not tamped
B	Flat bottom trench, backfilling tamped
C	Pipe supported on blocks, backfilling not tamped
D	Pipe supported on blocks, backfilling tamped
E	Bottom of trench shaped to fit bottom of pipe for about 90 deg (unevennesses filled in by sand as required), backfilling not tamped
F	Same as E except that backfilling is tamped

Field conditions E and F are common for sewer pipe but have been rarely used for cast-iron water and gas pipe though the superior strength of pipe so laid has caused some increase in the use of this practice for pipe lines. Another condition, *viz.*, pipe under embankment, will be treated in a future appendix.

In this method of design, the stress due to fluid pressure is computed by the "common formula for bursting pressure" (see page 33), to which may be added a stress corresponding to the conventional allowance for water hammer (see page 47). Earth loads for the various field conditions are computed by mathematical formulas developed at Iowa State College and based on full-scale tests extending over a period of more than 25 years. The fluid pressure stress and earth load are then combined according to the following assumptions so as to give the total stress which a given wall thickness can support, and a factor of safety of 2.5 used in design.

Cast-iron or other pipes laid in the ground are acted on by one or more of the following forces:

1. Internal static pressure of the liquid or gas which is contained in the pipe.
2. Water hammer (or hammer from other liquid).
3. Load from the backfill. (See page 1063 for computing Earth Loads on Pipe in Trenches.)
4. Load and impact from passing trucks or other vehicles.

Loads 2 and 4 are occasional and transitory. In the specifications of Committee A21 they are not considered as acting simultaneously but both are considered separately as loads added to static pressure and backfill load, and whichever of the two gives the greater computed thickness of pipe, when taken together with the normal loads 1 and 3, is controlling in the determination of pipe wall thickness for the case under consideration. Pipes for gas are free from water hammer caused by liquids and are hence always computed with truck load and its impact added to the load from backfill.

In addition to presenting formulas for this method of design, the A21.1 Manual contains many charts and pipe wall thickness tables computed therefrom. Since space limitations do not permit reproducing this material here, those having occasion to solve such problems should refer to the manual. Attention is called to the fact that tables of allowable working pressures under the several conditions of earth loading for commercially available thicknesses of pipe also are furnished in American Standard Specification A21.2 and in manufacturers' catalogues.

TABLE XVI.—STANDARD THICKNESSES AND WEIGHTS OF CAST-IRON PIT-CAST PIPE
(Table 3 of ASA A21.2-1939)

NOTE.—These weights are for pipe laid without blocks, on flat-bottom trench, with tamped backfill, under 5 ft of cover. For other conditions, see ASA A21.1 and A21.2.

Nominal inside diameter, in.	Class 50, 50 lb pressure, 115 ft head			Class 100, 100 lb pressure, 231 ft head			Class 150, 150 lb pressure, 346 ft head			Class 200, 200 lb pressure, 462 ft head		
	Thickness, in.	Weight based on 12-ft length[1]		Thickness, in.	Weight based on 12-ft length[1]		Thickness, in.	Weight based on 12-ft length[1]		Thickness, in.	Weight based on 12-ft length[1]	
		Average per ft	Per length		Average per ft	Per length		Average per ft	Per length		Average per ft	Per length
3	0.37	14.2	170	0.37	14.2	170	0.37	14.2	170	0.37	14.2	170
4	0.40	19.2	230	0.40	19.2	230	0.40	19.2	230	0.40	19.2	230
6	0.43	30.0	360	0.43	30.0	360	0.43	30.0	360	0.43	30.0	360
8	0.46	42.9	515	0.46	42.9	515	0.46	42.9	515	0.46	42.9	515
10	0.50	57.1	685	0.50	57.1	685	0.54	60.8	730	0.58	64.6	775
12	0.54	73.3	880	0.54	73.3	880	0.58	77.9	935	0.63	83.8	1,005
14	0.54	85.4	1,025	0.58	91.3	1,095	0.63	100.8	1,210	0.68	107.9	1,295
16	0.58	105.4	1,265	0.63	113.3	1,360	0.68	125.0	1,500	0.79	142.5	1,710
18	0.63	127.9	1,535	0.68	136.7	1,640	0.73	150.4	1,805	0.85	172.1	2,065
20	0.66	148.8	1,785	0.71	158.8	1,905	0.83	188.8	2,265	0.90	202.5	2,430
24	0.74	198.8	2,385	0.80	212.9	2,555	0.93	252.5	3,030	1.00	269.2	3,230
30	0.87	288.3	3,460	0.94	311.3	3,735	1.10	367.1	4,405	1.19	402.9	4,835
36	0.97	384.2	4,610	1.05	420.8	5,050	1.22	491.3	5,895	1.43	578.3	6,940
42	1.07	497.5	5,970	1.25	579.2	6,950	1.35	637.9	7,655	1.58	749.2	8,990
48	1.18	625.8	7,510	1.37	726.3	8,715	1.48	799.6	9,595	1.73	940.0	11,280
54	1.30	777.1	9,325	1.51	906.3	10,875	1.63	997.1	11,965	1.90	1,168.8	14,025
60	1.39	922.5	11,070	1.62	1,077.1	12,925	1.89	1,270.9	15,250	2.20	1,488.3	17,860

[1] Including bell-and-spigot bead. Calculated weight of pipe rounded off to nearest 5 lb.

TABLE XVI.—(*Concluded*)

Nominal inside diameter, in.	Class 250, 250 lb pressure, 577 ft head			Class 300, 300 lb pressure, 693 ft head			Class 350, 350 lb pressure, 808 ft head		
	Thickness, in.	Weight based on 12-ft length[1]		Thickness, in.	Weight based on 12-ft length[1]		Thickness, in.	Weight based on 12-ft length[1]	
		Average per ft	Per length		Average per ft	Per length		Average per ft	Per length
3	0.37	14.2	170	0.37	14.2	170	0.37	14.2	170
4	0.40	19.2	230	0.40	19.2	230	0.40	19.2	230
6	0.43	30.0	360	0.46	31.7	380	0.50	34.2	410
8	0.50	45.8	550	0.54	49.2	590	0.58	53.8	645
10	0.63	72.1	865	0.68	77.1	925	0.73	81.7	980
12	0.68	92.1	1,105	0.73	97.9	1,175	0.79	105.0	1,260
14	0.79	123.3	1,480	0.85	131.7	1,580	0.92	148.3	1,780
16	0.85	152.1	1,825	0.92	162.9	1,955	0.99	181.7	2,180
18	0.92	184.2	2,210	0.99	196.7	2,360	1.07	220.8	2,650
20	0.97	216.7	2,600	1.05	232.1	2,785	1.22	277.1	3,325
24	1.08	288.3	3,460	1.26	346.2	4,155	1.36	370.0	4,440
30	1.39	462.1	5,545	1.50	511.3	6,135	1.62	557.9	6,695
36	1.54	617.1	7,405	1.79	727.9	8,735	1.93	794.2	9,530
42	1.71	802.9	9,635						
48	2.02	1,077.1	12,925						
54	2.21	1,333.8	16,005						
60	2.38	1,594.6	19,135						

[1] Including bell-and-spigot bead. Calculated weight of pipe rounded off to nearest 5 lb.

American Standard Specification for CAST-IRON PIT-CAST PIPE FOR WATER OR OTHER LIQUIDS

ASA A21.2-1939

Abstracted[1]

The pipe shall be pit-cast with bell-and-spigot ends, plain ends, or such other type of ends as may be agreed on at the time of purchase. Pipe with bell-and-spigot ends shall conform accurately to the dimensions given in Table XV and shall be at least 12-ft nominal laying length, except where cut pipe is permitted. The iron shall be remelted for casting in a cupola, or other suitable furnace, and shall not contain more than 0.90 per cent phosphorus nor more than 0.10 per cent sulphur.

Marking.—Each pipe shall have distinctly cast upon it the initials of the maker's name. When cast especially to order, each pipe larger than 4 in. also may have certain other identification marks cast upon it. The weight and class shall be conspicuously painted in white in the inside or outside of each pipe after the coating has become hard.

Tolerances.—Tolerances or maximum permitted variations from standard dimensions for pipe barrels, spigot beads, and sockets are:

Nominal diameter, in.	Tolerance on diameter, ± in.	Tolerance on thickness, ± in.
3 to 8	0.06	0.07
10 to 16	0.06	0.08
18, 20, and 24	0.08	0.08
30, 36, and 42	0.10	0.10
48	0.12	0.10
54 and 60	0.15	0.10

In pipe-barrel thickness, tolerances 0.02 in. greater than those listed above shall be permissible over areas not exceeding 8 in. in length in any direction.

Hydrostatic Test.—Each pipe shall be subjected to a hydrostatic proof test made either before or after the tar dip or priming coat, but before the cement mortar or other special lining has been applied. Depending on wall thickness and diameter, test pressures range from 300 to 450 psi for sizes 3 to 18 in., and from 250 to 450 psi for sizes 20 to 60 in. inclusive. Pipe shall be under the test pressure for at least ½ min and shall be subjected to a hammer or shock test without leaking or sweating.

Tests of Material.—The physical characteristics of the iron shall be determined from test bars from the same iron as the pipe or, if specified by the purchaser, by the testing of Talbot strips and/or rings cut from the pipe. Test bars 2 in. wide, 1 in. thick, and not less than 26 in. long shall be cast vertically, using a small heated ladle taking its metal from the main ladle from which the pipe is

[1] For complete requirements, reference may be made to ASA Standard A21.2 (see note, p. 430). See also ASTM Designation A-44 which is identical.

poured. The test bars shall be broken as beams by placing them flatwise on supports 24 in. apart and applying the load at the center of the span. The bars shall be measured at the point of application of the load and results corrected to a standard 2- by 1-in. cross section by the conventional beam formula. Corrected results shall meet the following requirements:

MINIMUM CORRECTED BREAKING LOADS AND DEFLECTIONS

Metal thickness of pipe, in.	Minimum center breaking load, lb	Minimum center deflection at breaking, in.
Below 0.61	1,900	0.30 + 0.0001(breaking load − 1,900)
0.61 to 0.90	2,000	0.30 + 0.0001(breaking load − 2,000)
0.91 to 1.60	2,200	0.30 + 0.0001(breaking load − 2,200)
1.61 to 2.50	2,300	0.30 + 0.0001(breaking load − 2,300)

Standard Thickness and Weights.—A considerable choice of wall thicknesses is provided in another table (not reproduced here) for each nominal diameter in order to look after various combinations of fluid pressure and earth loading. Thicknesses corresponding to field condition *B* where pipe is laid without blocking in a flat-bottom trench with tamped backfill and 5 ft of cover have been classified according to pressure in Table XVI.

AWWA Standard Specifications for SPECIAL CASTINGS

Abstracted[1]

The AWWA Standard Specifications for Special Castings were adopted in 1908 as part of the AWWA Standard Specifications for Cast Iron Water Pipes and Special Castings. These specifications in so far as they relate to pipe have been superseded by the American Standard Specifications for Cast Iron Pit Cast Pipe for Water or Other Liquids, ASA A21.2. Those portions of the 1908 AWWA specifications relating to special castings remain in force pending adoption of an American Standard for bell-and-spigot fittings.

Designations and Dimensions.—The method of designating the sizes and outlets of fittings is shown in Fig. 8. Laying dimensions for common fittings are shown in Tables XVII and XVIII. (For other fittings, see AWWA Standard.) Bell-and-spigot dimensions are approximately the same as those shown in Table XV.

Ratings and Test Pressures.—Classes and hydrostatic test pressures for AWWA fittings are as follows:

Class	Working pressure		Hydrostatic test pressure, psi	
	Ft head	Psi	Under 20 in. diameter	20 in. diameter and over
A	100	43	300	150
B	200	86	300	200
C	300	130	300	250
D	400	173	300	300
E	500	217	500[1]	500[1]
F	600	260	500[1]	500[1]
G	700	304	500[1]	500[1]
H	800	347	500[1]	500[1]

[1] Not specified in standard, but manufacturers' practice at an extra charge when required.

[1] For complete requirements, reference should be made to the AWWA standard. Copies may be obtained from the American Water Works Association, 500 Fifth Ave., New York 18, N. Y.

For Classes A, B, C, and D there is one line of fittings in sizes 3 to 12 in. inclusive made from Class D patterns; two lines of fittings in sizes 14 to 24 in. inclusive made from Class B and D patterns; and four lines of fittings in sizes 30 to 60 in. inclusive made to patterns of the four respective classes. For Classes E, F, G, and H (for fire lines and other high-pressure service) one line of

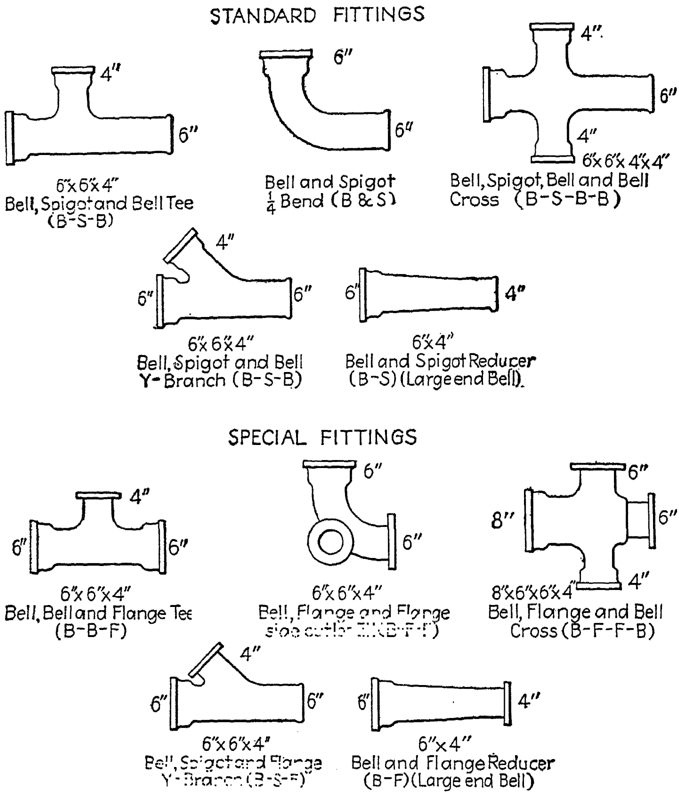
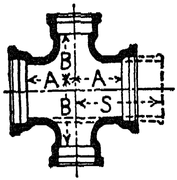


FIG. 8.—Method of designating size of AWWA and AGA fittings.

fittings is furnished for Classes E and F and another line for Classes G and H in sizes from 6 to 24 in. inclusive, and a separate line of fittings for each class in sizes 30 to 36 in. Bells and spigots for high-pressure service can be obtained with double instead of single grooves if desired.

When so ordered, bell-and-spigot pipe is furnished with bolting lugs. Lugs for Classes A to D are called "standard lugs." Lugs for Classes E to H are heavier

Table XVII.—Laying Dimensions of AWWA Standard Bell-and-spigot Tees and Crosses
(All classes)



Tee and Cross

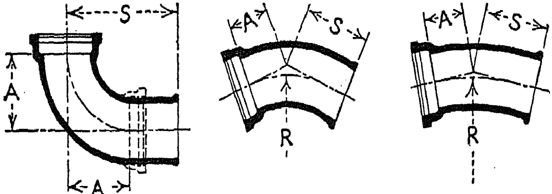
(Dimensions in inches)

Size	*A*	*B*	*S*	Size	*A*	*B*	*S*	Size	*A*	*B*	*S*
3	10	10	22	36x 8	14	27	26	48x16	19	35	31
4	11	11	23	36x10	15	27	27	48x18	20	35	34
6	12	12	24	36x12	16	27	28	48x20	21	35	36
8	13	13	25	36x14	18	29	30	48x24	23	35	38
10	14	14	26	36x16	19	29	31	48x30	26	35	43
12	15	15	27	36x18	20	29	34	48x36	29	35	46
14	16	16	28	36x20	21	29	36	48x42	32	35	49
16	17	17	29	36x24	23	29	38	48x48	35	35	52
18	18	18	30	36x30	26	29	43	54x20	28	38.5	46
20	19	19	31	36x36	29	29	46	54x24	30	40	48
24	21	21	33	42x12	16	30	28	54x30	33	40	51
30x 6	13	24	25	42x14	18	32	30	54x36	36	42	54
30x 8	14	24	26	42x16	19	32	31	54x42	39	42	57
30x10	15	24	27	42x18	20	32	34	54x48	42	45	60
30x12	15	24	27	42x20	21	32	36	54x54	45	45	63
30x14	18	26	30	42x24	23	32	38	60x20	28	42	46
30x16	19	26	31	42x30	26	32	43	60x24	30	44	48
30x18	20	26	34	42x36	29	32	46	60x30	33	44	51
30x20	21	26	36	42x42	32	32	49	60x36	36	44	54
30x24	23	26	38	48x12	17	33	29	60x42	39	48	57
30x30	26	26	43	48x14	18	35	30	60x48	42	48	60
								60x54	45	48	63
								60x60	48	48	66

Reducing tees and crosses in sizes up to and including 24 in. have same laying dimensions as straight sizes.

Large diameter tees and crosses furnished with ribs as required.
For weights and metal thicknesses, see AWWA standards.

Table XVIII.—Laying Dimensions of AWWA Standard Bell and Spigot Bends
(All classes)



¼ Bend (90°) ⅛ Bend (45°) 1/16 Bend (22½°)

(Dimensions in inches)

Size	90-deg bend (¼)		45-deg bend (⅛)			22½-deg bend (1/16)		
	A	*S*	*A*	*S*	*R*	*A*	*S*	*R*
3	16	24	9.94	15.94	24	9.55	15.55	48
4	16	24	9.94	15.94	24	9.55	15.55	48
6	16	24	9.94	15.94	24	9.55	15.55	48
8	16	26	9.94	15.94	24	9.55	15.55	48
10	16	28	9.94	15.94	24	9.55	15.55	48
12	16	28	9.94	15.94	24	9.55	15.55	48
14	18	30	14.91	20.91	36	14.32	14.32	72
16	24	36	14.91	20.91	36	14.32	14.32	72
18	24	36	14.91	20.91	36	14.32	14.32	72
20	24	36	19.88	25.88	48	19.10	19.10	96
24	30	42	24.85	30.85	60	23.87	23.87	120
30	36	48	24.85	30.85	60	23.87	23.87	120
36	48	60	37.28	37.28	90	35.80	35.80	180
42	48	60	37.28	37.28	90	35.80	35.80	180
48	54	66	37.28	37.28	90	35.80	35.80	180
54	..	..	37.28	37.28	90	35.80	35.80	180
60	..	..	37.28	37.28	90	35.80	35.80	180

Bell dimensions same as pipe, Table XV.
Fittings furnished without lugs unless otherwise specified.
For weights and metal thicknesses, see AWWA standards.
One thirty-second and one sixty-fourth bends are also standard; for dimensions, see AWWA standards.

and are described as "lugs for high-pressure bell-and-spigot pipe and fittings." Two lugs are placed on the vertical axis of each bell, the others at equal distances around the circumference. The number of lugs on each end is as follows:

Nominal diameter, inches	Standard lugs, Classes *A* to *D*	High-pressure lugs	
		Classes *E* and *F*	Classes *G* and *H*
6	..	4	4
8 to 12, incl	4	4	4
14	4	6	6
16 to 24, incl	6	6	6
30	6	8	
36	8	12	
42 to 60, incl	8		

Marking.—Letters and other symbols shall be cast on fittings to identify the class, the maker's name, etc.

Quality of Material.—Specimen bars of the metal used, each being 26 in. long by 2 in. wide and 1 in. thick, shall be made without charge as often as the engineer may direct and, in default of definite instructions, the contractor shall make and test at least one bar from each heat or run of metal. The bars, when placed flatwise upon supports 24 in. apart and loaded in the center, shall support a load of 2,000 lb, and show a deflection of not less than 0.30 in. before breaking; or, if preferred, tensile bars shall be made which will show a breaking point of not less than 20,000 psi.

AGA Standard Specifications for CAST-IRON PIPE AND SPECIAL CASTINGS

Abstracted[1]

The AGA Standard Specifications for Cast Iron Pipe and Special Castings, adopted in 1913 and revised in 1929, are still used by the gas industry. The principal differences between the bell-and-spigot products of the gas and water industries are (*a*) in the grooving of the bells (compare sketches on Tables XV and XX); (*b*) there is only one weight of gas pipe which is equivalent to old Class A and intermediate between new Classes 50 and 100; (*c*) gas fittings are a short body pattern with a smaller radius than water fittings.

Designations and Dimensions.—The method of designating the sizes and outlets of AGA fittings is similar to that shown for AWWA fittings in Fig. 8. The dimensions of bell-and-spigot pipe with No. 1 bell are given in Table XIX. Standard bell-and-spigot fittings for gas include ¼, ⅛, and $\frac{1}{16}$ bends, tees, crosses, and miscellaneous fittings. Laying dimensions, metal thickness, and other information relating to fittings will be found in the AGA standard.

[1] For complete requirements, reference should be made to the AGA Standard. Copies may be obtained from the American Gas Association, 420 Lexington Ave., New York 17, N. Y.

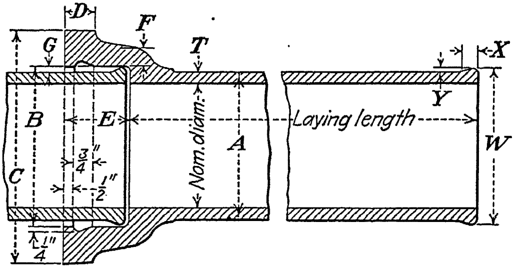
Hydrostatic Test.—All pipe, after having a general inspection, shall be subject to a water pressure test of at least 300 psi for sizes 16 in. and smaller, and at least 150 psi for sizes 20 in. and larger. If required by the engineer, they also shall be subjected to a hammer test under this pressure. Any pipe showing defects by leaking, sweating, or otherwise shall be rejected.

TABLE XIX.—DIMENSIONS AND WEIGHTS OF AGA STANDARD BELL-AND-SPIGOT PIPE[1]

(Standard No. 1 bell)[2]

X = 0.75 in. for 4 and 6 in.
1.00 in. for 8 to 48 in.

Y = 0.19 in. for 4 and 6 in.
0.25 in. for 8 to 48 in.



Nominal inside diameter, inches	Dimensions, inches									Approx. weight in pounds		
	A	B	C	D	E	F	G	T[3]	W	Bell	Per foot	12-ft length
4	4.80	5.80	8.40	1.50	4.00	0.59	0.50	0.40	5.18	27.00	19.50	234
6	6.90	7.90	10.70	1.50	4.00	0.62	0.50	0.43	7.28	39.50	30.58	367
8	9.05	10.05	13.50	1.50	4.00	0.69	0.50	0.45	9.55	52.80	42.42	509
10	[illegible]10	12.10	15.10	1.50	4.00	0.69	0.50	0.49	11.60	57.93	55.91	671
12	13.20	14.20	17.40	1.50	4.50	0.75	0.50	0.54	13.70	79.47	73.83	886
16	17.40	18.40	22.00	1.75	4.50	0.90	0.50	0.62	17.90	125.18	112.58	1351
20	21.60	22.85	26.85	1.75	4.50	0.97	0.63	0.68	22.10	169.10	153.83	1846
24	25.80	27.05	31.25	2.00	5.00	1.05	0.63	0.76	26.30	235.10	206.41	2477
30	31.74	32.99	37.59	2.00	5.00	1.15	0.63	0.85	32.24	315.20	284.00	3408
36	37.96	39.21	44.21	2.00	5.00	1.[illegible]	[illegible]	[illegible]	[illegible]	410.20	379.25	4551
42	44.20	45.45	51.05	2.00	5.00	1.40	0.63	1.07	44	537.50	497.66	5972
48	50.50	51.75	57.75	2.00	5.00	1.50	0.63	1.26	51	657.00	663.50	7962

[1] For dimensions of fittings, see AGA Standard.
[2] For dimensions of No. 2 bell, which is alternate for cement joint, see AGA Standard.
[3] Wall thicknesses of pipe given in the above table are nominal. An under-thickness tolerance of 0.08 in. is permitted for pipe sizes 4 to 36 in. inclusive and of 0.10 in. for pipe sizes 42 and 48 in.

Marking.—Each pipe and fitting shall have distinctly cast upon it the initials of the maker's name and, when made to special order, the year and other identifying symbols. The weight shall be painted in white on the inside of each pipe and fitting.

Quality of Material.—Specimen bars of the metal used, each being 26 in. long by 2 in. wide and 1 in. thick, shall be made without charge as often as the engineer may direct and, in default of definite instructions, the foundry shall make and test at least one bar from each heat or run of metal. The bars, when placed flatwise upon supports 24 in. apart and loaded in the center, shall support a load of 1,800 lb and show a deflection of not less than 0.30 in. before breaking; or, if preferred, tensile bars shall be made which will show a breaking point of not less than 18,000 psi. The foundry shall have the right to make and break three bars from each heat or run of metal, and the test shall be based on the average results of the three bars. Should the dimensions of the bars differ from those given above, a proper allowance therefor shall be made in the results of the tests.

TABLE XX.—JOINTING MATERIAL PER JOINT FOR AGA BELL-AND-SPIGOT PIPE[1]

Nominal inside diameter, in.	Cast lead			Lead wool			Cement	
	Lead		Yarn, oz.	Lead wool		Yarn, oz.	Cement, lb	Yarn, oz.
	Depth, in.	Pounds		Depth, in.	Pounds			
3	1¼	4½	4	1⅛	3	4		
4	1½	6	5	1⅛	4½	6	6	4
6	1¾	9	7½	1⅛	6½	9½	8	5¼
8	2	12	9	1⅛	9	13	10	6¾
10	2	15	11	1⅛	12½	16	11	8¼
12	2¼	22	14	1⅛	14	21	13	10
16	2½	36	22	1¼	20	36	17	14½
20	2½	50	44	1⅜	25	68	25	20½
24	2¾	62	57	1⅜	36	90		
30	3	75	67	1½	45	115		

[1] Reproduced by permission from "The American Gas Handbook." Quantities for cast-lead joints are from United Gas Improvement Co. Quantities are approximate only and will vary with the practice followed in making the joints. For cement joints ample allowance is made for waste.

Federal Specification WW-P-421 for PIPE: WATER, CAST-IRON (BELL-AND-SPIGOT AND BOLTED-JOINT)

Abstracted[1]

This specification covers cast-iron water pipe having bell-and-spigot joints and bolted joints in the following types:

[1] Embracing Amendment —3, issued Apr. 26, 1940. For subsequent amendments or for complete requirements, reference should be made to Federal Specification WW-P-421. Copies may be obtained from the Superintendent of Documents, Washington, D.C.

Type I.—Centrifugally cast in metal contact molds in 12- and 18-ft lengths with bell-and-spigot joints.

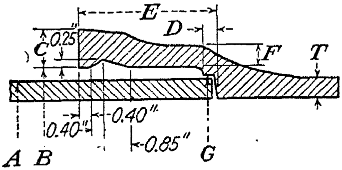
Type II.—Centrifugally cast in sand-lined molds in 16-, 16½-, and 20-ft lengths with bell-and-spigot joints.

Type III.—Horizontally cast in green-sand molds in 12- and 16-ft lengths with bell-and-spigot joints.

Type IV.—Cast in sand molds in 6- and 12-ft lengths with bolt lugs and integral machine-tapered joints (known in the trade as Universal pipe joints, see Fig. 5).

Type V.—Vertically cast in dry-sand molds in 12-ft lengths with bell-and-spigot joints (pit-cast pipe).

TABLE XXI-A.—DIMENSIONS OF TYPE I PIPE WW-P-421



Type I pipe, end details

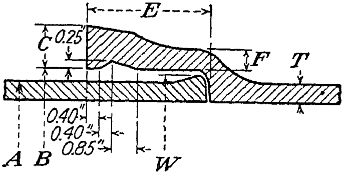
(All dimensions in inches)

Nominal inside diameter	Outside diameter of pipe	Inside diameter of bells	Total depth of bells	Depth of centering shoulder	Inside diameter of centering shoulder	Class 150, 150 lb working pressure, 346-ft head			Class 250, 250 lb working pressure, 576-ft head		
						Thickness of pipe	Thickness of bells		Thickness of pipe	Thickness of bells	
							At end	At shoulder		At end	At shoulder
	A	B	E	D	G	T	C	F	T	C	F
4	4.80	5.60	3.30	0.30	4.93	0.34	1.06	0.48	0.38	1.06	0.48
6	6.90	7.70	3.88	0.38	7.18	0.37	1.13	0.52	0.43	1.13	0.52
8	9.05	9.85	4.38	0.38	9.31	0.42	1.18	0.57	0.50	1.18	0.57
10	11.10	11.90	4.38	0.38	11.43	0.47	1.23	0.63	0.57	1.23	0.63
12	13.20	14.00	4.38	0.38	13.43	0.50	1.28	0.69	0.62	1.28	0.69
14	15.65	16.45	4.50	0.50	15.85	0.55	1.35	0.73	0.69	1.56	0.93
16	17.80	18.80	4.50	0.50	18.00	0.60	1.43	0.77	0.75	1.63	1.00
18	19.92	20.92	4.50	0.50	20.12	0.65	1.53	0.82	0.83	1.68	1.09
20	22.06	23.06	4.50	0.50	22.26	0.68	1.63	0.87	0.88	1.75	1.18
24	26.32	27.32	4.50	0.50	26.53	0.76	1.83	1.03	1.00	2.00	1.38

Pressure Classes and Hydrostatic Tests.—Two pressure classes are provided as follows: Class 150 for 150-lb working pressure, 346-ft head; and Class 250 for 250-lb working pressure, 576-ft head. Each length of pipe shall be subjected to a *hydrostatic test pressure* of 500 psi. The pipe shall be under this pressure at least ½ min and, while under pressure, shall be subjected to a hammer test. Any pipes showing defects by leaking, sweating, or otherwise, shall be rejected.

Quality of Material.—Cast iron shall be of good quality and of such character as to make the pipe strong, tough, and of even grain. It shall be soft enough to permit satisfactory drilling and cutting. The average Rockwell B number as determined from not less than three tests upon the machined edges of the test strips shall not exceed 95. An additional Rockwell B determination of not to exceed 95 shall be made upon the outside of each pipe cast in a metal contact mold using a portable machine.

TABLE XXI-*B*.—DIMENSIONS OF TYPE II PIPE (IN.), WW-P-421



Type II pipe, end details
(All dimensions in inches)

Nominal inside diameter	Outside diameter of pipe	Outside diameter of spigot end, max	Inside diameter of bells	Depth of bells	Class 150, 150 lb working pressure, 346-ft head: Thickness of pipe	Class 150: Thickness of bells, At end, min	Class 150: Thickness of bells, At shoulder, min	Class 250, 250 lb working pressure. 576-ft head: Thickness of pipe	Class 250: Thickness of bells, At end, min	Class 250: Thickness of bells, At shoulder, min
	A	*W*	*B*	*E*	*T*	*C*	*F*	*T*	*C*	*F*
4	4.80	5.36	5.60	3.50	0.34	1.28	0.44	0.38	1.28	0.44
6	6.90	7.46	7.70	3.50	0.37	1.39	0.48	0.43	1.39	0.48
8	9.05	9.61	9.85	4.00	0.42	1.47	0.52	0.50	1.47	0.62
10	11.10	11.66	11.90	4.00	0.47	1.59	0.57	0.57	1.59	0.69
12	13.20	13.76	14.00	4.00	0.50	1.75	0.62	0.62	1.75	0.74
14	15.65	16.23	16.45	4.00	0.55	1.47	0.62	0.69	1.62	0.77
16	17.80	18.38	18.80	4.00	0.60	1.52	0.67	0.75	1.72	0.82
18	19.92	20.50	20.92	4.00	0.65	1.62	0.72	0.83	1.82	0.87
20	22.06	22.64	23.06	4.00	0.68	1.77	0.77	0.88	1.92	0.92
24	26.32	26.90	27.32	4.00	0.76	1.92	0.82	1.00	2.02	0.97

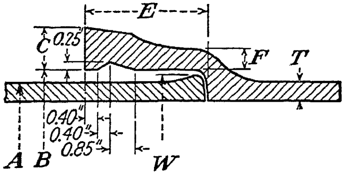
Sizes and Lengths.—Sizes from 4 to 24 in. inclusive are covered for Types I, II, IV, and V, and sizes from 4 to 12 in. inclusive for Type III. Nominal laying lengths shall be 6, 12, 16, 16½, 18, or 20 ft. Not more than 10 per cent of cut pipe of Types I, II, III, and V shall be accepted. Such cut pipe shall be not more than 1 ft under nominal laying length. No cut pipe of Type IV shall be accepted.

Dimensions and Weights.—The standard *dimensions* for Types I to IV inclusive are shown in Tables XXI*A*, *B*, *C*, and *D*. Dimensions for Type V are

identical with the corresponding pressure classes shown in Table XV for ASA A21.2 pit-cast pipe.

The nominal *weights* for Types I to IV inclusive differ somewhat from the corresponding pressure classes shown in Table XV for ASA 21.2 pit-cast pipe (with which Type V is identical), being 5 to 15 per cent lighter owing to the superior physical properties of the material. Where accurate weights are desired, reference should be made to the tables given in WW-P-421.

TABLE XXI-*C*.—DIMENSIONS OF TYPE III PIPE, WW-P-421



Type III pipe, end details

(All dimensions in inches)

Nominal inside diameter	Outside diameter of pipe	Outside diameter of spigot end, max	Inside diameter of bell	Thickness of pipe		Depth of bell	Thickness of bell at end	Thickness of bell at shoulder
				Class 150	Class 250			
	A	*W*	*B*	*T*	*T*	*E*	*C*	*F*
4	4.80	5.24	5.60	0.34	0.38	3.50	0.88	0 55
6	6.90	7.34	7.70	0.37	0.43	3.50	1.00	0.60
8	9.05	9.61	9.85	0.42	0.50	4.00	1.13	0.65
10	11.10	11.66	11.90	0.47	0.57	4.00	1.25	0.70
12	13.20	13.76	14.00	0.50	0.62	4.00	1.38	0.75

Tolerances.—For all types of pipe the tolerances in *wall thickness*, plus or minus, shall not exceed:

Nominal Diameter, In.	Tolerance ±, In.
4	0.04
6	0.045
8	0.05
10	0.055
12	0.06
14, 16, 18, 20, 24	0.08

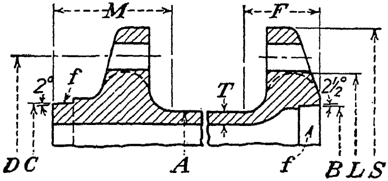
For all sizes of pipe, tolerances not exceeding 0.02 in. additional to the above will be allowed for spaces not exceeding 8 in. long in any direction.

For Types I, II, III, and V, the *inside diameters* of the bells and the outside diameters of the spigot ends (exclusive of the bead) shall not vary plus or minus from the tabulated dimensions by more than 0.06 in. for pipe 12 in. and less in nominal diameter, and 0.08 in. for pipe 14 in. or larger. For Type IV pipe the outside diameter of the body shall not vary plus or minus from the tabulated dimensions by more than the same amounts in the respective sizes. Dimensions

of the machined surfaces of joints for Type IV pipe shall conform to prescribed tolerances to be checked with plug-and-ring gauges furnished by the manufacturer.

The *weight* of no single pipe shall be less than the nominal tabulated weight by more than 5 per cent. The total weight of any order shall be not more than 2 per cent under nominal weight.

TABLE XXI-*D*.—DIMENSIONS OF TYPE IV (UNIVERSAL) PIPE (IN.), WW-P-421



End details of Type IV (Universal) pipe

(All dimensions in inches)

Nominal inside diameter	Outside diameter of pipe *A*	Thickness of pipe: Class 150 *T*	Thickness of pipe: Class 250 *T*	Bell, inside, bore *B*[1]	Spigot, outside, turn *C*[2]	Length of male end *M*	Length of female end *F*	Outside diameter of pipe end *L*	Diameter of bolt circle *D*	End to end of opposite lugs *S*	Size of bolts, diameter by length	Number of bolts: Class 150	Number of bolts: Class 250
4	4.80	0.34	0.38	5.25	5.28	3.25	2.13	7.19	8.00	9.63	⅝ × 5	2	2
6	6.90	0.37	0.43	7.00	7.03	4.44	2.63	9.25	10.25	12.25	¾ × 6	2	2
8	9.05	0.42	0.50	9.00	9.04	4.88	2.	1.50	12.75	15.25	⅞ × 6¾	2	2
10	1.10	0.47	0.57	11.25	11.29	5.75	3.38	14.00	15 38	18.00	1 × 7½	2	2
12	13.20	0.50	0.62	13.25	13.30	6.13	3.50	16.38	17 50	20.00	1 × 8	2	4
14	15.65	0.55	0.69	15.50	15.56	6.50	4.00	18 75	20 25	23 25	1¼ × 9	2	4
16	17.80	0.60	0.75	17.63	17.68	6.25	4.00	21.00	22.63	25.88	[illegible] × 9½	2	4
18	19.92	0.65	0.83	19.70	19.76	6.75	4.25	23.25	25.00	28.75	1[illegible] × 10	2	4
20	22.06	0.68	0.88	21.68	21.75	7.48	4.56	25.38	27.38	31.38	1[illegible] × 11½	2	4
24	26.32	0.76	1.00	25.92	26.00	7.69	4.69	29.75	31.75	35.75	1[illegible] × 12	2	4

[1] Machined taper of bells, 2½°.
[2] Machined taper of spigot, 2°.

Physical Requirements.—From each 300 lengths of pipe, or fraction thereof, of each size in the contract or order, one length of pipe shall be selected by the inspector before coating. From each sample pipe there shall be cut and machined one test strip 12 in. long, 0.50 in. deep, and the full thickness of the shell in width. This shall be tested as a beam (with machined surfaces top and bottom) on supports 10 in. apart with load applied at two points 3⅓ in. from the supports. The strip shall be accurately calipered at point of rupture and stress calculated by the formula

$$S = \frac{PLc}{\ }, \text{ or for the above specimen } S = \frac{40P}{b}.$$

The secant modulus of elasticity at the breaking load shall be calculated by the formula

$$E = \frac{23PL^3}{1{,}296Iy}, \text{ or for the above specimen } E = S\,\frac{42.6}{}$$

In these formulas S = modulus of rupture, psi.
E = modulus of elasticity, psi.
P = total load, lb.
L = length of span, in.
c = distance to extreme fiber, in.
I = moment of inertia, in.^{4}
b = width of specimen (thickness of pipe), in.
y = center deflection at load P, in.

The secant modulus of elasticity and the corresponding modulus of rupture shall conform to the following requirements for the respective types of pipe:

Type	Max secant modulus of elasticity, psi	Min modulus of rupture, psi
I	12,000,000	40,000
II	10,000,000[1]	40,000[1]
III	10,000,000[1]	40,000[1]
IV	10,000,000[1]	40,000[1]
V	10,000,000	30,000

[1] Where the modulus of elasticity exceeds 10,000,000 psi the modulus of rupture shall exceed 40,000 psi by at least the same percentage.

Chemical Composition.—Chemical analysis shall be made by the manufacturer from each heat to determine graphitic and combined carbon, or total carbon if the separate determinations are not available, manganese, phosphorus, sulphur, and silicon. Duplicate copies of test reports shall be furnished when requested. Sulphur shall not exceed 0.10 per cent and phosphorus 0.90 per cent in either ladle or inspection analysis.

Marking.—Each length of pipe shall be weighed and the weight plainly and indelibly marked on the pipe. Coated pipe shall be weighed after coating. Cement-lined pipe shall be weighed before lining and marked on the outside surface. In addition to the weight marking, each pipe of Types I, II, III, and V shall have distinctly cast or stamped into the metal on the face or outside surface of the bell, or on the body near the bell or spigot, the manufacturer's mark, and the year in which the pipe was cast. When so specified a designated symbol shall be cast or stamped on each piece. Type IV pipe shall be similarly marked near one end.

Coating and Lining.—Pipe shall be furnished coated with coal-tar pitch varnish unless cement-lined or uncoated pipe is specified. In general the requirements for cement lining are similar to the American Standard Specifications for Cement Mortar Lining for Cast Iron Pipe and Fittings, ASA A21.4 (see page 1270). Bolts and nuts for Type IV pipe shall be zinc-coated mild steel, galvanized by the hot process after threading.

ASTM Standard Specifications for CAST-IRON CULVERT PIPE

Serial Designation A142-38

Abstracted[1]

These specifications cover three classes of pipe intended for use in the construction of culverts:

Standard cast-iron culvert pipe.

Heavy cast-iron culvert pipe.

Extra-heavy cast-iron culvert pipe.

Standard pipe may be (*a*) smooth, (*b*) corrugated, or (*c*) ribbed. Each length shall be cast as a unit, either vertically or horizontally, in green-sand molds or by centrifugal processes. Suitable devices such as hub ends or interlocking ends shall be provided to prevent displacement at joints.

Dimensions and Weight.—The shell thicknesses and weights shall conform to the requirements of Table XXII.

TABLE XXII.—NOMINAL DIMENSIONS AND WEIGHTS OF CAST-IRON CULVERT PIPE, ASTM A142

Nominal diameter, in.	Smooth pipe						Corrugated or ribbed pipe	
	Standard		Heavy		Extra-heavy			
	Thickness, in.	Weight[1] per ft, lb	Thickness, in.	Weight[1] per ft, lb	Thickness, in.	Weight[1] per ft, lb	Thickness, in.	Weight[1] per ft, lb
12	0.37	45	0.37	45	0.40	49		
14	0.37	52	0.40	57	0.46	65		
15	....	...	....	...	....	...	0.25	45
16	0.40	64	0.46	74	0.53	86		
18	0.42	76	0.52	95	0.60	110	0.25	50
20	0.47	94	0.57	115	0.66	134		
24	0.56	135	0.69	167	0.80	195	0.31	85
30	0.70	211	0.86	261	1.00	304	0.38	125
36	0.84	304	1.03	374	1.20	438	0.44	165
42	0.98	414	1.20	509	1.40	597		
48	1.12	540	1.38	669	1.60	779		

[1] Nominal weight per foot of barrel in pounds, exclusive of hub.

Unless otherwise specified, the minimum laying length shall be 3 ft. The minimum inside diameter shall be not less than the nominal diameter by more than ⅛ in. The shell thickness at any point shall be not more than 15 per cent under the thickness given in Table XXII. The weight of any section of pipe shall be not more than 5 per cent under the weight given in Table XXII.

[1] For complete specification, reference may be made to ASTM A142 (see note p. 369).

Material.—The pipe shall be gray cast iron of good quality and of such character as to make the metal strong, tough, of even grain, and soft enough for drilling and cutting.

Chemical Composition.—The phosphorus content as determined by ladle analysis shall not exceed 0.90 per cent, and the sulphur 0.12 per cent. The manufacturer shall maintain a daily record of chemical analysis which shall be open to inspection by the purchaser.

Strength Requirements.—The pipe shall not fail or develop cracks under the following loads when tested by the three edge bearing method defined in the specification.

Class of Pipe	Load, Lb per Ft of Laying Length
Standard pipe	2,000d'
Heavy pipe	3,000d'
Extra-heavy pipe	4,000d'

NOTE.—d' = the nominal inside diameter of pipe in feet.

Full-length specimens shall not be tested to destruction if they will sustain, without cracking, a load 10 per cent in excess of that specified. If the purchaser desires that full-length specimens be tested to destruction, he shall specify in the order the number of such tests that will be required. Ring specimens cut from the spigot ends of pipe shall be tested to destruction. The purchaser may require strength tests in such numbers as he may deem necessary, provided, if the pipe meets the requirements as to shell thickness and weight, the number of specimens tested, unless otherwise specified in the purchase order, shall not exceed one pipe or 5 per cent, whichever may be larger, of each size and class ordered.

Coating.—All pipe shall be completely coated inside and outside by immersion in coal-tar pitch varnish.

American Standard for CAST-IRON SOIL PIPE AND FITTINGS

ASA A40.1-1935

Abstracted[1]

Cast-iron soil pipe is used extensively for plumbing drainage work. Pipe is cast with hub-and-spigot ends or with hubs on both ends. Hubs are provided with lead grooves and the spigots with beads to facilitate centering the joint and to provide greater holding strength for the leaded joint. The principal dimensions and weights of extra-heavy soil pipe and fittings are reproduced in Tables XXIII and XXIV. Approximate weights of lead and hemp needed per joint are given in Table XXV.

The castings shall be gray iron of good quality made by the cupola, air-furnace, electric-furnace, or other approved process. The resultant pipe and fittings shall be of compact, close-grained metal, soft enough to permit drilling and cutting.

[1] For complete standard, reference may be made to ASA A40.1 (see note, p. 430).

TABLE XXIII.—DIMENSIONS AND WEIGHTS OF AMERICAN STANDARD CAST-IRON SOIL PIPE, ASA A40.1

Size	Laying length, feet	Wall thickness, inches	Weight of single-hub pipe, 5-ft length, pounds	Telescoping length, inches
2	5	3/16	25	2½
3	5	¼	45	2¾
4	5	¼	60	3
5	5	¼	75	3
6	5	¼	95	3
8	5	5/16	150	3½
10	5	⅜	215	3½
12	5	⅜	270	4¼
15	5	7/16	375	4¼

The chemical composition and physical properties shall conform to the following requirements:

Sulphur—not over 0.11 per cent.
Phosphorus—not over 0.90 per cent.
Tensile strength, psi—not less than 21,000.
Transverse test bar on supports 18 in. apart.
(1.20 in diameter × 21 in. long):
Breaking load at center, min, 1,750 lb.
Average deflection at center, min, 0.20 in.

The tension test bar is optional. From each heat and for every 2-hr operation, a round transverse-test bar (arbitration-test bar) shall be cast. Tests are to be based on average results of all test bars. The transverse bars shall be tested in accordance with and the results conform to the values reproduced above from ASTM Specification A74-42.

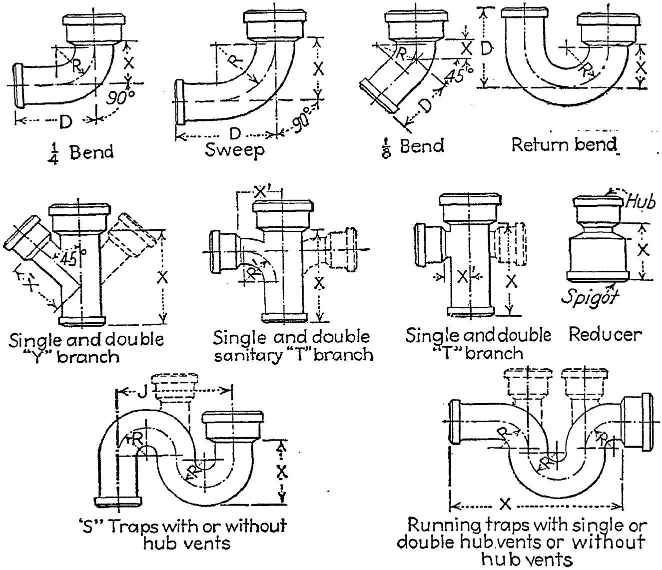
All pipe shall be tested to a hydrostatic pressure of not less than 50 psi before coating.

Soil pipe and fittings are usually coated by heating to about 300 F and dipping in a bath of coal-tar pitch and oil.

A large variety of soil-pipe fittings are required by the trade. In sizes 2 to 6 in., 50 separate and distinct items are included in ASA A40.1 while 23 items are required for sizes 8 to 15 in., inclusive. In developing this standard it was found possible to eliminate some 40 per cent of the items manufactured at that time.

Jointing of Bell-and-spigot Pipe.—In the absence of standard weights of jointing materials needed per joint for the different types of bell-and-spigot cast-iron pipe and fittings, the following estimates are furnished for convenience. For AGA gas pipe, see Table XX; for ASA A21.2, ASTM A44, AWWA, and Federal Specification WW-P-421 water pipe, see Table XXVI; for ASA A40.1 soil pipe, see Table XXV.

Table XXIV.—Laying Dimensions, In., American Standard Cast-iron Soil-pipe Fittings, ASA A40.1



Fitting		Size 2	3	4	5	6	8	10	12	15
¼ bend	X	3¼	4	4½	5	5½	6⅝	7⅝	8¾	10¼
	D	6	7	8	8½	9	11½	12½	15	16½
Short sweeps	X	5¼	6	6½	7	7½	8⅝	9⅝	10¾	12¼
	D	8	9	10	10½	11	13½	14½	17	18½
Long sweeps	X	8¼	9	9½	10	10½	11⅝	12⅝	13¾	15¼
	D	11	12	13	13½	14	16½	17½	20	21½
⅛ bend	X	1½	1 15/16	2 3/16	2⅜	2 9/16	3⅛	3½	4 1/16	4 11/16
	D	4¼	4 15/16	5 11/16	5⅞	6 1/16	8	8⅜	10 5/16	10 15/16
Return bend	X	3¼	4	4½	5	5½	6⅝			
	D	6	7	8	8½	9	11½			
Y branch[1]	X'	4	5½	6¾	8	9¼	11 13/16	14½	16⅞	20¾
	X	8	10½	12	13½	15	19½	22½	27	31½
Sanitary T branch	X'	2¾	4	4½	5	5½	6⅝	7⅝	8¾	10¼
	X	8	10	11	12	13	17	19	22½	25½
T branch[2]	X'	1¾	2½	3	3½	4	5¼	6¼	7½	9
	X	8	10	11	12	13	17	19	22½	25½
Reducers[3]	X	....	4¾	5	5	5	6	6	6½	6½
S trap	J center-to-center	8	10	12	14	16				
	X	4	5¾	7½	9½	11½				
Running trap	Size	2×2	3×3	4×4	5×5	6×6	8×6	10×6	12×8	15×8
	X	12	15	17½	19½	21½	27⅝	31⅝	38¾	45¾

[1] Single and double Y branch of same size run and branch.
[2] For venting and cleanout purposes, not approved for use as waste inlets.
[3] Same laying dimension for luctions, *i.e.*, 10 × 8 is same as 10 × 4.

TABLE XXV.—APPROXIMATE WEIGHTS OF CAST LEAD AND HEMP NEEDED PER JOINT FOR CAST-IRON SOIL PIPE AND FITTINGS

Nominal pipe size, inches	Weight per joint, pounds	
	Lead	Hemp[1]
2	1½	1/7
3	2¼	1/6
4	3	1/5
5	3¾	¼
6	4½	⅓
8	6	3/7
10	8	½
12	9½	4/7
15	13	⅞

[1] Very approximate only. The actual weight of hemp will vary with its own density as well as with the amount used in preparing a joint.

The requirements for *minimum depth of jointing materials* shown in Table XXVII and the following specifications for jointing materials and making joints are taken from the AWWA Standard Specifications 7D.1 for Laying Cast-Iron Pipe. Lead for calking purposes shall contain not less than 99.73 per cent pure lead. Impurities shall not exceed the following limits:

Materials	Per Cent
Arsenic, Antimony, and Tin Together	0.015
Copper	0.080
Zinc	0.002
Iron	0.002
Bismuth	0.250
Silver	0.020

Pipe joints shall be sealed with cast lead, with cast sulphur compound, or with portland cement as required, following the preparation of the joint base with yarning material. Braided hemp or jute, or other suitable yarning material shall be of proper dimension to center the spigot in the bell. Each strand of yarn shall be cut somewhat longer than the circumference of the pipe, so that the ends will overlap. The overlapped ends of successive strands shall be staggered. Each strand shall be thoroughly packed and hammered home into the joint with suitable yarning tools.

Each lead joint shall be made with one pour, filling the joint space to the depth specified in Table XXVII. After cooling to the temperature of the pipe, lead joints shall be calked by means of pneumatic tools, or by hand tools, by competent workmen until thoroughly compacted, making watertight joints without overstraining the bells.

Each sulphur compound joint shall be filled with one continuous pour while the compound is at the proper temperature. The solidified compound in the pouring gate shall be cut off flush with the top of the bell or broken off flush with the top of the joint.

Portland-cement joints after yarning shall be filled with neat cement, barely moistened, which shall be calked by competent workmen until it is thoroughly compacted without overstraining the bell to make a watertight joint. The cement shall be so dry that it will ring with a metallic sound while being calked. Cement joints shall be covered immediately with slightly moist earth, or with damp burlap, for the proper time to ensure complete hydration. Pipes joined with cement shall not be subjected to water pressure until 36 hr have elapsed after the last joint was made, unless the engineer shall authorize a shorter period. The cement shall have aged for 2 weeks before the joints are tested for leakage.

TABLE XXVI.—APPROXIMATE WEIGHTS OF CAST LEAD AND HEMP REQUIRED PER JOINT FOR ALL CLASSES OF BELL-AND-SPIGOT CAST-IRON WATER PIPE AND FITTINGS, COVERED BY THE FOLLOWING STANDARDS AND SPECIFICATIONS: AWWA, ASA A21.2, ASTM A44, FEDERAL WW-P-421[1]

Nominal pipe diameter, inches	Weight of lead per joint, pounds				Weight of hemp per joint, lb[3]
	Depth of lead in bell, inches[2]				
	2	2¼	2½	3	
3	6.0	6.6	7.2	8.3	0.18
4	7.5	8.2	8.9	10.3	0.21
6	10.3	11.3	12.2	14.2	0.31
8	13.3	14.5	15.8	18.3	0.44
10	16.0	17.7	19.2	22.3	0.53
12	19.0	20.8	22.6	26.2	0.61
14	22.0	24.1	26.1	30.3	0.81
16	30.0	33.0	36.0	42.0	0.94
18	33.8	36.9	40.2	46.7	1.00
20	37.0	40.6	44.3	51.6	1.25
24	44.0	48.3	52.6	61.3	1.50
30	54.3	60.0	65.3	76.1	2.06
36	64.8	71.6	78.0	90.7	3.00
42	75.3	83.0	90.5	105	3.62
48	85.6	94.7	103	120	4.37
54	97.6	106	115	134	6.25
60	108	117	128	149	8.25
72	146	162	178	210	12.5
84	170	188	207	244	15.0

[1] Courtesy Thos. F. Wolfe, Engineer, Cast Iron Pipe Research Association.
[2] For minimum depths required, see Table XXVII.
[3] Very approximate only. The actual weight of hemp will vary with its own density and the depth left for pouring lead.

Lead wool, consisting of finely spun lead in continuous strands, can be used without heating to fill bell-and-spigot joints of water or gas mains. This application originated in Germany where lead

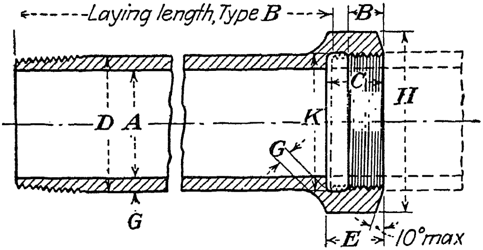
wool was used to a considerable extent for making joints in large-diameter cast-iron pipe. For pipes of less than 24 in. in diameter, lead-wool joints are said to cost more than cast-lead, but for larger pipes the cost of each kind is about the same. It has been used to a considerable extent in the United States, particularly for gas mains and for water pipes laid either in wet trenches or under water where there is a need for cold application. Because each skein or strand is calked separately as it is introduced into the joint, the lead compacts into a very dense material which has great holding qualities suited for pressures up to 500 psi. Well-calked lead wool becomes denser than cast lead which shrinks from the iron in cooling and can be expanded out again to only a limited depth. Hence depths of lead wool in the bell of only 1⅜ to 1⅝ in. are said to suffice for pipe diameters of 20 in. and greater.

TABLE XXVII.—MINIMUM DEPTHS OF JOINTING MATERIAL FOR BELL-AND-SPIGOT JOINTS, FROM AWWA SPECIFICATION 7D.1 FOR LAYING CAST-IRON PIPE

Jointing material	Min. depth, inches	Nominal pipe diameter, inches
Cast lead	2¼	3 to 20
	2½	24, 30, 36
	3	36 and larger
Sulphur compound	2½	3 to 24
	2¾	30, 36
	3½	48
	4	54, 60
Cement	3	All

One of the *sulphur compounds* known as *leadite,* which is composed of a finely ground mixture of iron sulphur, slag, and salt, is used to a considerable extent for making bell-and-spigot joints in cast-iron water and gas mains. Leadite weighs only 118 lb per cu ft when melted, whereas lead weighs 710 lb per cu ft. One ton of leadite will make about five times as many joints as a ton of lead. Leadite melts at a temperature of about 400 F and is poured like molten lead. As it expands in cooling and forms a vitreous, watertight substance, leadite requires no calking and, when the joints are properly made, can be used for pressures up to 250 psi. Since leadite is more elastic than lead, it is not squeezed out of a joint as lead may be when a pipe line settles.

TABLE XXVIII.—DIMENSIONS OF PIPE AND DRAINAGE HUBS FOR AMERICAN STANDARD THREADED CAST-IRON PIPE
(Table 1 of ASA A40.5-1943)



(All dimensions in inches, except where otherwise stated)

Pipe size	Pipe			Drainage hubs						Nominal weights	
	Nominal diameter		Wall thickness, min	Thread length[1]	Diameter of groove		End to shoulder[1]	Minimum band		Type A and barrel of Type B per ft	Additional weight of hubs for Type B
	Outside	Inside			Max	Min		Diameter	Length		
	D	*A*	*G*	*B*	*K*	*K*	*C*	*H*	*E*		
1¼	1.66	1.23	0.187	0.42	1.73	1.66	0.71	2.39	0.71	3.033	0.60
1½	1.90	1.45	0.195	0.42	1.97	1.90	0.72	2.68	0.72	3.666	0.90
2	2.38	1.89	0.211	0.43	2.44	2.37	0.76	3.28	0.76	5.041	1.00
2½	2.88	2.32	0.241	0.68	2.97	2.87	1.14	3.86	1.14	7.032	1.35
3	3.50	2.90	0.263	0.76	3.60	3.50	1.20	4.62	1.20	9.410	2.80
4	4.50	3.83	0.294	0.84	4.60	4.50	1.30	5.79	1.30	13.751	3.48
5	5.56	4.81	0.328	0.93	5.66	5.56	1.41	7.05	1.41	19.069	5.00
6	6.63	5.76	0.378	0.95	6.72	6.62	1.51	8.28	1.51	26.223	6.60
8	8.63	7.63	0.438	1.06	8.72	8.62	1.71	10.63	1.71	39.820	10.00
10	10.75	9.75	0.438	1.21	10.85	10.75	1.92	13.12	1.93	50.234	
12	12.75	11.75	0.438	1.36	12.85	12.75	2.12	15.47	2.13	60.036	

[1] The length of thread, *B*, and the end to shoulder, *C*, shall not vary from the dimensions shown by more than plus or minus the equivalent of the pitch of one thread.

American Standard for THREADED CAST-IRON PIPE

ASA A40.5-1943

Abstracted[1]

Two types of threaded cast-iron pipe are provided in 5 to 27 ft lengths, as follows:

[1] For complete standard, reference may be made to ASA A40.5 (see note, p. 430).

Type A, with external threads on both ends.

Type B, with external threads on one end and with internal threaded drainage hubs on the other.

The dimensions prescribed are based on high-quality gray iron which may be cut and threaded at any point in the length by conventional tools. Sulphur content shall not exceed 0.11 per cent, phosphorus shall not exceed 0.90 per cent. At the manufacturer's option, the pipe shall meet the transverse test requirements either of ASA A21.2 using the 2 by 1 bar, or ASTM A48, using bar *B*, or it shall show tensile strength not less than 21,000 psi using the tension test bar of ASTM A126. Threads shall conform to the American Standard for Pipe Threads, ASA B2.1.

Each length of pipe shall be tested to either a hydrostatic or air pressure of 50 psi. Higher test pressures may be used at the option of the manufacturer. Pipe shall be furnished uncoated unless otherwise specified.

Dimensions of pipe and drainage hubs shall be as given in Table XXVIII.

American Standard for CAST-IRON SCREWED DRAINAGE FITTINGS

ASA B16.12-1942

Abstracted[1]

These fittings are designed for drainage systems using threaded steel pipe or iron-pipe-size cast-iron pipe. The recesses on Types 1 and 2 fittings are intended to match the bore of Schedule 40 (standard-weight) steel pipe so that, when the joints are made up, the ends of the pipe practically meet the shoulders, thereby forming smooth passageways. Fittings are provided in a variety of shapes, some of which are shown in Tables XXIX to XXXI, inclusive.

Marking.—Each fitting shall be marked for identification with the maker's name or symbol.

Material.—The requirements of Grade A castings, ASTM A126, abstracted on page 625, shall be met.

Tolerances.—The minus tolerances given in the following table may be permitted on inspection to apply to the diameter and width of band:

Nominal pipe size	1¼	1½	2	2½	3	4	5	6
Diameter and width of band,	0.035	0.04	0.04	0.05	0.05	0.06	0.06	0.07

All dimensions are given in inches.

The center-to-end dimensions shall not vary from the given dimensions by more than shown on the following table:

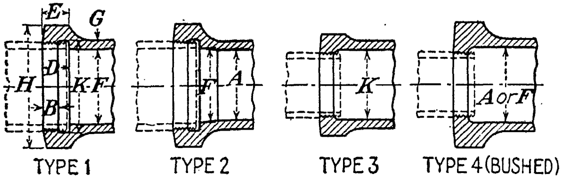
Nominal pipe size	1¼	1½	2	2½	3	4	5	6
Center to end ±	0.07	0.08	0.08	0.10	0.10	0.12	0.12	0.12

All dimensions are given in inches.

[1] For complete standard, reference may be made to ASA B16.12 (see note, p. 430).

The above limits apply to all fittings covered by this standard. Inspection limits for lengths of couplings, increasers, and offsets shall be double the limits for center-to-end dimensions. The largest opening in the fitting governs the tolerance to be applied to all openings.

Table XXIX.—American Standard Cast-iron Screwed Drainage Fittings, Dimensions of Threaded Ends (Table 3, ASA B16.12-1942)



(All dimensions in inches)

Nominal pipe size[5]	Length of threads[1]	Total length of thread chamber to shoulder[1]	Width of band[2]	Inside diameter of fitting[3]	Metal thickness[4]	Outside diameter of band[2]	Inside diameter of groove	
							Max	Min
A	B	D	E	F	G	H	K	K
1¼	0.420	0.7068	0.71	1.380	0.185	2.39	1.730	1.660
1½	0.420	0.7235	0.72	1.610	0.200	2.68	1.970	1.900
2	0.436	0.7565	0.76	2.067	0.220	3.28	2.445	2.375
2½	0.682	1.1375	1.14	2.469	0.240	3.86	2.975	2.875
3	0.766	1.2000	1.20	3.068	0.260	4.62	3.600	3.500
4	0.844	1.3000	1.30	4.026	0.310	5.79	4.600	4.500
5	0.937	1.4063	1.41	5.047	0.380	7.05	5.663	5.563
6	0.958	1.5125	1.51	6.065	0.430	8.28	6.725	6.625

[1] Length of thread, *B*, and total length of thread chamber to shoulder, *D*, shall not vary from the dimensions shown in the table by more than plus or minus the equivalent of the pitch of one thread.

[2] The minus tolerances given may be permitted on inspection to apply to the outside diameter of band, *H*, and the width of band, *E*.

[3] Inside diameter of fitting, *A* and *F*, shall not vary from dimensions given by more than ±1/32 in. for sizes 1 to 4 in. and ±1/16 in. for sizes 5 and 6 in.

[4] Patterns shall be designed to produce castings of a metal thickness given in the table. Metal thickness at no point shall be less than 90 per cent of the thickness given in the table.

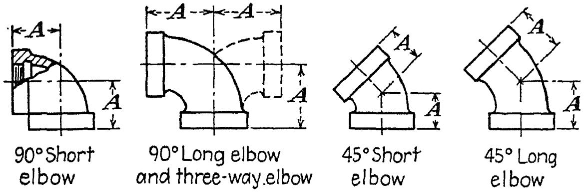
[5] Type 2 fittings having the nominal openings shown in the first column should be tapered as shown.

[6] The shoulder and groove, Types 1 and 2, are required on outlet connections. They are also required on inlet connections which are the same size as the outlet connections. Inlet connections which are smaller than the outlet connections may be made as shown in Type 3 or Type 4, at the manufacturer's option.

Threading.—The American Standard for Taper Pipe Threads, ASA B2.1, shall be used in threading all fittings covered by this standard. All internal thread shall be countersunk a distance not less than one-half the pitch of the thread and at an angle of 45 deg with the axis of the thread for the purpose of

easy entrance in making up a joint and for the protection of the thread. Countersinking and chamfering shall be concentric with the thread and shall be included in the length of the thread. The maximum allowable variation in the alignment of threads of all openings shall be 1/16 in. in one foot. All fittings having openings at 90 deg are tapped with a pitch of 1/4 in. to the foot.

TABLE XXX.—AMERICAN STANDARD CAST-IRON SCREWED DRAINAGE FITTINGS, CENTER-TO-END DIMENSIONS OF ELBOWS
(Table 4, ASA B16.12-1942)



(All dimensions in inches)

Nominal pipe size	90-deg elbow		45-deg elbow		60-deg elbow	22½-deg elbow	11¼-deg elbow	5⅝-deg elbow
	Short[1]	Long[2]	Short[1]	Long				
	A	A	A	A	A	A	A	A
1¼	1¾	2¼	1 5/16	1¾	1 9/16	1⅛	1 1/16	1 3/16
1½	1 15/16	2½	1 7/16	1⅞	1¾	1¼	1¼	1 5/16
2	2¼	3 1/16	1 11/16	2¼	2 1/16	1 7/16	1⅜	1½
2½	2 11/16	3 11/16	1 15/16	2⅝	2½	1¾	1⅝	1⅝
3	3 1/16	4¼	2 3/16	2 15/16	2⅞	2	1 13/16	1¾
4	3 13/16	5 3/16	2⅝	3½	3⅜	2 5/16	2	1⅞
5	4½	6⅛	3 1/16	4⅛	3⅞	2⅝	2¼	2
6	5⅛	7⅛	3 7/16	4⅞	4¼	2 15/16	2⅜	2¼

All drainage fittings having openings at 90 deg are tapped with the inlet opening pitched ¼ in. per ft.

[1] Same as adopted for 125 Lb Cast-Iron Screwed Fittings (ASA B16d).

[2] Three-way elbows have same dimensions as 90-deg long-radius elbows.

Coatings.—Fittings are uncoated, galvanized, or coated with a black composition.

Ribs.—The addition of ribs or lugs is permitted.

CONCRETE AND CLAY PIPE

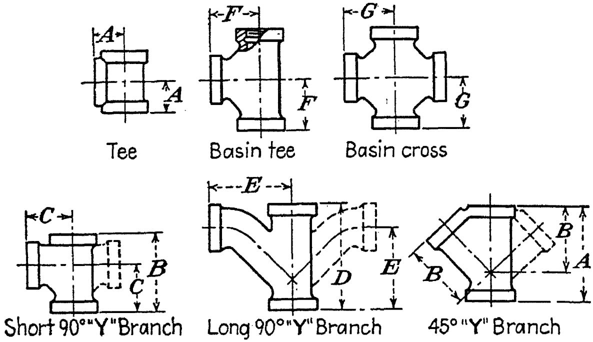
Concrete and clay sewer pipe, concrete irrigation pipe, and clay and concrete drain tile are covered in the following ASTM specifications.[1]

[1] For complete specifications, reference may be made to the ASTM specifications (see note, p. 369). See also "Concrete Pipe Lines," by M. W. Loving, American Concrete Pipe Association, Chicago, Ill., 1942.

Concrete Sewer Pipe.—ASTM Specification C-14 covers unreinforced-concrete sewer pipe, sizes 4 to 24 in., inclusive. The avail-

TABLE XXXI.—AMERICAN STANDARD CAST-IRON SCREWED DRAINAGE FITTINGS, DIMENSIONS OF TEES, CROSSES, AND Y-BRANCHES

(Tables 5 and 7, ASA B16.12)



(All dimensions in inches)

Nominal pipe size	Center to end of tee[1]	90-deg short[2] Y-branch		90-deg long[2] Y-branch		Center to end of basin tee	Center to end of basin	45-deg Y-branch	
		End to end	Center to end	End to end	Center to end			End to end	Center to end
	A	*B*	*C*	*D*	*E*	*F*	*G*	*A*	*B*
1¼	1¾	3¾	2¼	4¾	3⅝	/16	2 5/16	5	3¼
1½	1 15/16	4¼	2½	5⅜	4⅛	2 11/16	2 11/16	5½	3⅝
2	2¼	5 3/16	3 1/16	7	5¼	3½	3½	6½	4⅜
2½	2 11/16	6 5/16	3 11/16	8¼	¼	4¼		7⅞	5⅜
3	3 1/16	7¼	4¼	9⅞	7½				6 3/16
4	3 13/16	8¾	5 3/16	13	9⅞			⅞	7 11/16
5	4½	10 5/16	6⅛	15¾	12¼			12 15/16	9 3/16
6		11 15/1	7⅛	18¾	14⅝			14⅞	10¾

All drainage fittings having openings at 90 deg are tapped with the inlet opening pitched ¼ in. per ft.

[1] Same dimensions as adopted for 125 Lb Cast-Iron Screwed Fittings (ASA B16d).

[2] Double Y-branches have same dimensions as single Y-branches.

able laying lengths, thicknesses, and socket dimensions differ slightly from those given in Tables XXXII-*A* and -*B* for clay sewer pipe but not sufficiently to affect design of a sewer.

Clay Sewer Pipe.—ASTM Specifications C-13 and C-200 cover *standard-strength* and *extra-strength* clay pipe manufactured from

TABLE XXXII-*A*.—STRENGTH AND DIMENSIONS OF STANDARD-STRENGTH CLAY SEWER PIPE
(Table I, of ASTM C13-44T)

Size, in.	Minimum average strength, lb. per lin. ft		Laying length		Maximum difference in length of two opposite sides, in.	Outside diameter of barrel, in.	
	Three-edge bearing method	Sand bearing method	Nominal, ft	Limit of minus variation,[1] in. per ft of length		Minimum	Maximum
4	1,000	1,425	2, 2½, 3	¼	5/16	4⅞	5⅛
6	1,000	1,425	2, 2½, 3	¼	⅜	7 1/16	7 7/16
8	1,000	1,425	2, 2½, 3	¼	7/16	9¼	9¾
10	1,100	1,570	2, 2½, 3	¼	7/16	11½	12
12	1,200	1,710	2, 2½, 3	¼	7/16	13¾	14 5/16
15	1,400	2,000	3, 4	¼	½	17 3/16	17 13/16
18	1,700	2,430	3, 4	¼	½	20⅝	21 7/16
21	2,000	2,860	3, 4	¼	9/16	24⅛	25
24	2,400	3,430	3, 4	⅜	9/16	27½	28½
27	2,750	3,930	3, 4	⅜	9/16	31	32⅛
30	3,200	4,570	3, 4	⅜	⅝	34⅜	35⅝
33	3,500	5,000	3, 4	⅜	⅝	37⅝	38 15/16
36	3,900	5,570	3, 4	⅜	1 1/16	40¾	42¼

Size, in.	Inside diameter of socket at ½ in. above base, in.		Depth of socket, in.		Thickness of barrel, in.		Thickness of socket at ½ in. from outer end, in.	
	Minimum	Maximum	Nominal	Minimum	Nominal	Minimum	Nominal	Minimum
4	5¾	6⅛	1¾	1½	½	7/16	7/16	⅜
6	8 3/16	8⅝	2¼	2	⅝	9/16	½	7/16
8	10½	11	2½	2¼	¾	1 1/16	9/16	½
10	12¾	13¼	2⅝	2⅜	⅞	1 3/16	⅝	9/16
12	15⅛	15¾	2¾	2½	1	1 5/16	¾	1 1/16
15	18⅝	19¼	2⅞	2⅝	1¼	1⅛	1 5/16	⅞
18	22¼	23	3	2¾	1½	1⅜	1⅛	1 1/16
21	25⅞	26¾	3¼	3	1¾	1⅝	1 5/16	1 3/16
24	29⅜	30⅜	3⅜	3⅛	2	1⅞	1½	1⅜
27	33	34⅛	3½	3¼	2¼	2⅛	1 11/16	1 9/16
30	36½	37¾	3⅝	3⅜	2½	2⅜	1⅞	1¾
33	39⅞	41¼	3¾	3½	2⅝	2½	2	1 13/16
36	43¼	44¾	4	3¾	2¾	2⅝	2 1/16	1⅞

[1] There is no limit for plus variation.

surface clay, fire clay, or shale, or a combination of these materials. The strengths, laying lengths, thickness, and socket dimensions for sizes 4 to 36 in., inclusive, are given in Tables XXXII-*A* and -*B*.

TABLE XXXII-*B*.—STRENGTH AND DIMENSIONS OF EXTRA-STRENGTH CLAY SEWER PIPE
(Table I, of ASTM C200-44T)

Nominal size, in.	Minimum average strength, lb. per lin. ft		Laying length		Maximum difference in length of two opposite sides, in.	Outside diameter of barrel, in.	
	Three-edge bearing method	Sand bearing method	Nominal, ft	Limit of minus variation,[1] in. per ft of length		Minimum	Maximum
6	2,000	2,850	2, 2½, 3	¼	⅜	7 1/16	7 7/16
8	2,000	2,850	3	¼	7/16	9¼	9¾
10	2,000	2,850	3	¼	7/16	11½	12
12	2,250	3,200	3	¼	7/16	13¾	14 5/16
15	2,750	3,925	3, 4	¼	½	17 3/16	17 13/16
18	3,300	4,700	3, 4	¼	½	20⅝	21 7/16
21	3,850	5,500	3, 4	¼	9/16	24⅛	25
24	4,400	6,300	3, 4	⅜	9/16	27½	28½
30	5,000	7,100	3, 4	⅜	⅝	34⅜	35⅝
36	6,000	8,575	3, 4	⅜	1 1/16	40¾	42¼

Nominal size, in.	Inside diameter of socket at ½ in. above base, in.		Depth of socket, in.		Thickness of barrel, in.		Thickness of socket at ½ in. from outer end, in.	
	Minimum	Maximum	Nominal	Minimum	Nominal	Minimum	Nominal	Minimum
6	8 3/16	8⅝	2¼	2	1 1/16	9/16	½	7/16
8	10½	11	2½	2¼	⅞	¾	9/16	½
10	12¾	13¼	2⅝	2⅜	1	⅞	⅝	9/16
12	15⅛	15¾	2¾	2½	1 3/16	1 1/16	¾	1 1/16
15	18⅝	19¼	2⅞	2⅝	1½	1⅜	15/16	⅞
18	22¼	23	3	2¾	1⅞	1¾	1⅛	1 1/16
21	25⅞	26¾	3¼	3	2¼	2	1 5/16	1 3/16
24	29⅜	30⅜	3⅜	3⅛	2½	2¼	1½	1⅜
30	36½	37¾	3⅝	3⅜	3	2¾	1⅞	1¾
36	43¼	44¾	4	3¾	3½	3¼	2 1/16	1⅞

[1] There is no limit for plus variation.
NOTE.—The average actual inside diameters of pipe having the nominal thickness of barrel shown in table may be smaller than the nominal sizes.

Reinforced-concrete Sewer Pipe.—ASTM Specification C-75 covers steel wire or bar reinforced-concrete sewer pipe, sizes 24 to 108 in., inclusive. Pipe conforming to this specification is known as "standard reinforced-concrete sewer pipe."

Concrete Irrigation Pipe.—ASTM Specification C118 covers standard concrete irrigation pipe sizes 6 to 24 in., inclusive. The shell thickness, test requirements, and maximum head under which properly designed lines should be operated are specified. The

maximum head varies from 35 ft for the 6-in. size to 20 ft for sizes 16 to 24 in., inclusive.

Drain Tile.—ASTM Specification C4 covers farm, standard, and extra-quality drain tile made of clay or concrete. Freezing and thawing, absorption, and strength tests are provided to check physical properties of the several classes of drain tile.

CEMENT-ASBESTOS AND REINFORCED-CONCRETE PRESSURE PIPE

In addition to the types of portland-cement pipe for gravity or low-pressure flow listed under Concrete and Clay Pipe, there are two other general types suitable for medium-pressure water-supply and distribution systems (see also Chap. XII on Water-supply Piping) which are used extensively in their respective size ranges. Among the advantages of cement-asbestos and reinforced-concrete pressure pipe are freedom from the sort of corrosion and tuberculation associated with ferrous pipe, an immunity that helps in maintaining a favorable friction factor and extending the useful life of the pipe (see pages 269 to 288).

Cement-asbestos Pipe.—The first cement-asbestos pipe[1] was manufactured in 1913 in Genoa, Italy. It has since become available in this country under the trade names of Transite, Century, and Eternit in sizes from 3 to 36 in. and in pressure classes of 50, 100, 150, and 200 psi. Test pressures range from 2½ to 4 times the rated working pressure, depending on the practice of different manufacturers. This is said to be 40 to 50 per cent of the actual bursting pressure. Standard lengths are 5 and 10 ft for the 2- to 3½-in.-diameter sizes, and 13 ft for the 4 in. and larger sizes. There is no generally accepted standard for wall thickness and each manufacturer uses his own schedule which is based on the strength of his material. Since the nominal inside diameters are uniform, the products of different manufacturers have somewhat different outside diameters and the couplings are not necessarily interchangeable.

Cement-asbestos pipe is made with plain ends and can be cut to any desired length or angle with a carpenter's saw, or tapped for threaded service connections. Line joints are made by pulling couplings of cement-asbestos or metal (see Fig. 9) over rubber gasket rings, which effectively seal the joints and can be applied

[1] See "Transite Pipe," by G. W. Blakeley, *Jour. NEWWA*, September, 1937, Vol. LI, p. 317.

either in a dry trench or under water. These couplings serve as expansion joints but require stops or other means of preventing complete disengagement if the pipe is used aboveground. Up to 5 deg. angularity is possible at each joint in laying curved portions of lines. Joints in cement-asbestos pipe for conveying sewage sometimes are made up with special couplings into which melted asphalt is poured as a sealer.

Cement-asbestos pipe has high resistance to corrosion, is not affected by electrolysis, is light in weight and easy to handle, being

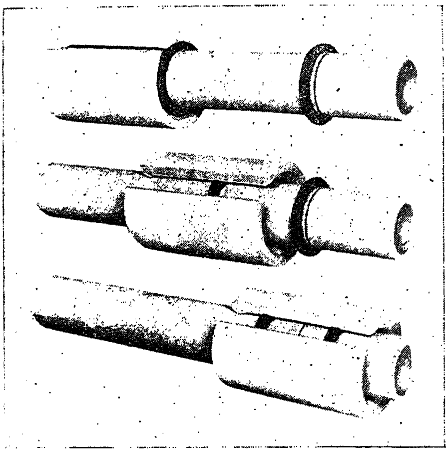


FIG. 9.—Cut-away views of coupling and rubber gasket rings used with cement-asbestos pipe showing how the rings are compressed between the coupling and pipe. Top: rings and coupling at start of operation. Center: coupling pulled over one ring. Below: final position, coupling centered over joint between pipes.

about 25 per cent of the weight of pit-cast cast-iron pipe, is easily cut and fitted but not easily broken in handling, is strong to resist external loads, is easily drilled and tapped for service connections, and possesses a Hazen and Williams hydraulic coefficient as high as 140, which does not reduce materially with age. The joints can be made without specially skilled labor and are flexible so that no expansion elements are needed. Owing to these properties cement-asbestos pipe is used extensively in sizes up to and including 24 in.

In the absence of a line of cement-asbestos fittings, it is the usual practice to use double-bell cast-iron fittings and make up these

joints as with cast-iron pipe using sulphur base compounds or lead. Cement-asbestos pipe is sold to manufacturer's specifications or to Federal Specification SS-P-351 of which an abstract follows.

Federal Specification SS-P-351 for PIPE: ASBESTOS-CEMENT

Abstracted[1]

Sizes and Pressures.—This specification covers cement-asbestos pipe in nominal inside diameters from 4 to 36 in. inclusive and for Classes 100, 150, and 200 suitable for 100, 150, and 200 psi maximum working pressure, respectively. With each length of pipe shall be furnished one complete coupling suitable for the specified pressure and pipe size. The couplings shall provide tight joints when subjected to 2½ times the designated working pressure of the pipe when pipe and couplings are restrained to maintain proper relative positions. Each length of pipe shall be tested under a hydrostatic pressure of 2½ times the maximum working pressure for its class. A further test shall be made on representative samples to determine the ultimate bursting strength of the pipe for the purpose of information.

Lengths.—The nominal length of pipe shall be 13 ft or more. Unless otherwise specified, not less than 90 per cent of the pipe furnished shall be within ± 1 in. of the specified nominal length and the remaining 10 per cent shall be not more than 4 ft under the nominal length.

Inside Diameter and Wall Thickness.—The wall thickness of the end portions of pipes within the coupling assembly area shall be in accordance with the manufacturer's standard. The inside diameter shall conform to the nominal size subject to the manufacturer's tolerances. The out-of-round tolerances applying to the nominal inside diameters of each length of pipe shall be as follows:

OUT-OF-ROUND TOLERANCES

Nominal Inside Diameter, Inches	Allowable Variation Plus or Minus, Inches
4 to 10, inclusive	0.06
12 to 16, inclusive	0.08
18 to 24, inclusive	0.10
30 to 36, inclusive	0.12

Flexure and Crushing Tests.—One 13-ft or longer length from each 100 lengths or fraction thereof of each class and size of pipe 4 to 8 in. inclusive shall be tested in *flexure*. The supports shall be 12 ft apart and the total load shall be applied equally at the third points. Each length of pipe so tested shall be capable of supporting the following total applicable load:

[1] Includes Amendment 2 of Jan. 14, 1942. Copies of complete specification and latest revisions can be obtained from the Superintendent of Documents, Washington, D.C.

MINIMUM LOADS FOR FLEXURE TEST, TOTAL APPLIED LOAD, POUNDS

Nominal size	Class 100	Class 150	Class 200
4	750	850	950
4½	950	1,050	1,250
5	1,150	1,350	1,650
6	1,650	2,100	2,600
7	2,300	3,000	3,800
8	2,900	4,000	5,400

From each 300 lengths of pipe, or fraction thereof, one specimen length of pipe shall be selected for a *crushing* test. From each specimen pipe, an unfinished section of pipe 1 ft long shall be cut and tested by the three edge bearing method described in SS-P-351 without failing before the total applied load exceeds the tabular value given in the accompanying table for that size and class of pipe.

MINIMUM APPLIED LOADS FOR CRUSHING TEST

Nominal size, I D., in.	Minimum total applied loads, lb per ft			Nominal size, I.D., in.	Minimum total applied loads, lb per ft		
	Class 100	Class 150	Class 200		Class 100	Class 150	Class 200
4	4,100	5,000	6,300	14	3,300	6,800	10,700
4½	3,900	4,800	6,400	16	3,700	7,200	12,300
5	3,600	4,700	6,500	18	4,000	8,000	14,100
6	3,200	4,600	6,700	20	4,300	8,700	15,800
7	2,900	4,600	7,000	24	5,000	10,300	18,600
8	2,500	4,400	7,400	30	6,000	13,000	23,500
10	2,600	5,300	8,700	36	7,100	16,200	28,200
12	2,900	5,900	9,300				

Marking.—The trade name, nominal size, and class of each length of pipe and coupling shall be stamped on its outside surface.

Reinforced-concrete Pipe.—In one form or another, reinforced-concrete pressure pipe has been in use since 1914 for long conduits and aqueducts, but it is not used extensively for distribution systems.[1] Its field is chiefly for large-diameter underground lines in the size range from 20 to 150 in. in diameter and for pressure services ranging up to 600 psi, depending on the kind of reinforcement. Generally speaking, there are three classes of such pipe. The first, known as *bar-type* pipe, is reinforced only with wire, bars, or welded fabric, and is designed primarily for low pressures.

[1] See "Manufacture and Construction of 48-inch Lock-joint Pipe Line," by F. F. Longley, *J. New Eng. Water Works Assoc.*, 1936, Vol. 49, p. 212.

Bar-type pipe can be cast either by pouring the concrete in vertical molds or by centrifugally casting in rotating horizontal molds. Circumferential reinforcement consists of one or two cages, depending upon the diameter, internal pressure, and external loading conditions. The pipes are reinforced longitudinally against beam loading and the longitudinals are welded to the steel rings that form the ends of the pipe.

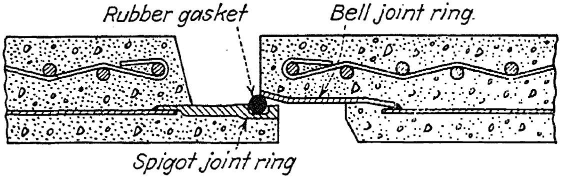
The second, known as the *cylinder type*, is reinforced with a continuous welded steel core surrounded by a cage of reinforcing bars or wire. The joint rings are welded to the ends of the steel cylinder and, before casting the concrete, this cylinder is hydraulically tested for strength and watertightness. This type of pipe is cast in vertical molds, the concrete lining of the cylinder and the exterior concrete being poured simultaneously. In all vertically cast pipes, the concrete is vibrated during the pouring to obtain the maximum strength and density. This class is used for medium to high pressures, although in certain conditions it is desirable even for low-pressure work where a high degree of tightness is wanted.

The third class, known as *prestressed concrete cylinder pipe*, also is reinforced with a continuous welded steel cylinder, but in addition has a high-tensile wire wrapped around the cylinder under considerable stress. This type of pipe is designed for medium to high pressures in diameters of from 20 in. to approximately 42 in.

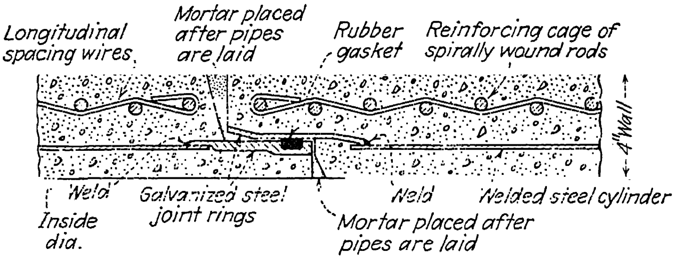
In making prestressed concrete cylinder pipe, the steel cylinder is tested first for strength and watertightness and then lined centrifugally with concrete. After curing, the lined cylinders are wrapped with high-tensile wire under considerable stress so as to put the concrete lining in compression. The pipes are then covered with a sprayed-on mortar coating to protect the exterior of the cylinder and the wire from corrosion. This process of manufacturing pipe lends itself particularly to the smaller diameters of pipe, and effects a very considerable saving of steel for high-pressure work.

Reinforced pressure pipe, regardless of type or design, is manufactured with a watertight flexible expansion sleeve at every joint. The joints are of the bell-and-spigot type (see Fig. 10) with the joint surfaces formed by steel rings cast in the ends of the pipe. In the bar-type pipe these rings are joined by the longitudinal reinforcement extending through the pipe. In the cylinder-type pipe the rings are welded to the ends of the cylinder so as to form a continuous watertight core throughout the length of each pipe.

On the rubber-gasket joint, there is a groove on the spigot ring wherein a continuous ring gasket is placed: This gasket is compressed into the groove by the flared portion of the bell as the pipes are pushed together. On the lead-gasket joint, a wedge-shaped opening is formed between the joint surfaces and a lead gasket is calked into this from the interior of the pipe after it has been



(a)
Relative position of joint and gasket prior to forcing spigot into bell



(b)
Completed joint showing details of construction of pipe wall

FIG. 10.—Steel-cylinder reinforced concrete pipe with rubber-gasket expansion joint. (a) Relative position of joint and gasket prior to forcing spigot ino bell. (b) Completed joint showing details of construction of pipe wall.

laid in the trench. In both types of joints, the shape and dimensions of the rings are such as to make the joints self-centering even without the presence of the gasket, so that the weight of one pipe can be transmitted to the adjoining pipe only by the metal contact of the rings. This prevents undue distortion of the gasket as the pipes move or deflect due to expansion and contraction or settlement. Cement mortar is placed in the spaces between the ends of the pipe both inside and outside of the joint merely for the purpose of further protecting the steel joint rings. This mortar serves no function in respect to rendering the joint watertight.

In the absence of a generally accepted standard for this product, the AWWA on Jan. 19, 1943, approved for publication the AWWA Tentative Emergency Specifications for Reinforced Concrete Pressure Pipe which is abstracted on succeeding pages.

AWWA Tentative Emergency Specifications for REINFORCED CONCRETE PRESSURE PIPE

Abstracted[1]

These specifications cover the three general types of concrete pressure pipe shown in the accompanying table.

(Table 1 of AWWA Specification)

Type	Designation	Size range, inches	Min design head, feet	Max operating head, feet
A	Steel-cylinder reinforced	20 to 150	100	600
B	Noncylinder reinforced	20 to 150	50	100
C	Centrifugal reinforced	6 to 84	50	150

The three types are not alternates, and the type should be chosen that is applicable to the individual installation. These specifications cover pipes made by experienced manufacturers and are intended for use under field conditions wherein the fill over the pipe is 5 ft or less and the pipe is to be laid in a flat-bottom trench with backfilling tamped as described in ASA A21.1-1939, Manual for the Computation of Strength and Thicknesses of Cast-Iron Pipe (see page 430). If conditions of laying are different from those stated, the need for additional reinforcing or a concrete cradle should be investigated.

Lengths.—Straight pipe, in general, shall be of the lengths shown in the following table while angle pipes, reducers, and other fittings may be made in shorter lengths.

LAYING LENGTHS FOR REINFORCED-CONCRETE PRESSURE PIPE

Type	Size range, inches	Laying length, feet
A	All sizes	12 or 16
B	All sizes	12 or 16
C	6 to 12	Not less than 6
C	14 to 84	Not less than 8

Rubber-gasket Joint.—For Types A and B pipe the ring forming the bell end of the pipe shall be covered on its exterior surface with reinforced concrete and

[1] For complete information or latest revisions, copies of this specification should be obtained from the American Water Works Association, 500 Fifth Ave., New York 18, N.Y.

the ring forming the spigot end shall be lined on its inner surface with concrete. The joint ring shall be so shaped as to enclose the gasket on all sides. The joints shall be self-centering and the rings forming the joints shall be of such shape and dimensions that the pipes will center themselves without the aid of the rubber gasket.

The joint shall be so designed and the gasket so placed that the gasket will not be required to support the weight of the pipe but will keep the joint tight under all conditions of service, including expansion, contraction, and normal earth settlement.

The gasket sealing the joint shall be made of rubber of special composition having a texture to assure a watertight and permanent seal and shall be the product of a manufacturer having experience in the manufacture of rubber gaskets for pipe joints. The gasket shall be a continuous ring, of approved cross section and of such size as completely to fill the groove on the spigot joint ring when the pipes are laid. The rubber gasket shall be the sole element depended upon to make the joint watertight. Cement mortar or plastic materials used to complete the joint making shall not be depended upon for watertightness.

Lead-gasket Joint.—For Types A and B pipe the bell end of the pipe shall be formed by the projection of the steel ring, covered on its exterior surface by reinforced concrete. The spigot end shall be formed by a ring having a ridge and covered on its inner surface with reinforced concrete. The joint shall be so designed that, when the two pipes are placed together, there will be a wedge-shaped recess between the two joint rings, into which recess there shall be placed and calked a continuous lead-covered fiber-filled gasket of approved design and manufacture. The design shall be such that the lead gasket can be calked from the inside of the pipe after the pipe is laid and backfilled and after all settlement has taken place.

Collar-type Joint.—Unless otherwise permitted or required for Class C pipe, a reinforced collar not less than 8 in. in length shall be attached to the bell end

TABLE XXXIII.—INSIDE DIAMETERS AND MINIMUM WALL THICKNESS FOR AWWA REINFORCED-CONCRETE PRESSURE PIPE

Inside diameter, inches	Minimum wall thickness, inches			Inside diameter, inches	Minimum wall thickness, inches		
	Type A	Type B	Type C		Type A	Type B	Type C
6 to 18	. . .	. . .	1¾	46	4½	4½	
20	3⅛	3⅛	. . .	48	4½	5	4⅛
21	. . .	. . .	2	51	5	5	4¼
24	3½	3½	2½	54	5	5½	4½
27	3½	3½	2⅝	57	. . .	. . .	4¾
30	3½	3½	2¾	60	5½	6	5
33	. . .	. . .	2⅞	63	. . .	. . .	5¼
34	3½	3½	. . .	66	6	6½	5½
36	3½	4	3⅛	69	. . .	. . .	5¾
39	4	4	3½	72	6½	7	6
42	4	4½	3¾	78	7	7½	
45	. . .	. . .	3⅞	84	7½	8	

of the pipe. The internal diameter of the collar shall be such that the space between the inside of the collar and the outside of the pipe shall be not less than $5/8$ in. nor more than $3/4$ in.

One-half of the length of the collars shall be calked or spun onto the pipe in the yard. If it is calked on, a suitable collar support shall be used to hold the collar concentric with the pipe and to furnish an unyielding surface against which to calk. Neat cement shall be used in calking on the collars and shall be slightly moistened and screened through a $3/16$-in. screen to break up the small lumps. The cement shall be calked in layers not exceeding $1\frac{1}{2}$ in. in thickness with suitable tools and a hammer weighing not less than 3 lb. After the calking is completed, the joint shall be covered and kept moist for at least three days.

Wall Thickness.—The respective classes of pipe shall have the inside dimensions and minimum wall thicknesses shown in Table XXXIII.

Methods of Manufacture and Tests.—For requirements as to quality of materials, methods of manufacture, and tests, reference should be made to the AWWA Specification.

Marking.—Each special and straight pipe shall have plainly marked on it the head for which it is designed, the date of manufacture, and marks of identification to show its proper location in the line.

WOODEN PIPE

Early American water-supply pipes were made by *boring holes* through solid *wood logs* which were joined together in place by push nipples of wood or metal. At the present time *wood-stave*[1] pipe made of redwood or Douglas-fir staves held together by metal hoops is still used extensively in the West which is close to the source of timber and remote from the iron and steel manufacturing centers. At one time it was used in the Midwest and Eastern states. Such pipe is made chiefly in large diameters for hydraulic-power developments, municipal water supplies, outfall sewers, mining, irrigation, and various other purposes requiring the transportation of water. The water carried may be hot, cold, or acid. Redwood and Douglas fir are used principally because of their availability and good resistance to decay. Wood staves for pipe may be either untreated or creosoted by a vacuum-and-pressure process with about 8 lb of creosote per cubic feet of wood treated. The maximum pressure at which wood-stave pipe may be used depends on the size of the pipe, ranging from 25 psi for large pipe up to 200 psi for small pipe. Machine-made pipe banded with wire is available in sizes from 2 to 32 in., and continuous-stave pipe from 6 in. to 20 ft. in diameter. The latter is shipped in knocked down condition and assembled in the field

[1] See "Conveyance and Distribution of Water for Water Supply," by Edward Wegmann, D. Van Nostrand Company, Inc., New York, 1918.

so as to break joints and eliminate the need for couplings. The staves are banded together by steel hoops having threaded ends which are passed through a cast or malleable-iron shoe and pulled up with nuts. The steel hoops ordinarily are designed for a factor of safety of 4 with respect to fluid pressure and the tensile strength of the steel. Smooth wood pipe with well-matched staves has a William-Hazen C coefficient of about 120 (see pages 276 to 288) which holds up well throughout the life of the pipe.

ASTM Standard Specifications for BRASS PIPE, STANDARD SIZES

Serial Designation B43-42

Abstracted[1]

These specifications cover seamless muntz metal, high brass, admiralty metal, and red brass pipe in all standard pipe sizes. Brass pipe is suitable for use in plumbing, boiler-feed lines and other services. The chemical compositions of the various alloys are given in the following tabulation:

COMPOSITION OF BRASS PIPE

	Muntz metal	High brass	Admiralty metal	
Copper, per cent	59.00 to 63.00	65.00 to 68.00	70.00 to 73.00	83.00 to 86.00
Lead, max, per cent	0.50	0.80	0.05	0.06
Iron, max, per cent	0.07	0.07	0.06	0.05
Tin, per cent	....	....	0.90 to 1.20	0.15 max
Zinc, per cent	Remainder	Remainder	Remainder	Remainder

Brass pipe is cold-drawn to size and is normally furnished in the annealed condition. The degree of anneal shall be sufficient to show complete recrystallization and to meet test requirements. If agreed upon between the manufacturer and the purchaser, red-brass pipe may be furnished in the hard-drawn condition. Annealed pipe shall withstand an expansion of 25 per cent of the inside diameter by a tapered pin having an included angle of 60 deg. Pipe shall withstand an immersion of 15 min in an aqueous solution of mercurous nitrate without cracking. When required for bending, annealed full sections of the pipe shall stand being bent cold through 180 deg around a pin 1½ times the inside diameter of the pipe without cracking. Each length of pipe shall be given a hydrostatic test which will develop 7,000 psi as computed by Barlow's formula (see page 41), except that no pipe shall be tested beyond 1,000 psi hydrostatic pressure unless so specified. The weight of the pipe shall not vary from the nominal weight per foot given in Table XXXIV by more than the following percentages: 6 in. size and smaller, 5 per cent; over 6 to 8 inclusive, 7 per cent; over 8 in., 8 per cent. The thickness at any point shall not be less than prescribed in Table XXXIV by

[1] For complete specification, reference may be made to ASTM B43 (see note, p. 369).

these same tolerances. Random mill lengths are from 7 to 14 ft. In the case of specified straight lengths with ends included, at least 75 per cent by weight shall be of the length specified, 25 per cent by weight may be shorter, but no piece shall be less than 5 per cent of the length specified. All pipes shall be bright annealed or acid cleaned after final annealing. Lengths shall conform to the tolerances prescribed in the accompanying table.

TABLE XXXIV.—DIMENSIONS AND WEIGHTS OF SEAMLESS BRASS AND COPPER PIPE (IRON-PIPE SIZE)
(Table I, ASTM Specifications B42-43 and B43-42)

Size of pipe, inches	Outside diameter, inches	Regular pipe				Extra-strong pipe			
		Thickness, inches	Nominal weight, pounds per foot of length			Thickness, inches	Nominal weight, pounds per foot of length		
			Yellow brass	Red brass	Copper		Yellow brass	Red brass	Copper
⅛	0.405	0.062	0.246	0.253	0.259	0.100	0.353	0.363	0.371
¼	0.540	0.082	0.435	0.447	0.457	0.123	0.594	0.611	0.625
⅜	0.675	0.090	0.609	0.627	0.641	0.127	0.805	0.829	0.847
½	0.840	0.107	0.908	0.934	0.955	0.149	1.19	1.23	1.25
¾	1.050	0.114	1.23	1.27	1.30	0.157	1.62	1.67	1.71
1	1.315	0.126	1.73	1.78	1.82	0.182	2.39	2.46	2.51
1¼	1.660	0.146	2.56	2.63	2.69	0.194	3.29	3.39	3.46
1½	1.900	0.150	3.04	3.13	3.20	0.203	3.99	4.10	4.19
2	2.375	0.156	4.01	4.12	4.22	0.221	5.51	5.67	5.80
2½	2.875	0.187	5.82	5.99	6.12	0.280	8.41	8.66	8.85
3	3.500	0.219	8.32	8.56	8.75	0.304	11.2	11.6	11.8
3½	4.000	2.250	10.9	11.2	11.4	0.321	13.7	14.1	14.4
4	4.500	0.250	12.3	12.7	12.9	0.341	16.4	16.9	17.3
4½	5.000	0.250	13.7	14.1	14.5	0.375	20.1	20.7	21.1
5	5.563	0.250	15.4	15.8	16.2	0.375	22.5	23.2	23.7
6	6.625	0.250	18.4	19.0	19.4	0.437	31.3	32.2	32.9
7	7.625	0.281	23.9	24.6	25.1	0.500	41.2	42.4	43.4
8	8.625	0.312	30.0	30.9	31.6	0.500	47.0	48.4	49.5
9	9.625	0.344	37.0	38.0	38.9	0.500	52.8	54.4	55.6
10	10.750	0.365	43.9	45.2	46.2	0.500	59.3	61.1	62.4
11	11.750	0.375	49.4	50.8	51.9				
12	12.750	0.375	53.7	55.3	56.5				

NOTE.—For inside diameter functions of IPS copper and brass pipe, see those for steel pipe on pages 357 to 367.

ASTM Tentative Specifications for COPPER PIPE, STANDARD SIZES

Serial Designation B42-43

Abstracted[1]

These specifications cover seamless copper pipe in all standard pipe sizes. Copper pipe is suitable for use in plumbing, boiler-feed lines, and for similar

[1] For complete specifications, reference may be made to ASTM B42 (see note, p. 369).

purposes. For weights and dimensions see Table XXXIV. Copper content is required to be not less than 99.90 per cent, silver counting as copper, and phosphorus not more than 0.040 per cent. Copper pipe is cold drawn to size and is normally furnished in the hard-drawn condition. When required for bending, the pipe is furnished in the temper agreed upon between the manufacturer and the purchaser. Annealed pipe shall withstand an expansion of 25 per cent of the inside diameter by a tapered pin having a 60-deg included angle. Annealed full sections of pipe 2 in. and under in outside diameter, when required for bending, shall stand being bent cold through 180 deg around a pin 1½ times the inside diameter of the pipe without cracking. Microscopic examination at 75X shall show that pipe is made from copper free of cuprous oxide. Each length of pipe shall be given a hydrostatic test which will develop a stress of 6,000 psi as computed by Barlow's formula (see page 41), except that no pipe shall be tested beyond 1,000 psi hydrostatic pressure unless so specified. The weight of the pipe shall not vary from the nominal weight per foot given in the Table XXXIV by more than the following percentages: 6-in. size and under, 5 per cent; over 6 to 8 in. inclusive, 7 per cent; over 8 in., 8 per cent. The thickness at any point shall not be less than prescribed in Table XXXIV by these same tolerances. Random mill lengths are from 7 to 14 ft. In the case of specified straight lengths with ends included at least 75 per cent by weight shall be the length specified, 25 per cent by weight may be shorter, but no piece shall be less than 50 per cent of the length specified. The major and minor diameters of drawn unannealed straight pipe shall not vary more than 1 per cent from the specified diameter. No tolerance for out-of-roundness has been established for annealed pipe. Lengths shall conform to the tolerances prescribed in the accompanying table. For closer tolerances, the manufacturer should be consulted.

TOLERANCES IN INCHES FROM SPECIFIED LENGTHS
(Table II, ASTM Specifications B42 and B43)
(All tolerances are plus)

Outside diameter, in.	Straight lengths				
	6 in. and under	Over 6 in. to 2 ft	Over 2 to 6 ft	Over 6 to 14 ft	Over 14 ft
1 and under	1/32	1/16	3/32	1/4	1/2
Over 1 to 4, incl.	1/16	3/32	1/8	1/4	1/2
Over 4	..	1/8	1/4	1/4	1/2

ASTM Standard Specifications for COPPER WATER TUBE

Serial Designation B88-41

Abstracted[1]

These specifications cover seamless copper tubes especially designed for plumbing purposes, underground water services, etc., but also suitable for copper-

[1] For complete specification, reference may be made to ASTM B88 (see note, p. 369).

TABLE XXXV.—STANDARD DIMENSIONS AND WEIGHTS, AND TOLERANCES IN DIAMETER AND WALL THICKNESS OF COPPER WATER TUBES
(Table II, ASTM B88-41)

Standard water tube size, inches	Actual outside diameter, inches	Average outside diameter Tolerance, inches		Wall thickness, inches						Theoretical weight, pounds per foot		
				Type K		Type L		Type M				
		Annealed	Drawn	Nominal	Tolerance	Nominal	Tolerance	Nominal	Tolerance	Type K	Type L	Type M
1/8	0.250	0.002	0.001	0.032	0.003	0.025	0.0025	0.025	0.0025	0.085	0.068	0.068
1/4	0.375	0.002	0.001	0.032	0.004	0.030	0.0035	0.025	0.0025	0.134	0.126	0.107
3/8	0.500	0.0025	0.001	0.049	0.004	0.035	0.0035	0.025	0.0025	0.269	0.198	0.145
1/2	0.625	0.0025	0.001	0.049	0.004	0.040	0.0035	0.028	0.0025	0.344	0.285	0.204
5/8	0.750	0.0025	0.001	0.049	0.004	0.042	0.0035	0.030	0.0025	0.418	0.362	0.263
3/4	0.875	0.003	0.001	0.065	0.0045	0.045	0.004	0.032	0.003	0.641	0.455	0.328
1	1.125	0.0035	0.0015	0.065	0.0045	0.050	0.004	0.035	0.0035	0.839	0.655	0.465
1 1/4	1.375	0.004	0.0015	0.065	0.0045	0.055	0.0045	0.042	0.0035	1.04	0.884	0.682
1 1/2	1.625	0.0045	0.002	0.072	0.005	0.060	0.0045	0.049	0.004	1.36	1.14	0.940
2	2.125	0.005	0.002	0.083	0.007	0.070	0.006	0.058	0.006	2.06	1.75	1.46
2 1/2	2.625	0.005	0.002	0.095	0.007	0.080	0.006	0.065	0.006	2.93	2.48	2.03
3	3.125	0.005	0.002	0.109	0.007	0.090	0.007	0.072	0.006	4.00	3.33	2.68
3 1/2	3.625	0.005	0.002	0.120	0.008	0.100	0.007	0.083	0.007	5.12	4.29	3.58
4	4.125	0.005	0.002	0.134	0.010	0.110	0.009	0.095	0.009	6.51	5.38	4.66
5	5.125	0.005	0.002	0.160	0.010	0.125	0.010	0.109	0.009	9.67	7.61	6.66
6	6.125	0.005	0.002	0.192	0.012	0.140	0.010	0.122	0.010	13.9	10.2	8.92
8	8.125	0.006	+0.002 −0.004	0.271	0.016	0.200	0.014	0.170	0.014	25.9	19.3	16.5
10	10.125	0.008	+0.002 −0.006	0.338	0.018	0.250	0.016	0.212	0.015	40.3	30.1	25.6
12	12.125	0.008	+0.002 −0.006	0.405	0.020	0.280	0.018	0.254	0.016	57.8	40.4	36.7

NOTES.—For copper gas and oil burner tubes, the tolerances shown above for various wall thicknesses (Type K) apply irrespective of diameter.
For tubes other than round no standard tolerances are established.
These tolerances do not apply to condenser and heat exchanger tubes. All tolerances in this table are plus and minus except as otherwise indicated.
For inside diameter functions of Type K copper water tubes, see p. 364.

coil water heaters, fuel-oil lines, gas lines, etc. For refrigeration and air conditioning, hard tubes are recommended for sweat fittings. If soft tube is made up with sweat fittings, rounding and sizing tools should be used.

Three thicknesses of copper water tube designated by types are covered:

Type K for underground services and general plumbing purposes.

Type L for general plumbing purposes.

Type M for use with soldered fittings only.

Types K and L tubes when furnished in coils shall be "annealed" after coiling. When furnished in straight lengths, they normally shall be "hard-drawn" but may be furnished "annealed" or "light-drawn" if so specified within the size limitation of ⅜ to 1¼ in. inclusive for the "light-drawn" condition. Type M tubes shall be furnished in straight lengths "hard-drawn."

The copper tubes shall have not less than 99.90 per cent copper, silver counting as copper, and phosphorus shall not exceed 0.04 per cent. The physical properties and grain size shall conform to the following requirements:

TENSILE STRENGTH AND GRAIN SIZE REQUIREMENTS

Temper	Size	Tensile strength, psi	Elongation in 4 in., min, per cent	Average grain size, min, mm
Annealed	All sizes	30,000 min	25	0.040
Light-drawn[a]	⅜ to 1¼ in., inclusive	36,000 to 50,000		
Hard-drawn	All sizes	48,000 min		

[a] Light drawn temper applies only to Types K and L tubing.

Standard dimensions, weights, wall thicknesses, and tolerances in diameter and wall thicknesses are given in Table XXXV. It may be noted that the outside diameters of copper water tubes are uniformly ⅛ in. larger than the nominal tube size, *i.e.*, the outside diameter of a 1-in. tube is 1⅛ in.

The nominal length for tubes furnished straight is 20 ft. The nominal lengths for tubes furnished in coils are 30, 45, and 60 ft and, for tubes up to 1 in. inclusive, 100 ft.

Fittings for soldered joints for use with copper water tubes are illustrated in cuts of Table LXXVIII page 590, which gives roughing-in dimensions for these fittings. Fittings for flared copper water tubes are covered by ASA A40.2 (see abstract, page 588), and Table LXXVII.

LEAD PIPE

Lead pipe is used for service connections between water mains and meters in water-supply systems and for plumbing. The Lead Industries Association[1] working in conjunction with the National Bureau of Standards of the U.S. Department of Commerce has developed Commercial Standard CS 95-41 for Lead Pipe.[2] This

[1] The Lead Industries Association, 420 Lexington Ave., New York 17, N.Y.

[2] U.S. Commercial Standards may be obtained from the Superintendent of Documents, Washington, D.C.

TABLE XXXVI.—LEAD PIPE SIZES
(Table 1, U.S. Commercial Standard, CS 95-41)

Classification		Outside diameter, in.	Wall thickness, in.	Weight per foot, lb	Working pressure, psi	Outside diameter, in.	Wall thickness, in.	Weight per foot, lb	Working pressure, psi	Outside diameter, in.	Wall thickness, in.	Weight per foot, lb	Working pressure, psi
East[1]	West[2]												
		3/8 in. nominal inside diameter				1/2 in. nominal inside diameter				3/4 in. nominal inside diameter			
D	XL	0.549	0.087	0.62	Waste	0.666	0.083	0.75	Waste	0.940	0.095	1.25	Waste
C	L	0.577	0.101	0.75	Waste	0.712	0.106	1.00	Waste	1.006	0.128	1.75	Waste
B	M	0.631	0.128	1.00	Waste	0.756	0.128	1.25	Waste	1.068	0.159	2.25	Waste
A	S	0.725	0.175	1.50	50	0.798	0.149	1.50	50	1.156	0.203	3.00	50
AA	XS	0.811	0.218	2.00	75	0.876	0.188	2.00	75	1.212	0.231	3.50	75
AAA	XXS	0.888	0.256	2.50	100	1.012	0.256	3.00	100	1.336	0.293	4.75	100
		1 in. nominal inside diameter				1½ in. nominal inside diameter				2 in. nominal inside diameter			
D	XL	1.232	0.116	2.00	Waste	1.776	0.138	3.50	Waste	2.284	0.142	4.75	Waste
C	L	1.284	0.142	2.50	Waste	1.830	0.165	4.25	Waste	2.354	0.177	6.00	Waste
B	M	1.356	0.178	3.25	Waste	1.882	0.191	5.00	Waste	2.410	0.205	7.00	Waste
A	S	1.428	0.214	4.00	50	1.984	0.242	6.50	50	2.503	0.252	8.75	50
AA	XS	1.492	0.246	4.75	75	2.076	0.288	8.00	75	2.751	0.376	13.75	75
AAA	XXS	1.596	0.298	6.00	100	2.272	0.386	11.25	100	3.008	0.504	19.50	100

[1] Symbols used generally for lead pipe sold in cities east of the Illinois-Indiana line.
[2] Symbols used generally for lead pipe sold in cities west of the Illinois-Indiana line.
NOTE.—The 5/8, 1¼, and 1¾ in. sizes are not included in above table. Sizes 2½ to 6 in. inclusive are classified by nominal weight per foot, or by wall thickness.

standard covers chemical composition, diameters, weight classification, weight per foot, defects, and labeling of lead pipe. Lead pipe is sized according to the inside diameter, and the outside diameter for each weight classification is obtained by adding the wall thicknesses to the nominal inside diameter (see Table XXXVI). Pipe smaller than 2 in. is furnished in coils usually not exceeding 200 lb. Pipe 2 in. and larger is furnished in 10-ft lengths. Dimensions, weights, and composition of lead traps and bends are covered in Commercial Standard CS 96-41.

GLASS PIPING

A line of glass pipe and pipe fittings is available for use in the chemical and food industries. A borosilicate glass of low thermal expansivity is the basic material. Pipe 1, 1½, 2, and 3 in. nominal size is designed to operate with 100-psi steam pressure; the 4-in. size is limited to 75 psi. A compression type of flanged joint using loose malleable-iron flanges is available for sizes up to and including 4 in. A method of welding glass pipe using pin-point oxyhydrogen flames and high-frequency electric current has been developed for installations not requiring disassembly. A less expensive type of sanitary glass piping using standard metal fittings is recommended for the dairy industry where pressures do not exceed 50 psi. Pipe is made in lengths up to 10 ft. Nominal dimensions and weights of flanged glass pipe are given in Table XXXVII-*A*.

TABLE XXXVII-*A*.—NOMINAL DIMENSIONS AND WEIGHTS OF FLANGED GLASS PIPE[1]

Nominal size, inches	Outside diameter, inches, average	Wall thickness, inches, average	Weight per foot, pounds
1	1.312	0.156	0.55
1½	1.844	0.172	0.79
2	2.344	0.172	1.14
3	3.406	0.203	1.98
4	4.500	0.250	3.24

[1] Corning Glass Works, Corning, N.Y.

Glass pipe for low-pressure service is made with socket or bell-and-spigot ends. The principal dimensions are as given in Table XXXVII-*B*.

The coefficient of thermal expansion of glass pipe is approximately one-third that of steel. Expansion slip joints are recom-

TABLE XXXVII-*B*.—DIMENSIONS OF SOCKET-END GLASS PIPE[1]

Nominal size, inches	Outside diameter, inches	Wall thickness, inches	Inside diameter, inches	Inside diameter of socket, inches	Over-all length, inches	Laying length, inches
6½	6½	0.25	6	8⅛	36	32½
9	9⅛	0.312	8½	10⅞	42	38¹¹⁄₁₆
12	12	0.312	11⅜	13½	30	26¹¹⁄₁₆
16	16	0.312	15⅜	17⅜	27	23¹¹⁄₁₆
18	18	0.312	17⅜	19⅜	24	20¹¹⁄₁₆

1 From Corning Glass Works, Corning, N.Y.

mended to be installed in straight runs every 50 ft for 1- and 1½-in. pipe and every 100 ft for 2- and 3-in. pipe. Recommended maximum distance between supports when carrying water is about 10 ft.

PLASTIC PIPING

The shortage of copper and other corrosion-resistant materials during the war emergency accelerated the development of plastic piping. A practical thermoplastic line of pipe and fittings is offered under the trade name of Saran.[1] This pipe is produced by a modified extrusion process to the dimensions of extra-strong pipe (see page 358). At present, sizes range from ½ in. through 4 in., although larger sizes are planned. Pipe is made in 20-ft. lengths. Under room temperature conditions, or temperatures not exceeding about 120 F, published data[2] indicate that the pipe is adequate for 100-psi service. The material has very low heat conductivity and a high coefficient of expansion. Its tensile strength at 170 F is about one-half that at room temperature. Standard methods of welding, threading, and installing Saran piping have been developed. In addition to pipe, a line of tubing in sizes from ⅛ to ¾ in. outside diameter, with wall thicknesses ranging from 0.031 to 0.062 in. is produced. This is suitable for ordinary cold-water residence and similar purposes. Flared-type injection-molded Saran fittings are used to connect the tubing.

1 Plastic made by Dow Chemical Co., Midland, Mich.

2 "Plastic Tubing a 'War Baby,'" by C. B. Branch and D. L. Gibb, *Heating, Piping and Air Conditioning*, June, 1942, p. 353.

Also, "Studies Relating to Use of Saran for Water Pipes in Buildings and for Service Lines," by F. M. Dawson and A. A. Kalinske, *Jour. AWWA*, August, 1943, pp. 1058–1064.

THREADS FOR PIPE AND HOSE COUPLINGS

Thread Cutting.—For first-class pipe fitting it is necessary to have smooth, clean threads, otherwise it will be difficult to produce joints that will go together easily and that will be and remain pressure-tight. Thread cutting should be regarded as a rather precise machining.

Threading Dies.—A proper form for a threading die with four chasers is shown in Fig. 11. Perhaps the most important feature is the lip angle, that is, the angle between the front face of the chaser and the actual cutting edge. This is also sometimes termed the "rake" or "hook." For threading wrought iron, the lip angle should be not over 16 deg; for ordinary Bessemer steel pipe, it should be from 15 to 20 deg; for open-hearth steel, it should be at least 25 deg; and for brass, it should be very small. An improper lip angle produces a rough thread and is sure to cause tearing of the metal.

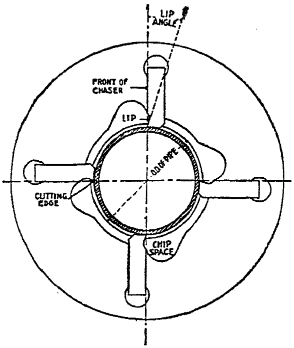


Fig. 11.—Proper form of threading die.

Another important point is the clearance, which is the space between the end or heel of the chaser and the work. Too much clearance results in chattering or a wavy thread. Too little clearance is indicated by the entire heel of the chaser showing signs of wear instead of showing a decreasing amount of wear back of the cutting edge. Too small clearance makes the die work hard. The clearance is determined by the original machining of the chaser and cannot be changed in the field.

The lead or throat is the cutting away of the first three or more threads on the chasers to enable the pipe to enter the die.

Another requirement in a well-constructed die is ample chip space (see Fig. 14). Insufficient space causes the chips to pack in and the die to work hard.

Pipe-joint Compounds.—Pipe threads are necessarily somewhat imperfect and to ensure tightness, as well as to lubricate them while being screwed up, some form of compound is used. As a lubricant, only a light oil is satisfactory and, if the threads are

very well made, this is sometimes sufficient. In plumbing work white lead or red lead is usually used. For steam piping, a prepared paste containing oil and a filling material quota is generally used. Recently, a compound containing powdered zinc has been used with much success in making tight joints. A mixture of litharge and glycerine is satisfactory for making joints tight against fluids, such as oil, which are difficult to hold, but joints made up with this compound are difficult to break.

American Standard for PIPE THREADS

ASA Standard B2.1-1942

Abstracted[1]

This standard specifies dimensions, tolerances, and gaging for taper and straight pipe threads, including certain special applications. The normal type of pipe joint employs a tapered external and tapered internal thread, but straight pipe threads are used to advantage for certain types of pipe couplings, grease cup,

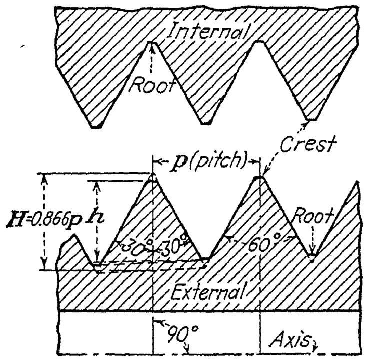


FIG. 12.—Basic form of American standard taper pipe thread.

fuel and oil fittings, mechanical joints for fixtures, conduit and hose couplings (see Fig. 12).

Form of Thread.—The angle between the sides of the thread is 60 deg when measured in an axial plane, and the line bisecting this angle is perpendicular to the axis for both taper and straight threads (see Fig. 12). The crest and root are truncated an amount equal to $0.033p$, except for 8 threads per inch which are truncated $0.045p$ at the crest and $0.033p$ at the root. The (basic) maximum

[1] For complete standard, reference may be made to ASA B2.1 (see note, p. 430).

depth of the truncated thread, h, is $0.80p$ except for 8 threads per inch where it is $0.788p$.

Pitch.—The pitch is the distance from a point on a screw thread to a corresponding point on the next thread measured parallel to the axis.

$$\text{Pitch in inches } (p) = \frac{}{\text{number of threads per inch}}.$$

Taper of Thread.—The taper of the thread is 1 in 16 or 0.75 in. per ft measured on the diameter and along the axis.

Diameter of Thread.—The pitch diameters of the taper thread are determined by the following formulas based on the outside diameter of the pipe and the pitch of the thread:

$$E_0 = D - (0.050D + 1.1)p,$$
$$E_1 = E_0 + 0.0625L_1$$

where E_0 = pitch diameter of thread at end of pipe.
E_1 = pitch diameter of thread at the gaging notch or large end of internal thread.
D = outside diameter of pipe.
L_1 = normal engagement by hand between external and internal threads.
p = pitch of thread.

Length of Thread.—The length of the effective external taper thread, L_3, is determined by the following formula based on the outside diameter of the pipe and the pitch of the thread:

$$L_3 = (0.80D + 6.8)p$$

where D = outside diameter of pipe.
p = pitch of thread.

This formula determines directly the length of effective thread which includes approximately two usable threads slightly imperfect at the crest.

Engagement between External and Internal Taper Threads.—The normal length of engagement between external and internal taper threads when screwed together by hand is shown in column 7, Table XXXVIII. This length is controlled by the construction and use of the gages. It is recognized that in special applications, such as flanges for high-pressure work, longer thread engagement is used, in which case the pitch diameter (dimension E_1 in figure over Table XXXVIII) is maintained and the pitch diameter at the end of pipe is proportionately smaller.

Manufacturing Tolerance of Threaded Product.—The variation in thread elements on steel products and all pipe made of steel, wrought iron, or brass should not exceed the following limits.

On pipe fittings and valves (not steel) for steam pressure 300 lb and below, it is intended that plug and ring gage practice as set up in this standard provide for a satisfactory check on accumulated variations in such product. Therefore no tolerances on thread elements have been established for this class.

For service conditions, where more exacting check is required, a procedure as developed by industry and found practical other than regulation plug and ring gage may be used.

For steel products and all pipe made of steel, wrought iron, or brass, the variation in thread elements should not exceed the following limits:

Taper.—Sizes ⅛ to ⅜ in., inclusive maximum taper, ⅞ in. per ft; minimum taper, 1⁄16 in. per ft.

Sizes ½ to 2 in., inclusive[1] maximum taper, 27⁄32 in. per ft; minimum taper, 11⁄16 in. per ft.

Sizes 2½ in. and larger maximum taper, 13⁄16 in. per ft; minimum taper, 23⁄32 in. per ft.

Lead.—(*a*) 27, 18, and 14 threads per inch ± 0.003 in. in length of effective thread.

(*b*) 11½ and 8 threads per inch ± 0.003 in. per in. ± 0.006 in. cumulative.

Angle.—(*a*) 27 threads per inch $\pm 2½$ deg, inclusive.

(*b*) 18 and 14 threads per inch ± 2 deg, inclusive.

(*c*) 11½ and 8 threads per inch $\pm 1½$ deg, inclusive.

Height.—Dependent on minimum and maximum truncation of crest and root (Table 2 of ASA B2.1 not reproduced here).

Pitch Diameter.—One turn large or small from gaging notch on plug or gaging face of ring when using working gages.

Pressure-tight Joints.—Pressure-tight joints for low-pressure service are sometimes made with straight internal threads and the American Standard taper external threads. The ductility of the coupling enables the straight thread to conform to the taper of the

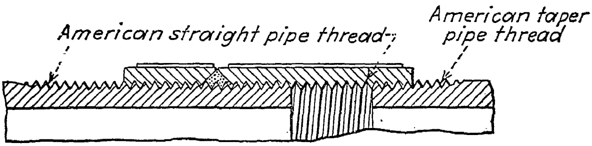


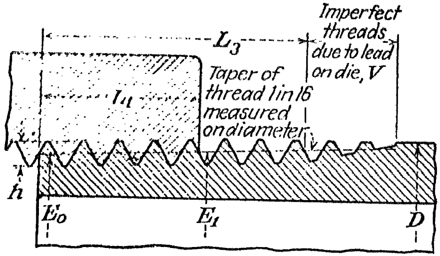
FIG. 13.—Long screw joint between straight threaded coupling and straight and taper threaded pipes.

pipe thread. It is commercial practice to furnish straight-tapped couplings for standard-weight (Schedule 40) pipe 2 in. and smaller. If taper-tapped couplings are required for standard-weight pipe sizes 2 in. and smaller, line pipe in accordance with API 5L should be ordered, thread lengths to be in accordance with the American Standard for Pipe Threads, ASA B2.1. Taper-tapped couplings are furnished on extra-strong (Schedule 80) pipe in all sizes and on standard-weight pipe 2½ in. and larger.

Couplings with straight pipe threads are sometimes used with straight external threads, in which case it is necessary to employ

[1] The tolerance on taper for 2-in. API line pipe shall be the same as that shown for sizes 2½ in. and larger.

TABLE XXXVIII.—BASIC DIMENSIONS, AMERICAN STANDARD TAPER PIPE THREAD[1]
(Table 1, ASA B2.1)



(All dimensions in inches)

1	2	3	4	5	6	7	8	9	10	11
Nominal pipe size	Outside diameter of pipe	Number of threads per inch	Pitch diameter at end of external thread	Pitch[2] diameter at end of internal thread	Length[3] of effective thread	Length[4] of hand-tight engagement	Vanish threads	Depth of thread, max	Pitch of thread	Diameter increase per turn
	D		E_0	E_1	L_3	L_1	V	h	p	
⅛	0.405	27	0.36351	0.37476	0.2638	0.180	0.1285	0.02963	0.03704	0.00231
¼	0.540	18	0.47739	0.48989	0.4018	0.200	0.1928	0.04444	0.05556	0.00347
⅜	0.675	18	0.61201	0.62701	0.4078	0.240	0.1928	0.04444	0.05556	0.00347
½	0.840	14	0.75843	0.77843	0.5337	0.320	0.2478	0.05714	0.07143	0.00446
¾	1.050	14	0.96768	0.98887	0.5457	0.339	0.2478	0.05714	0.07143	0.00446

1	1.315	11½	1.21363	1.23863	0.6828	0.400	0.3017	0.06957	0.08696	0.00543
1¼	1.660	11½	1.55713	1.58338	0.7068	0.420	0.3017	0.06957	0.08696	0.00543
1½	1.900	11½	1.79609	1.82234	0.7235	0.420	0.3017	0.06957	0.08696	0.00543
2	2.375	11½	2.26902	2.29627	0.7565	0.436	0.3017	0.06957	0.08696	0.00543
2*	2.375	11½	2.25453	2.29627	0.9884	0.668	0.3017	....	0.08696	0.00543
2½	2.875	8	2.71953	2.76216	1.1375	0.682	0.4337	0.09850	0.12500	0.00781
3	3.500	8	3.34063	3.38850	1.2000	0.766	0.4337	0.09850	0.12500	0.00781
3½	4.000	8	3.83750	3.88881	1.2500	0.821	0.4337	0.09850	0.12500	0.00781
4	4.500	8	4.33438	4.38713	1.3000	0.844	0.4337	0.09850	0.12500	0.00781
5	5.563	8	5.39073	5.44929	1.4063	0.937	0.4337	0.09850	0.12500	0.00781
6	6.625	8	6.44609	6.50597	1.5125	0.958	0.4337	0.09850	0.12500	0.00781
8	8.625	8	8.43359	8.50003	1.7125	1.063	0.4337	0.09850	0.12500	0.00781
10	10.750	8	10.54531	10.62094	1.9250	1.210	0.4337	0.09850	0.12500	0.00781
12	12.750	8	12.53281	12.61781	2.1250	1.360	0.4337	0.09850	0.12500	0.00781
14 OD	14.000	8	13.77500	13.87263	2.2500	1.562	0.4337	0.09850	0.12500	0.00781
16 OD	16.000	8	15.76250	15.87575	2.4500	1.812	0.4337	0.09850	0.12500	0.00781
18 OD	18.000	8	17.75000	17.87500	2.6500	2.000	0.4337	0.09850	0.12500	0.00781
20 OD	20.000	8	19.73750	19.87031	2.8500	2.125	0.4337	0.09850	0.12500	0.00781
24 OD	24.000	8	23.71250	23.86094	3.2500	2.375	0.4337	0.09850	0.12500	0.00781

[1] The basic dimensions of the American Standard Taper Pipe Thread are given in inches to five decimal places. While this implies a greater degree of precision than is ordinarily attained, these dimensions are so expressed for the purpose of eliminating errors in computations.

[2] Also pitch diameter at gaging notch.

[3] Also length of plug gage.

[4] Also length of ring gage and length from gaging notch to small end of plug gage.

* API Line Pipe. (Not an American Standard and tolerances in this standard do not apply.) This is the only size of line pipe that differs in length of thread from the American Standard. The standard thread chambers in the lower pressure fittings and valves do not accommodate this longer line pipe thread.

locknuts and packing when it is desired to seal the joint. This type of joint is illustrated in Fig. 13.

Hose Nipples and Couplings.—Hose coupling joints are ordinarily used with a gasket and made with straight internal and external loose-fitting threads. There are several standards of hose threads having various diameters and pitches, one of which is based on the American Standard Pipe Thread. With this thread series, it is possible to join small hose sizes ½ to 2 in. inclusive to ends of standard pipe having American Standard External Taper Pipe Threads, using a gasket to seal the joint.

American Standard
HOSE COUPLING SCREW THREADS

ASA B33.1-1935

Abstracted[1]

This standard applies to the threaded parts of hose couplings, valves, nozzles, and all other fittings used in direct connection with hose intended for fire protection or for domestic and industrial general services in sizes ½, ⅝, ¾, 1, 1¼, 1½, and 2 in.

TABLE XXXIX.—BASIC THREAD DIMENSIONS OF HOSE COUPLING SCREW THREAD
(Table 1, ASA B33.1)
(All dimensions in inches)

Service and nominal size	Number of threads per inch	Outside diameter of nipple thread, max
Garden and similar hose:		
½, ⅝, ¾		1.0625
Chemical engine and booster hose:		
¾, 1		1.3750
Fire protection hose:		
1½		1.9900
Steam, water, air, [illegible] and all other hose connections:		
½	14	0.8248
¾	14	1.0353
1	11½	1.2951
1¼	11½	1.6399
1½	11½	1.8788
2	11½	2.3528

[1] For complete standard, reference may be made to ASA B33.1 (see note, p. 430).

American Standard
FIRE-HOSE COUPLING SCREW THREAD

ASA B26-1925

Abstracted[1]

This standard covers the threaded parts of fire-hose couplings, hydrant outlets, stand-pipe connections, and all other fittings on fire lines where fittings 2½, 3, 3½, and 4½ in. nominal diameter are used. The threads conform to the American Standard (National) form having an included angle of 60 deg and are truncated top and bottom (see Tables XL, XLI, and XLII).

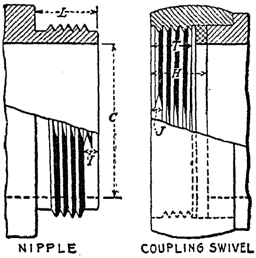


FIG. 14.—American standard fire-hose coupling.

TABLE XL.—CHARACTERISTICS OF THE AMERICAN STANDARD FIRE-HOSE COUPLING SCREW THREAD
(Table 1, ASA B26)
(All dimensions in inches)

1	Nominal inside diameter of hose coupling (C)	2½	3	3½	4½
2	Number of threads per inch	7½	6	6	4
3	Total length of threaded part of coupling and hydrant nipple, external thread (see L, Fig. 14)	1	1⅛	1⅛	1¼
4	Distance from face of nipple to start of second turn (see I, Fig. 14)	¼	5⁄16	5⁄16	7⁄16
5	Depth of coupling swivel to washer seat (see H, Fig. 14)	15⁄16	1 1⁄16	1 1⁄16	1 3⁄16
6	Distance from face of coupling swivel to start of second turn (see J, Fig. 14)	3⁄16	¼	¼	⅜
7	Depth of thread of coupling swivel (see T, Fig. 14)	11⁄16	13⁄16	13⁄16	15⁄16

[1] For complete requirements, reference may be made to ASA B26 (see note, p. 430).

TABLE XLI.—LIMITING DIMENSIONS FOR THREADS OF COUPLING SWIVELS AND HYDRANT CAPS (INTERNAL)
(Table 2, ASA B26)
(All dimensions in inches)

Nominal size	Number of threads per inch	Minimum major diameter	Pitch diameter		Minor diameter	
			Maximum	Minimum	Maximum	Minimum
2.500	7.5	3.0836	3.0130	2.9970	2.9424	2.9104
3.000	6.0	3.6389	3.5486	3.5306	3.4583	3.4223
3.500	6.0	4.2639	4.1736	4.1556	4.0833	4.0473
4.500	4.0	5.7859	5.6485	5.6235	5.5111	5.4611

TABLE XLII.—LIMITING DIMENSIONS FOR THREADS OF COUPLING AND HYDRANT NIPPLES (EXTERNAL)
(Table 3, ASA B26)
(All dimensions in inches)

Nominal size	Number of threads per inch	Major diameter		Pitch diameter		Maximum minor diameter
		Maximum	Minimum	Maximum	Minimum	
2.500	7.5	3.0686	3.0366	2.9820	2.9660	2.8954
3.000	6.0	3.6239	3.5879	3.5156	3.4976	3.4073
3.500	6.0	4.2439	4.2079	4.1356	4.1176	4.0273
4.500	4.0	5.7609	5.7109	5.5985	5.5735	5.4361

THE WELDED JOINT

The application of fusion welding to the fabrication and erection of piping has evolved an entirely new form of construction for high-pressure high-temperature services and has brought about signal improvements in lower pressure work. In many cases, through the use of steel valves with ends machined to weld directly to the pipe, complete piping systems have been constructed without bolted line joints. In other cases flanged-end valves of steel or cast iron have been retained in conjunction with welding-neck steel flanges welded to the adjacent pipe. Pipe-to-pipe joints have been welded in either case.

For pressures below 125 lb, the relatively low cost of flanged or screwed construction having cast-iron valve bodies, fittings, and flanges has restricted the use of wholly welded construction requiring steel fittings and valve bodies, especially where these items

predominate over pipe. The lack of inexpensive valves and fittings that can be welded imposes a handicap in low-pressure work through the necessity for using steel valves and welding

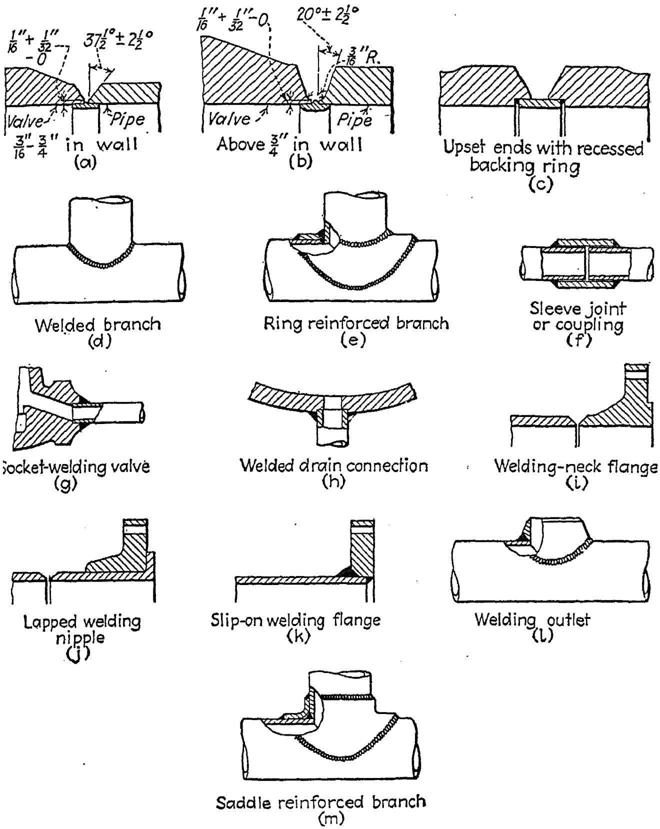


FIG. 15.—Typical welding details.

fittings actually suitable for pressures of 200 to 300 lb or more. As the advantages of welded piping systems are better appreciated, development of inexpensive steel valves and fittings suited for welding into low-pressure lines is to be expected.

Electric-arc Welding.[1]—The development of covered and coated electrodes has made it possible to make strong, ductile, and sound welds by the metallic-arc process. The weld metal is deposited in thin layers or beads shielded by a gaseous atmosphere or slag produced through the burning or melting of the covering on the electrode. If sound welds are to be produced, each layer or bead must be cleaned, and all trapped slag and unfused areas chipped out before applying the next bead. Welding electrodes for arc welding are covered in ASTM Specification A233.

Proving of welding processes and qualifying of welding operators for making butt welds in pipe are covered in the rules of the ASME Boiler Construction Code, the American Welding Society,

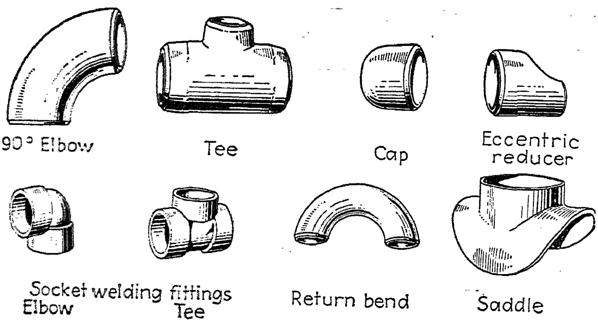


FIG. 16.—Welding fittings.

and the ASA Code for Pressure Piping. Although these rules are sponsored and issued under three auspices and may be slightly at variance from time to time, it is the aim of the several bodies concerned to keep all versions in harmony as far as possible.

Typical welding details are shown in Fig. 15. The ends of pipe and fittings in the past have been beveled in a variety of ways. The use of a standard 37½-deg bevel with 1/16-in. land for pipe wall thicknesses from 3/16 to 3/4 in., inclusive, and of a 20-deg 3/16-in. radius U bevel with 1/16-in. land for thicknesses greater than 3/4 in. has been adopted as recommended practice for electric-

[1] See also: (a) "Welding Handbook," American Welding Society, 33 West 39th St., New York 18, N.Y.

(b) "The Practical Design of Welded Steel Structures," by H. Malcolm Priest, *Welding J.*, September, 1943, pp. 677–711. Contains extensive bibliography.

arc welding by ASA Sectional Committee B16. (For details of backing rings and bevels, see Figs. 15*a* and 15*b*.) A 30-deg bevel for line pipe is generally used in the oil and gas industries. Details of socket welding-end valves, welding couplings, and method of attaching small pipe to large pipe also are shown in Fig. 15. Similar details are illustrated in the Code for Pressure Piping, ASA B31.1. A group of ready-made butt-welding and socket-welding fittings is shown in Fig. 16. For dimensions, see abstracts of ASA B16.9 and B16.11, pages 502 to 507, inclusive.

Preheating.—Probably most arc welds of carbon steel would be improved by preheating the material before laying the first few beads, at least, but the advantages of so doing are not sufficiently manifest to warrant such procedure. In the case of welding carbon-molybdenum and some other alloy steels, however, the tendency for excessive air hardening without preheating is sufficient to make that precaution worth while in many cases.

Molybdenum gives steel an air-hardening tendency when present in amounts of the order of 0.5 to 0.75 per cent, which percentage overlaps the permissible range of molybdenum in carbon-molybdenum pipe. When preheating is not employed in welding carbon-molybdenum pipe, there may be a tendency to form fine cracks in the weld metal before it can be stress-relieved. This tendency is particularly pronounced in the layers first deposited, since the chilling effect is more severe at the outset of welding before the arc has had a chance to heat the joint. Preheating usually is carried on at 300 to 600 F, which temperature range is maintained throughout the welding operation. The Code for Pressure Piping requires that carbon-molybdenum steel shall be heated to not less than 400 F before and during welding. Carbon steel having a carbon content in excess of 0.35 per cent and a wall thickness of ½ in. or greater also is required by that Code to be preheated to not less than 400 F.

The same equipment used for stress-relieving field welds usually can be applied to preheating, although simpler heaters can be used owing to the lower temperature required for reheating.

Examinations for hardness value of cross sections of preheated and unpreheated welds of carbon-molybdenum pipe show greater variations and higher hardness values throughout the unpreheated specimens. Such variation in hardness value is considered undesirable since it is thought to indicate a condition tending toward easier fracture by shock than where uniform hardness obtains.

Gas Welding.—Earlier pipe-joint welding was done to a large extent with oxyacetylene torches. Recent development of multilayer gas welding and improved technique enable gas welds to be produced of a quality said to be comparable with good arc welds. Backing rings for gas welding are available which have a recessed groove in place of the projecting ridge shown in Fig. 15*b*.

For pipe wall thickness up to about ¾ in., a 30- to 37½-deg bevel, as in Fig. 15*a*, is suitable for gas welding. Some form of straight bevel for thicknesses greater than ¾ in. is preferred for gas welding to the U-shaped bevel shown in Fig. 15*b*. Welding rods for gas welding are covered in ASTM specification A251.

The development of oxyacetylene cutting torches capable of neatly cutting steel plate up to 6 in. or more thick has created an entirely new industry based on the use of "flame machining." In pipe fabrication, gas torches cut and bevel pipe for branch connections and are in other respects indispensable tools.

Welding Lines under Pressure.—Where it is inconvenient to shut down for making repairs or attaching saddles for service connections, welding frequently is done with pressure on the line. This practice is well established in the gas industry (see pages 1204 to 1207) where it would be necessary otherwise to purge the line before welding to prevent getting an explosive mixture of air and gas inside the pipe. Welding under pressure is particularly applicable to air, gas, steam, or water-distribution systems where continuity of service is essential. Saddles for service connections can be welded to the main, which is subsequently drilled under pressure with a Mueller tapping machine operating through a valve and nipple (see page 1080). Pipe flaws, pits, or defective seams can be repaired by welding patches over them, and defective welds in line joints can be repaired by covering them with split welding sleeves (see page 1206).

Strength of Welds.—Both electric-arc and gas welds of low-carbon steel can be made stronger than the parent metal if the welders possess the requisite skill and follow approved technique. Welds in higher carbon and alloy steels may be made equal in strength to the parent metal under favorable conditions where higher strength welding rods may be used, although for overhead-position welding lower strength electodes have been found more workable. Consequently, welds in medium-carbon (0.25 to 0.35 carbon) steel usually do not develop the full strength of the parent

metal, although exceeding the minimum physical properties specified for medium-carbon-steel pipe and plate.[1]

Stress-relieving Welds.—In making welded joints in thick-walled material, contraction stresses are set up during solidification of the fused metal which probably approach the yield point of the material. That such stresses are of serious proportions is demonstrated by the tendency for welding parts to pull out of alignment if not rigidly supported and welded joints to draw together enough to cold-spring the pipe line and by similar manifestations frequently encountered. The intensity of locked-in stresses seems to depend upon the thickness of the parts joined, the ductility of the weld metal, the welding procedure, and the rigidity of support during welding. Locked-in stresses of this sort in arc welds can be reduced to satisfactory limits by low-temperature annealing called "stress-relieving." It is customary to stress-relieve arc welds made in material ½ in. or more thick, especially if intended for severe service conditions.

The Code for Pressure Piping requires that welded joints in carbon steel shall be stress-relieved when the normal pipe wall thickness is ¾ in. or greater. Welded joints in carbon steel having a carbon content in excess of 0.35 per cent, and in carbon-molybdenum steel shall be stress-relieved when the thickness is ½ in. or greater.

Stress-relief of arc-welded carbon steel usually is done at a temperature of 1100 to 1200 F for a period of 1 hr per inch of thickness, followed by slow cooling to 600 F or below. For carbon-molybdenum and some other alloy steels, the temperature should be held at 1150 to 1300 F for a period of 2 hr per inch of thickness, followed by slow cooling. Various means are employed for stress-relieving field welds, including electric-induction or electric-resistance coils placed around the pipe, gas, or oil-fired refractory

[1] (*a*) "Fusion Welding—The Modern Method of Fabrication," by Eric R. Seabloom, *The Valve World* (Crane Company), April, 1937.

(*b*) "An Examination of Welds Made under Field Conditions for High-pressure, High-temperature Steam Station Piping," by A. E. White, D. H. Corey, and C. L. Clark, *J. Am. Welding Soc.*, September, 1934.

(*c*) "High Pressure and High Temperature Piping," by F. C. Fantz, *Welding J.*, August, 1941.

(*d*) "Pipe Welding Burlington Generating Station," by Philip Salmon, *Welding J.*, January, 1941.

(*e*) "Investigation of Gas and Arc Fillet Welds in Piping," by I. H. Carlson and E. R. Seabloom, *Welding J.*, November, 1940, pp. 846–854.

rings, and even simple torch heating, although this is not sufficiently uniform to be desirable. Shop welds may be stress-relieved by one of the heating devices just mentioned, or the assembled parts may be placed in an annealing furnace if their nature permits. The general effect of stress-relief is to reduce the locked-in stresses through plastic flow to intensities approaching the yield point of the material at the stress-relieving temperature.

The need for stress-relieving gas welds and the advantages gained thereby are not so well established as in the case of arc welds. Where gas welds are stress-relieved, the operation frequently is more in the nature of a normalizing heat treatment which is carried on at a somewhat higher temperature.

The chief purpose in stress-relieving an arc weld intended for use at temperatures above 850 F is to reduce locked-in stresses to safe proportions *before* placing the line in service, thus minimizing the possibility of starting cracks or opening up incipient flaws in or adjacent to the weld. If it were not for the need of getting the weld through this rather difficult initial period of warming up the line and placing it in use, there would not be so much occasion for stress-relieving joints in lines working at temperatures above 850 F, since the operating temperature, if given enough time, would materially reduce the intensity of such stresses through plastic flow. In the case of boiler-feed lines and other high-pressure piping operating at temperatures below 700 F, where there is little possibility of this self-annealing effect, the desirability of stress-relieving is even more apparent.

Methods of Quality Control.—In addition to setting up a welding procedure and qualifying welders in accordance with code requirements, some means of quality control is needed to ensure obtaining consistently good field welds. One method of quality control employs a portable instrument known as the *Arcronograph*[1] which draws a continuous graph of the conditions under which each bead is deposited. The Arcronograph is an electrical recording instrument which employs a three-element vacuum tube to take account of the ratio of arcing time to short-circuiting time in the process of depositing each globule of weld metal. Since this ratio should be essentially constant for a given size and type of electrode, the graphical record serves to show how consistently the operator has performed in laying each bead. Welders ordi-

[1] See "Evolution of the Arcronograph," by Bela Ronay, *J. Am. Soc. Naval Engrs.*, August, 1934.

narily check the record as they finish each bead so that any flaws become apparent as the work proceeds and can be chipped out or otherwise corrected while the bead is still exposed without having to wait for an X-ray or other examination of the completed weld. The fact that the welder works under the constant supervision of a graphical record and cannot conceal his mistakes is held to be one of the principal advantages of his instrument. The Arcronograph is readily moved from place to place and is suitable for quality control for either shop or field welding.

Another means of quality control extensively used under shop conditions in the manufacture of welded boiler drums and pressure vessels is the taking of photographs by either the *X-ray or gamma-ray methods.*[1] Such photographic examination has been applied under shop conditions to valve bodies and welded joints, and, to a limited degree, the X-ray method has been extended to field-welded pipe joints. The necessity for taking a considerable number of shots from different angles around each joint examined in order to locate defects with sufficient certainty makes this an expensive method to use. The fact that joints are photographed after completion is said to increase the care taken by the welding operators, thus tending toward more uniform and reliable results.

Among other nondestructive tests sometimes used for quality control in welding are *magna-fluxing* each ⅜ in. or so in thickness of deposited metal.

In the magna-flux method iron oxide powder is sprinkled on the material to be examined while the joint is magnetically energized by an electric current. The presence of any fine cracks or other flaws near the surface is pointed out through the pattern taken by the iron oxide powder.

Trepanning.—The trepanning of specimens from welds is a useful method of examination which requires patching the hole afterward. Trepanning of cylindrical plugs ranging from ⅝ to 4½ in. in diameter can be accomplished readily with an inexpensive tool called a "hole saw"[2] which resembles a circular hacksaw and uses a centering drill in starting the cut. Such plugs have to be

[1] "The Gamma-Ray Radiography of Welded High Pressure Power Plant Piping," by R. W. Emerson, Symposium of Radiography, *Proc. ASTM*, Vol. 42, 1942, p. 1111. Some twelve other papers dealing with X-ray and gamma-ray methods were presented in this symposium (see pp. 1023–1176 of same volume).

[2] Hole saws are fairly common and inexpensive tools. A satisfactory article can be obtained from the Armstrong-Blum Mfg. Co., 5700 W. Bloomingdale Ave., Chicago, Ill.

cut through the entire pipe wall in order to remove them from the hole, thus leaving a difficult job to patch. If plugs can be trepanned from the *underside* of a pipe when using a hole saw this will facilitate removing chips so as not to get them inside the pipe. A more expensive device called a "weld prober,"[1] which cuts in from two directions with a saw shaped like a segment of a sphere to remove a boat-shaped specimen, has the advantage of being able to free the specimen without cutting entirely through the pipe wall. Such specimens can be cut either with or across the weld depending on what one wishes to see. Somewhat equivalent specimens can be taken from a pipe wall in a crosswise direction only by making two intersecting cuts with an ordinary hack saw. Trepanning can be employed on an occasional weld as a spot check on penetration, fusion, and porosity. It also has a place in examining selected welds from time to time for possible deterioration in service.

Procedure Specifications.—In order to comply with requirements of the Code for Pressure Piping and of the Boiler Code it is necessary to qualify both the welding procedure and the operator in accordance with definite specifications. Recommended forms of Procedure Specifications are given in Appendix II of the American Standard Code for Pressure Piping, ASA B31.1[2] from which the following requirements are abstracted:

Procedure Qualification.—When a procedure has been qualified under a particular Procedure Specification, requalification is required only to care for significant changes in the material, welding rods, or conditions of welding. For welded-butt joints, the type and number of tests on pipe 6-in. nominal size or larger are as shown in the accompanying table for each procedure and position to be used in construction. If a test is made in the maximum thickness, no test need be made on the ⅜ in. thickness.

PROCEDURE QUALIFICATION TESTS FOR WELDED-BUTT JOINTS IN PIPE
(Table 48 of ASA A31.1-1942)

Maximum thickness to be welded in construction	Test pipe wall thickness	Number and type of tests required				
		Reduced section tensile	Free bend	Root bend	Face bend	Side bend
Up to and including ¾ in.	⅜ in.	2	2	2	2	
Over ¾ in.	Maximum but need not be over 1 in.	2	2	..	..	4

[1] Manufactured by the Fibre-Metal Products Co., Chester, Pa.

[2] Copies may be obtained from the American Society of Mechanical Engineers, 29 West 39th St., New York 18, N.Y.

If fillet joints only are to be made, it is necessary, in addition to making the fillet-weld soundness test, that the procedure shall be qualified for butt joints; in such case, only the reduced-section-tensile and the free-bend tests for butt joints need be made.

Butt joints in pipe are classified as (1) horizontal rolled, (2) horizontal fixed, and (3) vertical fixed. Definite procedures for removing and testing specimens from test welds made in each position are given in the Code for Pressure Piping.

Reduced-section-tensile Test.—The tensile strength is required to be not less than 100 per cent of the minimum tensile strength of the base material.

Free-bend Test.—The elongation determined by the free-bend test is required to be not less than 30 per cent for stress-relieved welds and 25 per cent for non-stress-relieved welds. Detailed descriptions of the initial and final bending of free-bend specimens are given in the Code for Pressure Piping. The load is removed and the elongation measured when either a crack or other open defect exceeding 1/16 in. in any direction appears on the face of the weld. Cracks occurring on the corners of the specimen during testing are disregarded.

Root-, Face-, Side-bend and Fillet-weld Soundness Test.—Tests for soundness are all made by means of a guided-bend test jig which has been developed for this purpose. A crack in the weld metal or between the weld and the base metal which during, or at, the completion of bending opens up by more than 1/8 in. in any direction is cause for rejection. Cracks occurring in the corners of the specimen during testing are disregarded. For specific instructions as to use of the guided bend jig in testing the different types of specimens, reference should be made to the Code for Pressure Piping.

Operator Qualification.—The qualification tests for operators are intended to determine ability to made sound welds and results are judged solely by the bend tests described above. Requalification of an operator is required if changes are made in the Procedure Specification with respect to (1) class of parent metal, (2) position of welding, (3) change in method of welding.

Number of Test Welds.—The number and type of tests for each position for which an operator is to be qualified are shown in the accompanying table.

OPERATOR QUALIFICATION TESTS FOR WELDED-BUTT JOINTS IN PIPE
(Table 49 of ASA A31.1-1942)

Thickness for which operator is to be qualified	Thickness of material for test weld	Number and type of tests required		
		Root bend	Face bend	Side bend
Up to and including 3/4 in. . .	3/8 in.	2	2	
Over 3/4 in.	Maximum but not over 1 in.	. .	. .	4

If a test weld is made in the maximum thickness, no test weld need be made in the 3/8 in. thickness. If an operator is tested in the position in which the axis of the pipe is vertical, he need not be tested for a weld on a pipe with the axis horizontal and rolled during welding. If tested on a weld in which the axis of the pipe is horizontal and not rolled while welding, he need not be tested on the horizontal rolled weld.

Test Specimens.—Detailed descriptions of the locations from which the face-, root-, and side-bend specimens shall be taken are given in the Code for Pressure Piping.

Grades of Pipe Welding.—Some welding organizations find it desirable to set up procedure specifications for their high-pressure, high-temperature services which differ in certain respects from their requirements for low-pressure low-temperature services. The need for preheating and for post heating or stress relieving is dependent to some extent upon the wall thickness of the pipe, hence, unless alloy piping is used, the welding procedure specification for low-pressure services in general does not include either preheating or post-heating requirements. Aside from the need for qualifying both the procedure and the operators for greater pipe thickness and for alloy material, if used, the principal difference between the two grades of welding as specified by one organization lies in the requirements for cleaning each bead and layer of weld metals as abstracted below:

Cleaning High-pressure Welds.—All slag or flux shall be removed from each crater by means of a light cleaning hammer before proceeding with the next electrode. Each completed bead or layer shall be thoroughly cleaned with a power-driven wire brush. It shall then be further prepared for deposition of the succeeding bead or layer by chipping out any defects such as cracks or blow holes, that may appear, removing globules of spattered weld metal from the surface of the weld and the surface of the pipe ends, and generally correcting any condition which might prevent or interfere with the deposition of sound weld metal, adequately fused to the existing bead or layer and to the base metal. This cleaning shall be done by a chipper, assigned to the welder, using power-driven tools.

Cleaning Low-pressure Welds.—All slag or flux shall be removed from each crater by means of a light cleaning hammer before proceeding with the next electrode. Each completed bead or layer shall be thoroughly cleaned with a wire-bristled handbrush. Surface defects shall be chipped out, by the welder, using a hammer and hand chisels.

ASTM Standard Specifications for FACTORY-MADE WROUGHT CARBON-STEEL AND CARBON-MOLYBDENUM-STEEL WELDING FITTINGS

Serial Designation A234-44

Abstracted[1]

These specifications cover factory-made wrought carbon-steel and carbon-molybdenum-steel welding fittings for pressure piping. For dimensional standards for butt-welding fittings, see abstract of American Standard for Steel Butt-Welding Fittings, ASA B16.9, on page 502. For socket welding fittings,

[1] For complete specification, reference may be made to ASTM A234 (see note, p. 369).

see abstract of Proposed American Standard for Steel Socket Welding Fittings, ASA B16.11, on page 505. These specifications do not cover cast welding fittings which are covered by ASTM A216 and A217 for carbon and alloy steel castings, respectively (see abstracts, pages 685 and 686).

Process and Material.—The steel shall be made by either or both the open-hearth or electric-furnace process. When fittings are to be used for service above 750 F and when specified on the order, the steel shall be silicon-killed. The steel for welding fittings may consist of blooms, billets, slabs, and of forging quality bars, plates, or seamless tubes.

Manufacture.—Forging or shaping operations may be performed by hammering, pressing, piercing, rolling, extruding, upsetting, or fusion welding, or by a combination of these operations. Fusion welding of seams or for joining parts of fittings shall be performed in accordance with the requirements for fabrication details of the Code for Pressure Piping, ASA B31.1, Section 6, Chap. 4 (see abstract, page 496).

Chemical Composition.—The chemical composition shall conform to the requirements for the respective material prescribed in the table on pages 500–501.

Heat Treatment.—Heat-finished carbon-steel fittings upon which the final forming operation is completed at a temperature above 1150 F need not be heat-treated provided they are cooled in still air. Any fitting finished at a temperature in excess of 1650 F shall subsequently be normalized. (For definition of normalizing, annealing, and drawing see page 324.)

Cold-finished carbon-steel fittings completed at a temperature below 1150 F shall be stress-relieved after or during final forming and prior to machining by heating to 1100 to 1200 F for 1 hr per in. of thickness and cooling in the furnace or in still air. Carbon-molybdenum-steel fittings shall be heat-treated as prescribed in the respective specifications listed in the accompanying table. Fusion-welded fittings shall be heat-treated in accordance with the requirements for fabrication details of the Code for Pressure Piping ASA B31.1, Section 6, Chap. 4 (see abstract, page 493). Forge-welded fittings shall be normalized or annealed.

Tensile Properties.—Tension tests of the finished fittings are not required, unless so agreed between the manufacturer and the purchaser.

Hydrostatic Tests.—Welding fittings shall be capable of withstanding a hydrostatic pressure test determined by Barlow's formula where D = outside diameter at bevel (see page 41 for formula) which will produce a stress equal to one-half the minimum specified yield point of the designated material. Hydrostatic acceptance tests shall be applied only when specified in the order.

Marking.—The manufacturer's name or trade-mark, the schedule number, the identification symbol for the respective material and size shall be stamped, stenciled, or otherwise suitably marked on each fitting. See abstract of MSS Standard Marking System, SP-25, page 687, for order of omitting markings on small sizes. The schedule number and material marking may be omitted on Schedule 40, Grade A, low-carbon steel fittings. On all alloy-steel fittings the manufacturer's name or trade-mark and the designation of the material shall be marked with a steel stamp or identified by any other suitable method agreed upon by the manufacturer and the purchaser so that the marking will be of a permanent nature. Other identification marks may be stenciled or otherwise suitably labeled.

Certification.—Unless otherwise specified in the order, a certification that the finished fittings conform to the requirements of these specifications shall be the basis of acceptance.

Chemical Requirements for Factory-made Welding Fittings
(Table 1 of ASTM A 234)

Type of material	ASTM designation[a]	Identification symbol	Carbon, per cent: Thickness, in.	Carbon, max, per cent	Manganese, per cent	Phosphorus, max, per cent: Acid	Phosphorus, max, per cent: Basic	Sulphur, max, per cent	Silicon, per cent	Molybdenum, per cent
Seamless steel pipe:										
Carbon steels	A 106, Grade A	WPA		0.25 max	0.39 to 0.90	0.04		0.06	[b]	
	A 106, Grade B, silicon killed	WPB		0.35 max	0.35 to 1.00	0.04		0.06	0.10 min	
Carbon-molybdenum steels	A 206	WP1		0.10 to 0.20	0.30 to 0.60	0.04		0.05	0.10 to 0.50	0.45 to 0.65
Rolled steel plate or strips:										
Carbon steels	A 70	WR	Up to ¾ Over ¾	0.25 0.30	0.80 max	0.05[c] 0.04[d]	0.04[c] 0.035[d]	0.05[c] 0.04[d]		
	A 89, Grade B	WRA	Up to ¾ Over ¾	0.20 0.22	0.80 max	0.06[c] 0.04[d]	0.04[c] 0.035[d]	0.05[c] 0.04[d]		
	A 212, Grade A	WRB	Up to 1 1 to 2 Over 2 to 4½	0.28 0.31 0.33	0.90 max	0.04[c] 0.035[d]		0.05[c] 0.04[d]	0.15 to 0.30	
Carbon-molybdenum steels	A 204, Grade A	WR1A	Up to 1 1 to 2 Over 2 to 4 Over 4 to 6	0.18 0.21 0.23 0.25	0.90 max	0.04[c] 0.035[d]		0.05[c] 0.04[d]	0.15 to 0.30	0.40 to 0.60
	A 204, Grade B	WR1B	Up to 1 1 to 2 Over 2 to 4 Over 4 to 6	0.20 0.23 0.25 0.27	0.90 max	0.04[c] 0.035[d]		0.05[c] 0.05[d]	0.15 to 0.30	0.40 to 0.60

Chemical Requirements for Factory-made Welding Fittings.—(*Concluded*)

Type of material	ASTM designation	Identification symbol	Carbon, per cent	Manganese, per cent	Phosphorus, max, per cent	Sulphur, max, per cent	Silicon, per cent	Molybdenum, per cent
Forging blooms and billets:								
Carbon steels............	A 105, Grades I and II	WF	[e]	[e]	0.05	0.05	[b]	
Slab or bars:								
Carbon-molybdenum steels	A 182, Grade F1	WF1	0.35 max	0.30 to 0.80	0.04	0.05	0.20 to 0.50	0.40 to 0.60

[a] These designations are given only for cross reference to the respective chemical compositions and refer to the following ASTM specifications:
Carbon-Steel Plates for Stationary Boilers and Other Pressure Vessels (A 70)
Low Tensile Strength Carbon-Steel Plates of Flange and Firebox Qualities (A 89)
High Tensile Strength Carbon-Silicon Steel Plates for Boilers and Other Pressure Vessels (Plates 4½ in. and under in Thickness) (A 212)
Molybdenum-Steel Plates for Boilers and Other Pressure Vessels (A 204)
Forged or Rolled Steel Pipe Flanges, Forged Fittings, and Valves and Parts for High-Temperature Service (A 105)
Forged or Rolled Alloy-Steel Pipe Flanges, Forged Fittings, and Valves and Parts for Service at Temperatures from 750 to 1100 F (A 182)
Seamless Carbon-Molybdenum Alloy-Steel Pipe for Service at Temperatures from 750 to 1000 F (A 206)
Lap-Welded and Seamless Steel Pipe for High-Temperature Service (A 106)

[b] This steel shall be silicon-killed when so specified in the order.

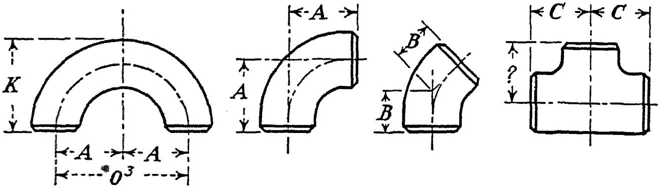
[c] Flange grade.

[d] Firebox grade.

[e] The carbon and manganese contents of this steel shall be a matter of agreement between the manufacturer and the purchaser, but the contents for this welding steel shall not exceed 0.35 ond 0.90 per cent, respectively.

TABLE XLIII.—AMERICAN STANDARD STEEL BUTT-WELDING FITTINGS

DIMENSIONS OF ELBOWS, TEES,[1] AND RETURN BENDS
(Tables 1 and 4, ASA B16.9)



(All dimensions in inches)

Size	Outside diameter at bevel	Center to end			Back to face
		90-deg welding elbows	45-deg welding elbows	Of run welding tee	180-deg return bend
		A	*B*	*C*	*K*
1	1.315	1½	⅞	1½	2³⁄₁₆
1¼	1,660	1⅞	1	1⅞	2¾
1½	1.900	2¼	1⅛	2¼	3¼
2	2.375	3	1⅜	2½	4³⁄₁₆
2½	2.875	3¾	1¾	3	5³⁄₁₆
3	3.500	4½	2	3⅜	6¼
3½	4.000	5¼	2¼	3¾	7¼
4	4.500	6	2½	4⅛	8¼
5	5.563	7½	3⅛	4⅞	10⁵⁄₁₆
6	6.625	9	3¾	5⅝	12⁵⁄₁₆
8	8.625	12	5	7	16⁵⁄₁₆
10	10.750	15	6¼	8½	20⅜
12	12.750	18	7½	10	24⅜

[1] The dimensions of welding tees cover those which have side outlets from one size less than half the size of the runway opening of the tees to full size.
[2] *Author's Note:* Several manufacturers offer tees having this dimension same as *C*.
[3] Center-to-center dimension, $O = 2A$.

American Standard for STEEL BUTT-WELDING FITTINGS

ASA B16.9—1940

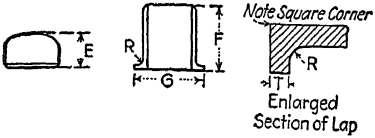
Abstracted[1]

A dimensional standard has been formulated under the procedure of the American Standards Association for welding fittings suitable for use with the

[1] For complete requirements, reference may be made to ASA B16.9 (see note, p. 430).

TABLE XLIV.—AMERICAN STANDARD STEEL BUTT-WELDING FITTINGS

DIMENSIONS OF CAPS AND LAPPED-JOINT STUB ENDS (Table 2, ASA B16.9)



(All dimensions in inches)

Size	Outside diameter at bevel	Welding caps[1,2]	Lapped-joint stub ends: Length[5]	Radius of fillet[3]	Diameter of lap[4]
		E	F	R	G
1	1.315	1½	4	1/8	2
1¼	1.660	1½	4	3/16	2½
1½	1.900	1½	4	1/4	2⅞
2	2.375	1½	6	5/16	3⅝
2½	2.875	1½	6	5/16	4⅛
3	3.500	2	6	3/8	5
3½	4.000	2½	6	3/8	5½
4	4.500	2½	6	7/16	6 3/16
5	5.563	3	8	7/16	7 5/16
6	6.625	3½	8	1/2	8½
8	8.625	4	8	1/2	10⅝
10	10.750	5	10	1/2	12¾
12	12.750	6	10	1/2	15
14	14.000	6½	12	1/2	16¼
16	16.000	7	12	1/2	18½
18	18.000	8	12	1/2	21
20	20.000	9	12	1/2	23
24	24.000	10½	12	1/2	27¼

THICKNESS (T). The basic minimum lap thickness (T) shall not be less than nominal pipe wall thickness (see page 361). The tolerance on lap thickness is −0 in. + 1/16 in.

[1] The shape of these caps shall be ellipsoidal and shall conform to the shape requirements as given in the ASME Boiler Construction Code.

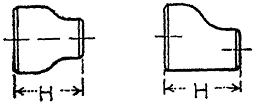
[2] Dimension E for sizes 12 in. and smaller is applicable only up to and including Schedule 80. For sizes 14 in. and larger, it is applicable for a thickness up to ½ in. only. A thicker wall will require a greater dimension.

[3] These dimensions conform to the radius established for lap joint flanges in American Standard for Steel Pipe Flanges and Flanged Fittings (ASA B16e).

[4] This dimension is for standard machined facings in accordance with American Standard for Steel Pipe Flanges and Flanged Fittings (ASA B16e). The back face of the lap shall be machined to conform to the surface of the flange on which it seats. Where ring joint facings are to be applied use dimension K as given in ASA B16e.

[5] Dimension F for sizes 12 in. and smaller is applicable only up to and including Schedule 80. For sizes 14 in. and larger, it is applicable for a pipe wall thickness up to ½ in. only.

TABLE XLV.—AMERICAN STANDARD STEEL BUTT-WELDING FITTINGS—DIMENSIONS OF REDUCERS
(Table 3, ASA B16.9)



(All dimensions in inches)

Size	Outside diameter at bevel, Large end O.D.	Outside diameter at bevel, Small end O.D.	End to end H
1 x ⅜	1.315	0.675	2
1 x ½	1.315	0.840	2
1 x ¾	1.315	1.050	2
1¼x ½	1.660	0.840	2
1¼x ¾	1.660	1.050	2
1¼x1	1.660	1.315	2
1½x ½	1.900	0.840	2½
1½x ¾	1.900	1.050	2½
1½x1	1.900	1.315	2½
1½x1¼	1.900	1.660	2½
2 x ¾	2.375	1.050	3
2 x1	2.375	1.315	3
2 x1¼	2.375	1.660	3
2 x1½	2.375	1.900	3
2½x1	2.875	1.315	3½
2½x1¼	2.875	1.660	3½
2½x1½	2.875	1.900	3½
2½x2	2.875	2.375	3½
3 x1¼	3.500	1.660	3½
3 x1½	3.500	1.900	3½
3 x2	3.500	2.375	3½
3 x2½	3.500	2.875	3½
3½x1¼	4.000	1.660	4
3½x1½	4.000	1.900	4
3½x2	4.000	2.375	4
3½x2½	4.000	2.875	4
3½x3	4.000	3.500	4

Size	Outside diameter at bevel, Large end O.D.	Outside diameter at bevel, Small end O.D.	End to end H
4x 1½	4.500	1.900	4
4x 2	4.500	2.375	4
4x 2½	4.500	3.500	4
4x 3	4.500	3.500	4
4x 3½	4.500	4.000	4
5x 2	5.563	2.375	5
5x 2½	5.563	2.875	5
5x 3	5.563	3.500	5
5x 3½	5.563	4.000	5
5x 4	5.563	4.500	5
6x 2½	6.625	2.875	5½
6x 3	6.625	3.500	5½
6x 3½	6.625	4.000	5½
6x 4	6.625	4.500	5½
6x 5	6.625	5.563	5½
8x 3½	8.625	4.000	6
8x 4	8.625	4.500	6
8x 5	8.625	5.563	6
8x 6	8.625	6.625	6
10x 4	10.750	4.500	7
10x 5	10.750	5.563	7
10x 6	10.750	6.625	7
10x 8	10.750	8.625	7
12x 5	12.750	5.563	8
12x 6	12.750	6.625	8
12x 8	12.750	8.625	8
12x10	12.750	10.750	8

more common schedules of pipe wall thickness. Th marking and tolerances on wall thickness conform to ASA standards for mating pipe (see page 361). The type of bevel conforms to recommended practice of ASA B16e (see Fig. 15). The center-to-end and end-to-end dimensions given in Tables XLIII to XLV, inclusive, have been agreed to by the various manufacturers. A tolerance of

$\frac{1}{16}$ in. for sizes 8 in. and smaller, and of $\pm \frac{3}{32}$ in. for sizes 10 in. and larger has been established for center-to-end dimensions.

Wrought fittings covered by this standard shall be in accordance with ASTM Specification A234 (see page 498); cast fittings shall be in accordance with ASTM Specifications A216 for carbon steel-welding grade castings (see page 685) and A217 for alloy steel-welding grade castings (see page 686). Hydrostatic testing of wrought fittings is not required. Bursting tests on pilot fittings are prescribed to demonstrate the adequacy of fitting design.

Proposed American Standard for STEEL SOCKET-WELDING FITTINGS

ASA B16.11

This proposed standard covers dimensions, finish, tolerances, testing, marking, and minimum requirements for socket- and fillet-weld dimensions for wrought and cast carbon and alloy-steel socket-welding fittings. Fittings are given the same pressure-temperature ratings as pipe of the same schedule numbers if made of a material having allowable stresses which are equal to or greater than the pipe material. Fittings shall be marked in accordance with MSS Standard Practice, SP25 (see abstract, page 687), to indicate the grade of ASTM wrought or cast material to which they conform. Wrought fittings shall be in accordance with ASTM A234 (see page 498), cast carbon steel with A216 (see page 685), and cast alloy steel to A217 (see page 686).

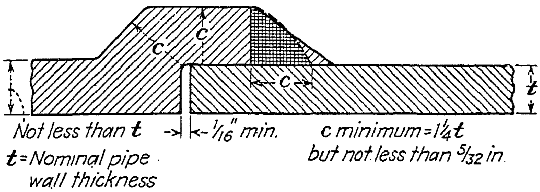
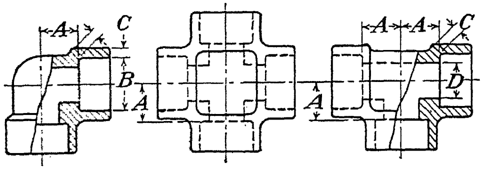


FIG. 17.—Minimum requirements for socket and fillet-weld dimensions.

The actual bursting strength of fittings shall be not less than the computed bursting strength of pipe of the designated schedule number and material. Provision is made for hydrostatic bursting tests to determine the relative strength of the fittings with respect to the computed strength of straight pipe. Wrought fittings shall be capable of withstanding hydrostatic tests which are computed by Barlow's formula (see page 41) to develop a stress in straight pipe 50 per cent of the yield strength, but routine hydrostatic testing of each fitting is not required. Cast fittings shall be given hydrostatic tests as specified in ASTM A216 or A217.

The center-to-bottom-of-socket dimensions are fixed for each size and type of fitting with a tolerance of $\pm \frac{1}{32}$ in. The depth of socket is given only as a minimum dimension of $\frac{3}{8}$ in. for sizes $\frac{3}{8}$ to $\frac{1}{2}$ in. inclusive, $\frac{1}{2}$ in. for sizes $\frac{3}{4}$ to $1\frac{1}{2}$ in. inclusive, and $\frac{5}{8}$ in. for sizes 2 in. and larger. The bore of the fitting corresponds to the inside diameter of the designated schedule of pipe with a tolerance of $\pm$ 0.007 in. The bore of the socket is equal to the maximum outside

TABLE XLVI.—PROPOSED DIMENSIONS OF SOCKET-WELDING ELBOWS, TEES, AND CROSSES[1]
(Table 1, ASA B16.11)



(All dimensions in inches)

Nominal pipe size	Depth of socket, min	Center to bottom of socket[3] Sched. 40 and 80	Center to bottom of socket[3] Sched. 160	Bore diameter of socket, min	Socket wall thickness,[2] min Sched. 40	Socket wall thickness,[2] min Sched. 80	Socket wall thickness,[2] min Sched. 160	Bore diameter of fitting Sched. 40	Bore diameter of fitting Sched. 80	Bore diameter of fitting Sched. 160
		A		B	C			D		
1/4	3/8	17/32	...	0.555	0.156	0.156	.....	0.364	0.302	
3/8	3/8	17/32	...	0.690	0.156	0.158	.....	0.493	0.423	
1/2	3/8	5/8	3/4	0.855	0.156	0.184	0.234	0.622	0.546	0.466
3/4	1/2	3/4	7/8	1.065	0.156	0.193	0.273	0.824	0.742	0.614
1	1/2	7/8	1 1/16	1.330	0.166	0.224	0.313	1.049	0.957	0.815
1 1/4	1/2	1 1/16	1 1/4	1.675	0.175	0.239	0.313	1.380	1.278	1.160
1 1/2	1/2	1 1/4	1 1/2	1.915	0.181	0.250	0.351	1.610	1.500	1.338
2	5/8	1 1/2	1 5/8	2.406	0.193	0.273	0.429	2.067	1.939	1.689
2 1/2	5/8	1 5/8	2 1/4	2.906	0.254	0.345	0.469	2.469	2.323	2.125
3	5/8	2 1/4	2 1/2	3.535	0.270	0.375	0.546	3.068	2.900	2.626

thickness, but not less than 5/32 in.
socket dimension as the largest size of the
reducing fitting.

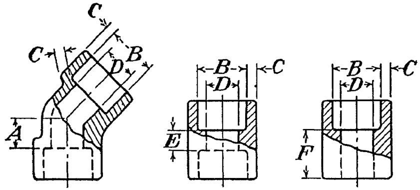
diameter of pipe in accordance with pipe manufacturing tolerances with a variation from that maximum diameter of -0, $+0.010$ in.

The socket- and fillet-weld dimensions as given in the American Standard Code for Pressure Piping, ASA B31.1, are intended to provide a throat dimension in the weld not less than the nominal pipe thickness of the designated schedule. The minimum requirements for socket- and fillet-weld dimensions are shown in Fig. 17.

THE FLANGED JOINT

Flanged joints are required in general where pipe lines or equipment must be disassembled for maintenance work. Connections to cast-iron valves and fittings and to steel flanged-end valves,

TABLE XLVII.—PROPOSED DIMENSIONS OF SOCKET-WELDING 45-DEG ELBOWS, COUPLINGS, AND HALF COUPLINGS[1]
(Table 2, ASA B16.11)



(All dimensions in inches)

Nominal pipe size	Depth of socket, min	Center to bottom of socket for 45-deg ells[3]		Couplings distance between bottoms of sockets[3]	Half couplings, bottom of socket to opposite face	Bore diameter of socket, min	Socket wall thickness,[2] min			Bore diameter of fitting		
		Sched. 40 and 80	Sched. 160				Sched. 40	Sched. 80	Sched. 160	Sched. 40	Sched. 80	Sched. 160
		A		E	F	B	C			D		
1/4	3/8	5/16	.....	1/4	5/8	0.555	0.156	0.156	.....	0.364	0.302	
3/8	3/8	5/16	.....	1/4	11/16	0.690	0.156	0.158	.....	0.493	0.423	
1/2	3/8	7/16	1/2	3/8	7/8	0.855	0.156	0.184	0.234	0.622	0.546	0.466
3/4	1/2	1/2	9/16	3/8	15/16	1.065	0.156	0.193	0.273	0.824	0.742	0.614
1	1/2	9/16	11/16	1/2	1 1/8	1.330	0.166	0.224	0.313	1.049	0.957	0.815
1 1/4	1/2	11/16	13/16	1/2	1 3/16	1.675	0.175	0.239	0.313	1.380	1.278	1.160
1 1/2	1/2	13/16	1	1/2	1 1/4	1.915	0.181	0.250	0.351	1.610	1.500	1.338
2	5/8	1	1 1/8	3/4	1 5/8	2.406	0.193	0.273	0.429	2.067	1.939	1.689
2 1/2	5/8	1 1/8	1 1/4	3/4	1 11/16	2.906	0.254	0.345	0.469	2.469	2.323	2.125
3	5/8	1 1/4	1 3/8	3/4	1 3/4	3.535	0.270	0.375	0.546	3.068	2.900	2.626

[1] Tolerances on these dimensions are given in the text, p. 505.
[2] Minimum dimension C is 1 1/4 times the nominal pipe thickness, but not less than 5/32 in.
[3] Reducing sizes have the same center to bottom of socket dimension as the largest size of the reducing fitting.

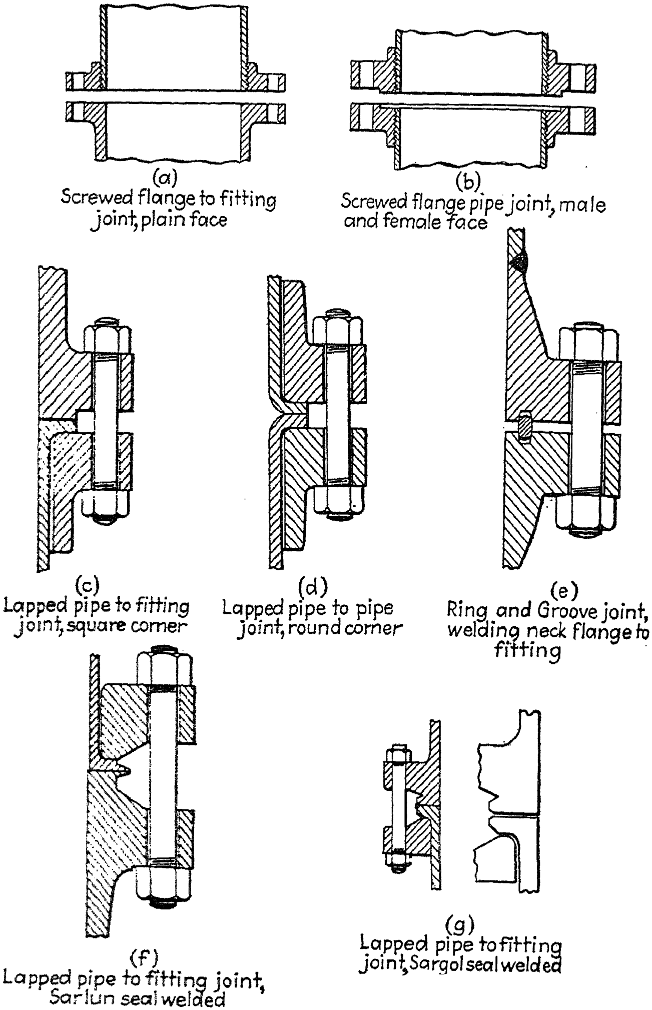


FIG. 18.—Types of flanged joints.

where such construction is considered more economical, must be flanged, and valve-bonnet joints usually are flanged.

Flanges differ in method of attachment to the pipe, whether screwed, welded, or lapped. Contact surface facings may be plain, serrated, grooved for ring joints, seal-welded, or ground and lapped for metal-to-metal contact. The more important methods of attaching flanges to pipes and some common types of joints and facings are shown in Fig. 18 and Table XLVIII.

The important problem of flange design is to select materials and to proportion dimensions of bolts, flanges, and gaskets so that the necessary compression will be maintained on the joint faces over the expected life of the equipment.

Several distinct phases of the problem are involved: (1) type of flange facing, (2) finish of contact surfaces, (3) gasket type and proportions, (4) bolt load required to secure and maintain a tight joint, and (5) proportions of flange needed to support the bolt load.

Type of Flange Facing.—There are numerous types of contact facings for flanges, the most simple of which is the plain face provided with what is known as a "smooth tool finish." Class 125 cast-iron flanged fittings are provided with this type of facing. For steel flanges and fittings, the typical facings shown on page 636 are taken from the American Standard for Steel Pipe Flanges and Flanged Fittings, ASA B16e. The raised face, the lapped, and the large male-and-female facings have the same dimensions, which provide a relatively large contact area. Where metal gaskets are used with these facings, the gasket area should be reduced to increase the gasket compression. American War Standard Ratings for Valves, Flanges, and Fittings, ASA B16e5, (see page 626) may be applied to flanges with these facings *only* if the gasket area is not greater than that of the large tongue-and-groove gasket.

Other commonly used facing types shown on page 636 range in size and contact area in the following order: large tongue-and-groove, small tongue-and-groove, small male-and-female, and ring joint. Because of the small gasket contact area, a tight joint may be secured with the ring-type facing using low bolting loads, thereby resulting in lowered flange stresses.[1] The Sargol and Sarlun facings having lips for seal welding are used frequently for severe service conditions. Seal welding is not always per-

[1] See "The Use of the Ring Joint on Oil or Steam Piping Service," by F. R. Venton and A. W. Marburg, *Heating, Piping and Air Conditioning*, July, August, and September, 1940.

formed since, if properly made, a tight joint often can be maintained without the welded seal, thus facilitating disassembly. Typical facing dimensions for Sarlun and Sargol joints are shown on page 680. Special types of facing of individual design intended for a specific service are numerous but from an economic standpoint it is desirable to use a standard facing wherever possible.

Selection of the type of facing depends to a considerable extent on the nature of the service, but often there is no precise means of determining exactly which facing should be used. Prior experience usually is relied on as a guide. Plain-face joints with red-rubber gaskets have been found satisfactory for temperatures up to 220 F, whereas serrated raised-face joints with asbestos composition gaskets are commonly used for temperatures up to 750 F. For high temperatures and pressures, faces giving a high contact pressure for a given bolt load are customary, such as the tongue-and-groove and ring joints, but an equally successful joint can be made by using a profile-serrated metal gasket contacting the flange facing, which may be the plain male-to-male raised-face type.

Contact Surface Finish.—The surface finish is an important factor in determining the extent to which a gasket must flow to secure an impervious seal. Bolting that is adequate to flow the gasket to form a satisfactory seal with a smooth contact surface may be inadequate to secure a tight joint with a rough surface. The finish may vary from that presented by rough casting to that produced by grinding and lapping. Less gasket flow will be necessary for the latter, of course, than for the former. The finish most frequently provided on cast-iron and steel pipe flanges is the smooth tool finish. A serrated finish frequently is provided for steel flanges, particularly when using an asbestos composition gasket with a wide contact area such as furnished on raised, lapped, or large tongue-and-groove facings. The serrated finish consists of spiral or concentric grooves, usually about $\frac{1}{64}$ in. deep with 32 serrations per inch.

Where metal gaskets are used, a smooth surface produced by grinding or lapping usually is provided. The Sargol and Sarlun facings mate metal to metal without a gasket, in which case a mirrorlike finish is necessary. This is produced usually by grinding and lapping. It is evident that the surface finish varies with the type of contact face and gasket used and therefore should be specified accordingly.

Gaskets.—Since it is expensive to grind and lap joint faces to obtain fluid-tight joints, a gasket of some softer material usually is inserted between contact faces. Tightening the bolts causes the gasket material to flow into the minor machining imperfections, resulting in a fluid-tight seal. A considerable variety of gasket types is in common use, several of which are indicated in Table XLVIII. Soft gaskets, such as cork, rubber, or asbestos, usually are plain with a relatively smooth surface. The semi-metallic design combines metal and a soft material, the metal to withstand the pressure, temperature, and attack of the confined fluid, and the soft material to impart resilience. Various designs involving corrugations, strip-on-edge, metal-jacketed, etc., are available. In addition to the plain, solid, flat-surface metal gaskets, various modified designs and cross-sectional shapes including profile, corrugated, serrated, etc., are used. The object in general has been to retain the advantage of the metal gasket, but to reduce the contact area to secure a seal without excessive bolting load. Gas-filled gaskets have been proposed for high-pressure high-temperature service but sufficient experience has not been had to demonstrate their practicability.[1]

Gasket Materials.—Gaskets are made of materials not chemically affected by the fluid in the pipe and resistant to deterioration by temperature. Typical selections of gasket materials for different services are shown in the accompanying table.

SELECTIONS OF GASKET MATERIALS FOR DIFFERENT SERVICES

Gasket material	Fluid	Temperature
Red rubber..	Steam, air, and water	Up to 220 F
[illegible]	Steam, water, and oil	Up to 750 F
Fiber and paper.	Oil	Up to 200 F
[illegible]etic rubber........................	Oil	Up to 200 F
[illegible]er, corrugated or plain..............	Steam or water	Up to 600 F
[illegible], corrugated or plain................	Steam or water	Up to 1000 F
[illegible]less steel 12 to 14 per cent chromium, corrugated...........................	Steam or water	Up to 1000 F
Hydrogen-annealed furniture iron........	Steam or water	Up to 1000 F
Monel, corrugated or plain..............	Steam or water	Up to 1000 F
Ingot iron, special gasket for ring-type joint	Steam, water, and oil	Up to 1000 F

[1] Asbestos-composition gaskets have been used successfully on 850 F steam service but are not generally considered suitable for temperatures above 750 F.

Gasket Compression.—In the usual type of high-pressure flange joint, a narrow gasket face or contact surface is provided to obtain

[1] For a description of gas-filled gaskets, see *Heating, Piping and Air Conditioning*, September, 1939, p. 568.

higher unit compression on the gasket than is obtainable on full-face gaskets used with low-pressure joints. The compression on this surface, and on the gasket if the gasket is used, before internal pressure is applied, depends on the bolt loading used. In the case of standard raised-face joints of the ASA steel-flange standards, these gasket compressions range from 28 to 43 times the rated working pressure in the 150- to 400-lb standards and from 11 to 28 times in the 600- to 2,500-lb standards for an assumed bolt stress of 60,000 psi. For the lower pressure standards, using composition gaskets, a bolt stress of 30,000 psi usually is adequate. The effect of applying the internal pressure is to decrease the compression on the contact surfaces since part of the bolt tension is used to support the pressure load.

The *initial compression* required to force the gasket material into intimate contact with the joint faces depends upon the gasket material and the character of the joint facing. For soft-rubber gaskets, a unit compression of 4,000 to 6,000 psi usually is adequate. Laminated-asbestos gaskets in serrated faced joints perform satisfactorily if compressed initially at 12,000 to 18,000 psi. Metal gaskets such as copper, monel, and soft iron should be given initial compressions considerably in excess of their yield strengths.[1] Unit pressures of 30,000 to 60,000 psi have been used successfully with metal gaskets. Various forms of corrugated and serrated metal gaskets are available which enable high unit compression to be obtained without excessive bolt loads. These are designed to provide a contact area that will flow under initial compression of the bolts so as to make an initially pressuretight joint, but at the same time the compressive stresses in the body of the gasket are sufficiently low so as to be comparable in load-carrying ability at high temperatures with the long-time strength of the bolting and flange material.

The *residual compression* on the gasket necessary to prevent leakage depends on how effective the initial compression has been in forming intimate contact with the flange joint faces. Tests show[2] that a residual compression on the gasket of only one to two times the internal pressure, with the pressure acting, may be

[1] "Designing Soft Copper Gaskets for High Pressure Equipment," by W. L. Edwards, *Power Plant Equipment*, March, 1937, p. 134.

[2] "Holding Properties of Gaskets Studied," summary by E. Siebel, *VDI*, May 5, 1935, trans. by F. E. Wertheim, *Heating, Piping and Air Conditioning*, July, 1936, p. 367.

sufficient to prevent leakage where the joint is not subjected to bending or to large temperature changes. Since joints in piping customarily must withstand both these disturbing influences, minimum residual gasket compressions of four to six times the working pressure should be provided for in the design of pipe joints.[1]

Relation of Gaskets to Bolting.—There is a tendency as indicated in the ASME Rules for Bolted Flanged Connections[2] to assign lower residual contact pressure ratios ranging from about 1 for soft-rubber gaskets to 6 or 7 for solid-metal gaskets. Whereas these are said to have proved satisfactory for heat-exchanger and pressure-vessel flanges,[3] the more severe service encountered by pipe flanges due to bending moments and large temperature changes are considered by many to warrant designing on the basis of the larger residual gasket compression ratios recommended in the previous paragraph. The lack of understanding of the mechanics of gasket action, the variety of gasket materials, shapes, widths, and thicknesses, the variety of facings used, the variation in flange stiffness, and the uncertainties in bolt pull-up are among the factors that render difficult a precise solution to the problem of gasket design.

The following is abstracted from the ASME Rules for Bolted Flanged Connections:

The minimum required bolt load (W_m) in pounds must be sufficient under maximum operating or working conditions to resist the hydrostatic end force (H) in pounds exerted by the internal working pressure upon the area bounded by the mean diameter of gasket or joint contact surface and, in addition, to maintain a predetermined compression load (H_p) on the gasket which will be sufficient to assure a tight joint. This is equal to

$$W_m = H + H_p = 0.785G^2p + (2b\pi Gmp). \tag{1}$$

The bolt load also must be at least sufficient under atmospheric conditions without internal pressure to exert a load (H_y) to initially seat the gasket or joint contact surfaces so as to assume a tight joint.

$$W_m = H_y = \pi bGyr. \tag{2}$$

The minimum bolt load required is the greater of the values determined from formulas (1) and (2). The bolt load required to seat the gasket may be the

[1] "Gasketed Joints for High-pressure High-temperature Piping Service," by Arthur McCutchan, *Heating, Piping and Air Conditioning*, July, 1941, pp. 423–427.

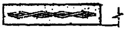
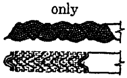
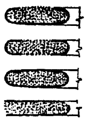
[2] See "Revised Rules for Bolted Flanged Connections," ASME Unfired Pressure Vessel Code.

[3] See "Gasket-loading Constants," by D. B. Rossheim and A. R. C. Markl, *Mech. Eng.*, September, 1943, pp. 647–648.

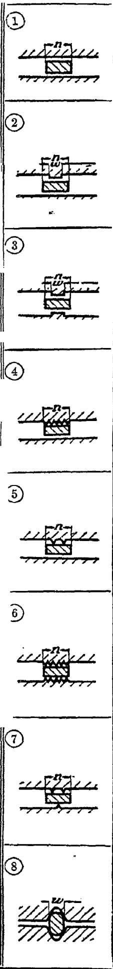
Table XLVIII.—Gasket Materials and Contact Facings for Flanged Joints

(Table UA7 of ASME Rules for Bolted Flanged Connections)

Section A. Gasket factors (m) for operating conditions, yield point y ‖ Se ga

Gasket material		factor	Yield	Sketch and notes
A. Gum rubber sheet		0.50	50(	
B. Cloth-inserted soft rubber, or hard-rubber sheet		0.75	75	
C. Cloth-inserted hard rubber		1.00	1,00(	
D. Vegetable fibre sheet (hemp or jute)		1.50	2,00(	
E. Compressed asbestos, o asbestos composition		2.50	4,500	
F. Wire-mesh reinforced asbestos		2.50	4,500	
G. Corrugated metal, asbestos inserted, or spiral-wound metal, asbestos filled		2.50	4,500	Facing (1), Section *B* only 
H. Corrugated metal jacket, asbestos filled		3.00	6,000	Facing (1), Section *B* only
J. Corrugated metal	(*a*) Copper	3.00	6,000	'acing (1), Section *B* only 
	(*b*) Monel (*c*) Iron (*d*) Soft steel	3.25	7,000	
K. Flat metal jacket, asbestos filled	(*a*) Aluminum (*b*) Copper	3.25	7,000	
	(*c*) Monel (*d*) Iron (*e*) Soft steel	3.50	8,000	
	(*f*) 4–6% chrome (*g*) 11–13% chrome (*h*) KA2S (*j*) Type 316	3.75	9,000	
L. Solid metal	(*a*) Soft aluminum	4.00	10,000	
	(*b*) Soft copper (*c*) Admiralty	4.75	14,000	
	(*d*) Iron (*e*) Soft steel (*f*) Monel	5.50	18,000	
	(*g*) 4–6% chrome (*h*) 11–13% chrome	6.00	21,000	
	(*j*) KA2S (*k*) Type 316	6.50	24,500	

Facing	Effective gasket yield width b
(1)	$\frac{n}{2}$
(2)	$\frac{n+w}{4}$
(3)	$\frac{n}{4}$
(4), (5)	$\frac{n}{3}$
(6), (7)	$\frac{n}{4}$
(8)	$\frac{w}{8}$



greater, particularly in low-pressure designs requiring a high seating load where the load under operating conditions may be insufficient to seat the gasket initially. The factor r, which is the ratio of the maximum allowable bolt stress at operating temperature (S_b) to that at atmospheric temperature (S_a), is included in formula (2) to allow a direct comparison of the operating temperature loading and initial loading, thus permitting reference only to the allowable operating temperature loading in subsequent calculations.

In some cases, bolting in excess of the minimum calculated from formula (1) or (2) is provided in a flange design because of selecting bolts in multiples of four, or to provide bolt spacings within reasonable limits to assure uniform loading. In such cases, the actual bolt load is the force available in pounds (W_a) when the actual total bolt area at the root of the threads (A_b) is stressed to the maximum allowable working stress at the operating temperature (S_b). W_a may be determined as follows:

$$W_a = A_b \times S_b. \tag{3}$$

For purposes of flange design, the flange design bolt load (W) should not be less than the average of the minimum required bolt load (W_m) and the actual bolt load (W_a) or

$$W = \frac{W_m + W_a}{2}. \tag{4}$$

This provides a margin of 50 per cent in excess of the minimum requirements to cover abuse from overbolting. It is assumed that reasonable care will be exercised in tightening bolts since overpulling may dish the flanges seriously, thus affecting the satisfactory operation of the joint. If it is desired that the flange should be capable of withstanding the full available bolt load, the flange should be designed on the basis of the actual bolt load (W_a).

In Table XLVIII (same as Table UA7 of the ASME Rules for Bolted Flanged Connections) are listed several commonly used gasket materials and contact facings with suggested values of m, b, and y. These values are suggestive only and subject to change as the subject of gasket design becomes better understood.

FLANGE DESIGN

Considerable attention has been given to the design of flanges for high-pressure work, in the effort to produce flanges that are amply safe and economically proportioned. The stresses which are set up by the tightening of the bolts and by other causes are rather complicated and require careful analysis.

Support from Hubs.—There are two general types of flanges—those attached to fittings and the loose type such as used with a Van Stone joint or threaded to the pipe. The latter could be merely a flat ring, but the addition of a cylindrical hub increases its strength materially, and all the higher pressure flanges are so The flange attached to a fitting is considerably rein-

forced by the wall of the fitting and is the more favorable case from a standpoint of strength. The tapered hubs of welding-neck flanges approach conditions existing with fitting flanges, although the reinforcement offered by the attached pipe is less than with the heavier wall customary with cast fittings.

For loose flanges, such as lapped companion flanges, it is usually necessary to provide hubs on the flanges which serve to strengthen them in the same way that stiffening angles support flat surfaces on rectangular tanks, etc. In the case of flanged fittings the body of the fitting furnishes the support which is derived from hubs on lapped flanges. It is possible, of course, in the case of loose flanges to increase the thickness of a flat flange sufficiently to obtain the required strength, but, as in the case of many other flat surfaces, this is neither the most economical nor the most desirable method to use.

Standard Dimensions.—Satisfactory proportions for cast-iron flanges have been evolved through long experience backed by computations and tests for 25-, 125-, 250-, and 800-lb service at temperatures not exceeding about 450 F (see pages 592 to 624 for dimensions of ASA standards for these pressures).

Bolting a steel-lapped flange against a 125-lb cast-iron flange or flanged fitting is not recommended as there is danger of cracking the cast-iron flange. Bolting a steel lapped flange against a 250-lb cast-iron flange or flanged fitting is not recommended, but is permitted by the Code for Pressure Piping provided the lap is extended to the inner edge of the bolt holes and carbon-steel bolts are used. For satisfactory results where a cast-iron flange is bolted to a steel flange, it is desirable to use a flat-faced steel flange and a full-face gasket.

The proportions of steel flanges, bolting, and gaskets for service at pressures of 100, 300, 400, 600, 900, 1,500, and 2,500 psi at 750 F, when using carbon-steel flanges and raised face joints, have been evolved and standardized by ASA Sectional Committee B16. Modifications in these ratings to take into account the use of alloy flange materials, the effect of temperatures above and below 750 F, and the reduction in bolt loading possible with the ring-type joint are given in the steel-flange standards. See pages 626 to 680 for dimensions of flanges for these standards.

Flange Materials.—Flanges may be made of rolled or forged steel, cast steel, and in some cases, of plate steel. Cast carbon-steel flanges are covered by ASTM Specifications A95 and A216,

the latter including welding grades. Cast alloy-steel flanges are provided in ASTM Specifications A157 and A217, the latter also consisting of welding grades. Forged carbon-steel flanges may be ordered in accordance with ASTM Specifications A105 and A181, and alloy steel in accordance with A182. Flanges made from plate material are permitted for some conditions in the ASME Boiler Code [see Par. UA 19(*f*) of the Unfired Pressure Vessel Code]. Allowable stresses for flange material as permitted by the Boiler Code are contained in Table VI on page 44. Abstracts of specifications for flange materials in accordance with these specifications are given on pages 529 to 534.

Flange Design Formulas.—Numerous methods and formulas have been developed for design of flanges. Some of these, although presenting a more rigorous and theoretically correct solution, are difficult to apply because of their complexity. One of the simpler but less precise methods which has been used extensively assumes, in accordance with the Rankine stress theory, that if the maximum normal stress exceeds a certain limit, the flange will fail. If more than one stress, such as hoop or radial, exists, the larger of these two principal stresses is the determining factor.

The analyses given in connection with Figs. 19 and 20 apply in the determination of the Rankine criterion in the particular sections designated. In correctly designing a lapped flange in accordance with the Rankine theory, it is necessary to consider the maximum stress existing in at least two different sections, in neither of which should the maximum normal stress exceed a safe limiting value. Figure 20 was developed for use in determining safe proportions for the hubs of lapped flanges, while the methods of Fig. 19 are applicable only to determining safe proportions for the ring part of the flange. All three methods may be applied to their corresponding sections in either flat or hubbed flanges. Obviously, the methods of Fig. 19 are directly applicable to computing the stress in fitting flanges, where again each particular application must be analyzed to locate the weak sections. Two weak sections through flanges cast integrally with thick-walled fittings are illustrated in Fig. 19. The methods of analysis and formula for calculating the maximum stress are indicated with each figure.

In the method of Fig. 20, the hoop stress is taken as the criterion of the failure of the material since it is the principal stress to which the material is subjected. It is a straightforward application of

the Rankine theory. Lapped flanges, whose hubs are proportioned in accordance with this method, are of well-balanced design and have been found entirely satisfactory in service.

The solutions shown in Fig. 19, along with that proposed by J. R. Tanner [see Waters and Taylor paper, reference 2(*a*), page 522] were used by members of the ASA Sectional Committee on Pipe Flanges and Fittings in computing safe dimensions for flat-ring flanges in connection with the original development of the American steel-flange standards. The dimensions so computed were satisfactorily checked later by extensive tests on actual flanges. It is interesting to note that the methods of analysis described are in general agreement with the Dubs formulas[1] used in Europe for computing stress in similar flanges.

The use of alloy-steel heat-treated bolts having a tensile strength of from 100,000 to 150,000 psi is customary in high-pressure work, whereas the tensile strength of castings and forged-steel flanges seldom exceeds 75,000. Although it is not intended that the working stress in the bolt material should approach anywhere near the ultimate strength, yet it is easily possible to set up a stress in the bolts which will exceed the yield point of the flange material if both are designed for equal stresses. Under these circumstances, it would appear good practice to design flanges for a working stress enough lower than the bolt stress to compensate for the difference in strength of the two materials.

Although bolt forces generally are considered as acting on the circumference of the bolt circle, there is evidence that as deflection of the flange takes place, the load is no longer uniformly distributed on the underside of the nuts but shifts toward the inner edge of the bolt holes. This condition is clearly borne out in tests where the imprint of the nuts has been left distinctly in the flange material, as illustrated in Fig. 21. If this condition is assumed to exist in service, a marked reduction in flange stress will be effected through the shortening of the moment arm of the deforming couple.

This shifting of the bolt loading, however, sets up an eccentric loading with a resulting bending moment which may produce a high bolt stress. Because only a comparatively small shifting of the load will produce a bolt stress equal to the yield point of the bolting material and also because of the indeterminate nature of this action in any particular flange, the bolt load in flange

[1] See report of the International Technical Conference on Pipes and Fittings held at Zurich, Switzerland, Oct. 26 to Nov. 9, 1926.

calculations is generally assumed to remain at the bolt circle circumference.

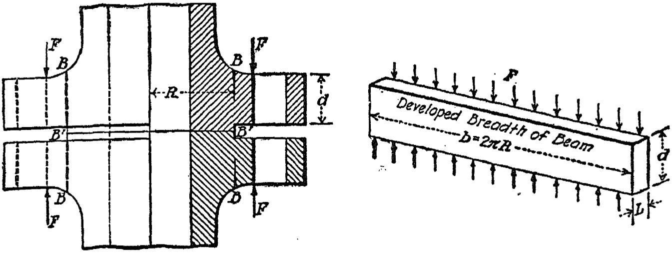


FIG. 19a.—Stress conditions in ring section B-B' of raised face and lapped flanges.

Formula Used in Determination of Maximum Stress in Circular Section B-B'.

$$S = \frac{6 \times F \times L}{b \times d^2} = \frac{6 \times a \times L \times t}{b \times d^2}$$

where S = bending stress, psi.
6 = constant from section modulus.
F = force in pounds = at.
a = total bolt-root area, sq in.
t = bolt tension, psi.
L = moment arm from inner edge of bolt hole to section B-B', in.
b = breadth of beam = $2\pi R$, in.
d = depth of beam = flange thickness, in.

Formula Used in Determination of Maximum Stress in Diagonal Section C-C'. The formula developed below for calculating the maximum stress obtaining

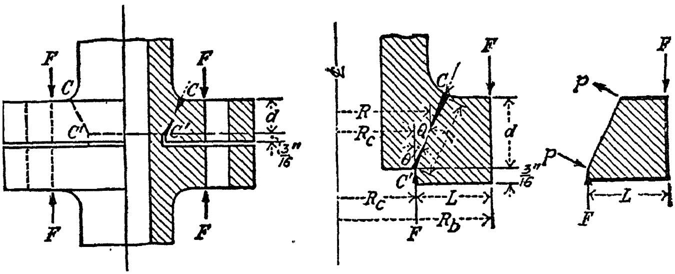


FIG. 19b.—Stress conditions in diagonal ring section C-C' of flanged fitting. Small male-female type of joint.

in the diagonal ring section C-C' is particularly applicable to the small male-female fittings but can be applied equally well to any fittings which are sus-

pected of being weak in this section. An isometric view of a unit length of section C-C' is given in Fig. 19c.

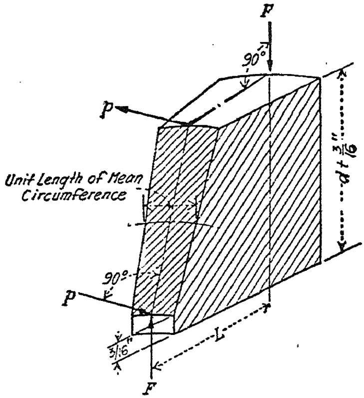


FIG. 19c.—Isometric view of length of section C-C' small female fitting.

The symbols used in the development of the formula are

S = bending stress, psi.
$F = at$ = total bolt load, lb.
a = total bolt area at root of threads, sq in.
t = bolt tension, psi.
M = bending moment, in. lb.
$L = R_b - R_c$ = moment arm of external-force couple composed of bolt forces and gasket reactions.
R_b = radius of a circle tangent to inner edge of bolt holes, in.
R_c = outer radius of small female, in.
$R = R_c + Q$ = radius of mean circle, in.
$Q = \sin \theta h/2$.
$h = \dfrac{d}{\cos \theta}$ = depth of section C-C'.
θ = angle between center line of fitting and slope of section C-C'. (This angle may be obtained by laying out flange to scale and assuming a fillet radius.)
c = distance from neutral axis to outer fiber of beam section, in. (in this case $c = d/2$).
d = flange thickness minus depth of female ($3/16$ in. in the case of the small male-female fittings), in.

From the above relations it follows that the total bolt load acting upon the flange is

$$F = at.$$

(Heavily shaded area represents portion of diagonal ring section C-C' or the critical section of the small female-fitting flange.)

The bolt forces and resisting stresses may be found per unit length of the circumference of the mean circle of radius R by dividing the total forces and stresses by $2\pi R$.

The external moment of the couple composed of the bolt forces and the equal and opposite gasket reaction per unit length of the mean circumference is

$$M = \frac{F}{2\pi R} L = \frac{atL}{2\pi R}.$$

From the properties of rectangular beams, the moment of resistance of the rectangular section of depth h per unit length of the mean circumference is the product of the section modulus and the maximum stress in the outer fibers of the flange material is

$$M = S\frac{I}{c} = S\frac{bh^2}{6}.$$

Substituting the value of $h = \frac{d}{\cos\theta}$

$$M = S\frac{bd^2}{6\cos^2\theta}.$$

Equating the external and internal moments and noting that, for a unit length of the mean circumference b, the breadth of the section is equal to unity, the following equations result:

$$\frac{Sd^2}{6\cos^2\theta} = \frac{atL}{2\pi R}$$

or

$$S = \frac{6\cos^2\theta atL}{2\pi Rd^2},$$

$$= 0.96\cos^2\theta\frac{aL}{Rd^2}t.$$

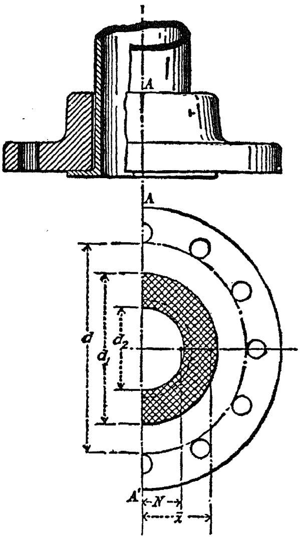


FIG. 20.—Stress conditions in lapped flanges due to tightening of bolts.

Bolt forces are considered as acting at inner edge of bolt holes rather than on bolt circle, because as deflection takes place the load is no longer uniformly distributed on the under side of the nuts. This is shown by the imprints of the nuts as illustrated in Fig. 21. The circle tangent to the inner edge of the bolt holes was taken as being a more representative load line for the average condition of loading.

Method of Calculating Stress in Lapped Flange Section A-A'.—This method assumes: (1) That the flange may be treated as a beam acted upon by all forces occurring on one side of a diameter: (2) that one-half the total bolt load may be considered as concentrated at the center of gravity of a semicircle tangent to the inside edge of the bolt holes; (3) that the reacting forces may be considered as concentrated at the center of gravity of the semicircular pressure area which is the lapped-over end of the pipe. The maximum stress in the hub is obtained by solving the formula $S = M/J$, where S is the stress sought in pounds per square inch, M is the product of one-half the total bolt load in pounds times the

distance in inches between the centers of gravity described above $= F/2(x - N)$; J is the section modulus of section AA' of the flange, taken through the bolt holes in inches cubed. Distance, center of gravity of semicircle to section $A\text{-}A' = x = d/\pi$. Distance, center of gravity of semicircular pressure area to section $A\text{-}A' = N = \frac{2(d_1^3 - d_2^3)}{3\pi(d_1^2 - d_2^2)}$.

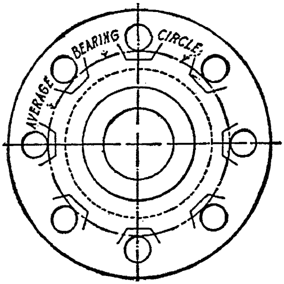


FIG. 21.—Shifting action of points of application of bolt forces upon deflection.

ASME Rules for Bolted Flanged Connections.[1]—These rules are based on the more rigorous and detailed flange analysis presented in the Waters-Taylor and subsequent papers.[2] Critical stresses are assumed to be the radial and hoop stresses at the inside diameter of the flange ring, and the axial hub stress at the surface of junction with the ring. In addition to taking full account of the radial stress, the effect of the longitudinal bending stress in the hub also is evaluated. Extensive tests and design experience over a number of years have indicated the validity and practicability of these design methods. War standard ratings for steel

[1] See "Revised Rules for Bolted Flanged Connections," ASME Unfired Pressure Vessel Code.

[2] For further data, see

(a) "The Strength of Pipe Flanges," by E. O. Waters and J. Hall Taylor, *Mech. Eng.*, Vol. 49, pp. 513–542, mid-May 1927. Also see discussion "Methods of Determining the Strength of Pipe Flanges," Vol. 49, pp. 1340–1347, December, 1927.

(b) "Formulas for Stresses in Bolted Flanged Connections," by E. O. Waters, D. B. Wesstrom, D. B. Rossheim, and F. S. G. Williams, *Trans. ASME*, FSP 59-4, p. 61. Also see extensive bibliography appended thereto and papers on high-temperature joints, December, 1937, meeting of ASME.

(c) "Tests of Heat-exchanger Flanges," by D. B. Rossheim, E. H. Gebhardt, and H. G. Oliver, *Trans. ASME*, FSP 60-10, p. 305, May, 1938.

(d) "Design of Flanged Joints for Valve Bonnets," by J. D. Mattimore, N. O. Smith-Petersen, and H. C. Bell, *Trans. ASME*, FSP 60-9, p. 297, May, 1938.

valves, flanges, and fittings given in the American War Standard B16e5 were based on this method of flange analysis.

The Rules for Bolted Flanged Connections may be applied to flanges of any diameter, but they are not applicable to flanges having gaskets extending beyond the bolt circle. It is not intended to prohibit use of flanges with full-face gaskets, however, provided they are designed in accordance with good engineering practice. The rules are set up on the basis that the general flange proportions are known so that determination of the flange thickness, hub diameter, and hub height is a cut-and-try process until a combination is attained which will give stresses within the limits allowed by the Code rules. Practical limitations on hub diameter and hub height enable selection of tentative dimensions fairly close to the final result, however, and approximations of flange thickness can be made by means of charts which reduce the amount of computation required.[1] The following abstract of the revised rules for design of bolted flanged connections is taken from Par. UA18 to UA24 of the 1943 ASME Unfired Pressure Vessel Code.

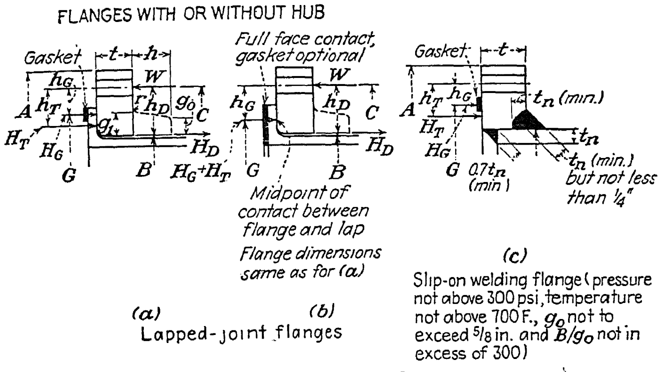


FIG. 22.—Loose-type flanges.

Flange Types.—For purposes of calculation, flanges are classified in two types, loose flanges such as shown in Fig. 22, and integral-type flanges, shown in Fig. 23.

Calculation of Flange Stresses.—The stresses in the flange, determined in accordance with the following formulas, shall not exceed the values of flange design stresses specified in the following paragraph (for definition of symbols, see pages 526 to 529).

[1] See "Modern Flange Design," 1941 Edition, Bulletin 421 of the Taylor Forge and Pipe Works, Chicago, Illinois, or the section on flange design in *General Catalog* 423.

1. For integral-type flanges and all hubbed flanges:

$$\text{Longitudinal hub stress } S_H = \frac{fM_0}{Lg_1^2B}.$$

$$\text{Radial flange stress } S_R = \frac{(1.33te + 1)M_0}{Lt^2B}.$$

$$\text{Tangential flange stress } S_T = \frac{YM_0}{t^2B} - ZS_R.$$

2. For ring flanges of the loose type:

$$S_T = \frac{YM_0}{t^2B} \qquad S_R = 0 \quad \cdot \; S_H = 0.$$

Flange Design Stresses.—The stresses in the flange as calculated by these formulas shall not exceed the values indicated as follows:

Longitudinal hub stress S_H not greater than $1.5S_f$.

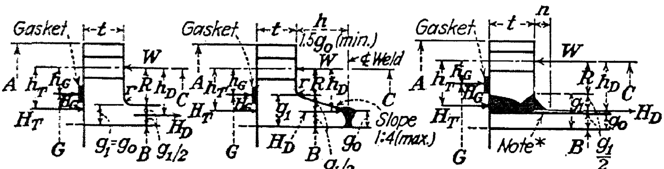
Radial flange stress S_R not greater than S_f.

Tangential flange stress S_T not greater than S_f.

Also $\qquad S_H + S_R$, not greater than S_f.

$\qquad S_H + S_T$, not greater than S_f.

When flange parts are subject to shearing stresses such as the lap in the case of a loose flange (Fig. 22*a*) or the weld in Fig. 22*c*, the shearing stress shall not exceed 0.8 times the maximum allowable working stress S_f for the lap, weld, or other part in shear. The shearing stress shall be calculated on the basis of H_p or H_y as defined on page 513, whichever is greater.



(a) Integral flange, hub of uniform thickness

(b) Butt-welding flange with single tapered hub having slope not greater than 1 to 4

(c) Slip-on welding flange (where limits of FIG. 22c are exceeded)

FIG. 23.—Integral-type flanges.

*NOTE.—The minimum for either leg is $0.25g_0$, but not less than $\frac{3}{16}$ in. This weld may be machined to a corner radius of at least $0.25g_1$, but not less than $\frac{3}{16}$ in., the same as for detail (*a*).

Flange Moments.—Moments acting upon the flange and used in the calculation of flange stresses shall be determined as follows:

(*a*) For loose-type flanges such as shown in Figs. 22*a* and 22*c*, the total moment shall be determined as for integral-type flanges in (*b*), except that the force H_D shall be considered to act at the inside diameter of the flange, then

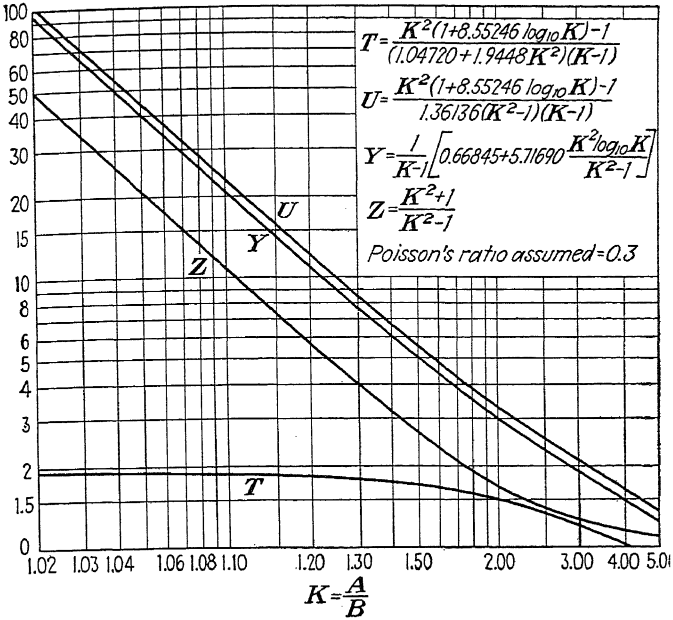


FIG. 24.—Flange design. Values of T, U, Y and Z (terms involving K).

For loose-type flanges such as shown in Fig. 22b, the total moment shall be as determined for integral-type flanges in (b), except that the force H_D shall be considered to act at the inside diameter of the flange, in which case

$$h_D = C - B$$

$$h_G = h_T = \frac{C - G}{2}.$$

(b) For integral-type flanges such as shown in Fig. 23, the total moment shall be at least equal to the sum of the moments acting upon the flange, or

Flange loads ×	lever arms =	moments
$H_D = 0.785B^2p$	$h_D = R + \frac{g_1}{2}$	$M_D = H_D \times h_D$
$H_T = H - H_D$	$h_T = \frac{R + g_1 + h_G}{2}$	$M_T = H_T \times h_T$
$H_G = W - H$	$h_G = \frac{C - G}{2}$	$M_G = H_G \times h_G$

and total moment $M_0 = M_D + M_T + M_G$.

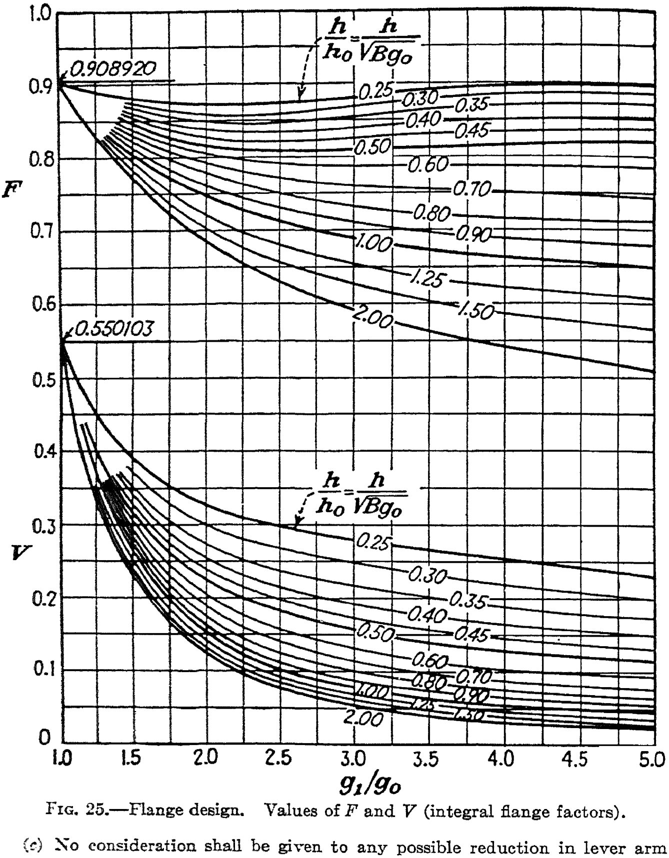


FIG. 25.—Flange design. Values of F and V (integral flange factors).

(c) No consideration shall be given to any possible reduction in lever arm due to cupping of flanges or to inward shifting of the line of action of the bolts as a result thereof.

Nomenclature and Values

H = total hydrostatic end force, lb = $0.785G^2p$ (see page 513).

W = flange design bolt load, lb (see page 515), W_m = minimum bolt load, W_a = actual bolt load.

p = maximum working pressure, psi.

S_b = maximum allowable bolt stress at operating temperature, psi = 1.25 times values given in Table VI, Chap. II, page 44.

(*Continued on p.* 527)

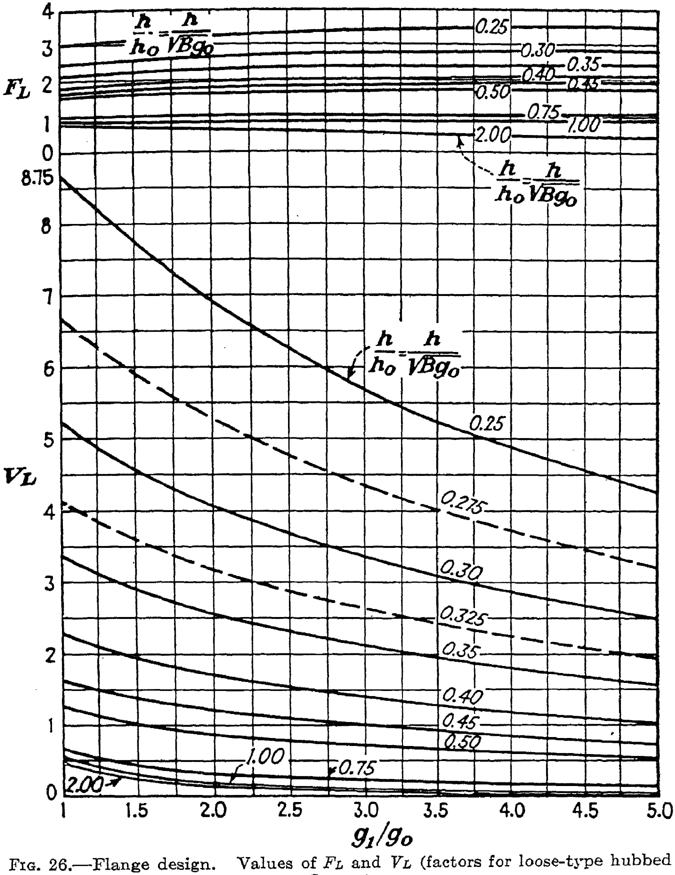


FIG. 26.—Flange design. Values of F_L and V_L (factors for loose-type hubbed flanges).

S_a = maximum allowable bolt stress at atmospheric temperature, psi = 1.25 times values given in Table VI, Chap. II, page 44.

A_b = total cross-sectional area of bolts at root of thread or section of least diameter under stress, sq in.

A = outside diameter of flange, in.

B = inside diameter of flange, in. When B is less than $20g_1$, B_1 may be substituted for B in the formula for longitudinal hub stress, where $B_1 = B + g_1$ for loose-type hubbed flanges and for integral-type flanges when f is less than 1.

(*Continued on p.* 528)

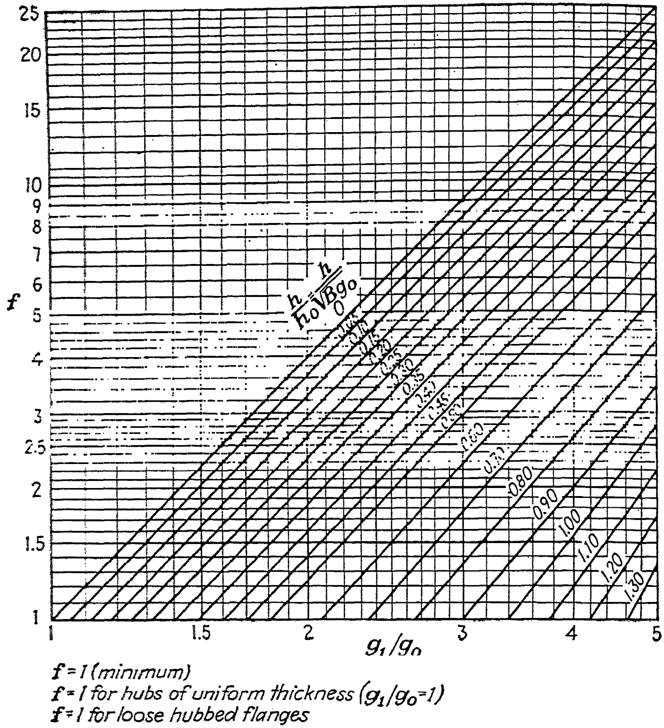


FIG. 27.—Flange design. Values of f (hub stress correction factor).

$B_1 = B + g_0$ for integral-type flanges when f is equal to or greater than 1.

C = bolt circle diameter, in.

R = radial distance from bolt circle to point of intersection of hub and back of flange, in. (integral and hubbed flanges).

g_1 = thickness of hub at back of flange, in.

g_0 = thickness of hub at small end, in.

M_0 = total moment acting on flange, in. lb.

K = A/B or ratio of outside diameter to inside diameter of flange.

T, U, Y, Z = terms involving K, obtain from Fig. 24.

F, V = factors for integral-type flange, obtain from Fig. 25.

F_L, V_L = factors for loose-type hubbed flanges, obtain from Fig. 26.

f = hub stress correction factor (for integral flanges) obtain from Fig. 27. For values below chart of Fig. 27, use $f = 1$.

(*Continued on p.* 529)

h = hub length, in.
$h_0 = \sqrt{Bg_0}$, in.
t = flange thickness, in.

$$L = \frac{te + 1}{T} + \frac{t^3}{d}.$$

$$d = \frac{U}{V} h_0 g_0^2 \text{ for integral-type flanges.}$$

$$= \frac{U}{V_L} h_0 g_0^2 \text{ for loose-type flanges.}$$

$$e = \frac{F}{h_0} \text{ for integral-type flanges.}$$

$$= \frac{F_L}{h_0} \text{ for loose-type flanges.}$$

S_f = for steel, maximum allowable working stress for flange material, psi = 1.25 times values given in Table VI, Chap. II, page 44.
G = mean diameter of gasket, in.
y = yield point of gasket material, psi (see Section A, Table XLVIII).
b = effective gasket yielding width, in. (see Section B, Table XLVIII).
$2b$ = effective gasket pressure width, in.
m = unit gasket compression factor (see Section A, Table XLVIII).

ASTM Standard Specifications for FORGED- OR ROLLED-STEEL PIPE FLANGES FOR GENERAL SERVICE

Serial Designation A181-42

Abstracted[1]

These specifications cover two grades of forged- or rolled (carbon)-steel flanges to be attached to piping or pressure vessels for general service. Unless otherwise specified on the order, a certification that the material conforms to the requirements of these specifications shall be the basis of acceptance of the material.

The steel shall be made by either or both the open-hearth or the electric-furnace process and shall conform to the following requirements as to chemical composition and tensile properties:

CHEMICAL COMPOSITION

	Grade I	Grade II
Carbon, max, per cent	0.35[1]	[1]
Manganese, max, per cent	0.90	0.90
Phosphorus, max, per cent	0.05[2]	0.05[2]
Sulphur, max, per cent	0.05[2]	0.05[2]

[1] The carbon content of Grade II material shall be a matter of agreement between the manufacturer and the purchaser. When flanges will be subject to fusion welding, the carbon content shall not exceed 0.35 per cent. When the carbon is restricted to 0.35 per cent, max, it may be necessary to add silicon to the composition for Grade II and for the heavier thicknesses of Grade I flanges in order to meet the required tensile properties. The silicon content shall not exceed 0.30 per cent.
[2] The phosphorus and sulphur contents on check analysis may each be 0.055 per cent.

[1] For complete specifications, reference may be made to ASTM A181 (see note, p. 369).

PHYSICAL PROPERTIES

	Grade I	Grade II
Tensile strength, min, psi........	60,000	70,000
Yield point, min, psi............	30,000	36,000
Elongation in 2 in., min, per cent	22.0	18.0
Reduction of area, min, per cent.	35.0	24.0

Heat Treatment.—No heat treatment is required for flanges offered under these specifications.

Tension Tests.—One tension test shall be made from one flange, or at the option of the manufacturer from the billets or forging bar entering into the finished product, from each melt. If specimens are cut from the billets or forging, they shall have undergone approximately the same working and treatment as the flanges they represent. A standard 2-in.-gage-length, ½-in.-diameter test specimen shall be used.

ASTM Standard Specifications for FORGED- OR ROLLED-STEEL PIPE FLANGES, FORGED FITTINGS, AND VALVES AND PARTS FOR HIGH-TEMPERATURE SERVICE

Serial Designation A105-40 (ASA G17.3)

Abstracted[1]

These specifications cover two grades of material for forged- or rolled (carbon)-steel flanges, forged fittings, and valves and parts for high-temperature service. Unless otherwise specified in the order, a certification that material conforms to these specifications shall be the basis of acceptance of the material.

The steel shall be made by either or both the open-hearth or the electric-furnace process and shall conform to the following requirements as to chemical composition and tensile properties:

CHEMICAL COMPOSITION

	Grades I and II
Carbon, per cent......................................	[a]
Manganese, max, per cent.............................	0.90
Phosphorus, max, per cent............................	0.05
Sulphur, max, per cent................................	0.05

[a] When flanges will be subject to fusion welding, the carbon content shall not exceed 0.35 per cent. When the carbon is restricted to 0.35 per cent, max, it may be necessary to add silicon to the composition for grade II and for the heavier thicknesses of Grade I flanges in order to meet the required tensile properties. The silicon content shall not exceed 0.30 per cent.

[1] For complete specifications, reference may be made to ASTM A105 (see note, p. 369).

PHYSICAL PROPERTIES

	Grade I	Grade II
Tensile strength, min, psi	60,000	70,000
Yield point, min, psi	30,000	36,000
Elongation in 2 in., min, per cent	25.0	22.0
Reduction of area, min, per cent	38.0	30.0

Manufacturing Practice.—Material for forgings shall consist of blooms, billets, slabs, or bars either forged or rolled from an ingot, and cut to the required length by a process that will not produce injurious defects in the forging. Forgings shall be brought as nearly as practicable to the finished shape and size by hot working and shall be so processed as to cause metal flow in the direction most favorable for resisting the stresses encountered in service.

Heat Treatment.—Flanges, forged fittings, valves, and parts shall be heat-treated by annealing or normalizing (see ASTM E44 for description of heat-treating procedures). Forgings shall not be quenched in any liquid medium.

Tension Tests.—One tension test shall be made from each melt in each heat-treatment charge. The tension specimens taken from forgings whose size permits shall be machined to the form and dimensions of standard ½-in.-diameter, 2-in.-gage-length tensile specimens (see page 309). In the case of small sections the dimensions of the tensile specimen shall be proportional to those of the standard specimen. The forging manufacturer shall supply suitable test specimens from the flange or, at his option, furnish test blanks prepared from the fillets or forging bars which have undergone approximately the same working and heat treatment as the finished product.

ASTM Standard Specifications for
FORGED OR ROLLED ALLOY-STEEL PIPE FLANGES, FORGED FITTINGS, AND VALVES AND PARTS
For Service at Temperatures from 750 to 1100 F

Serial Designation A182-44

Abstracted[1]

These specifications cover forged or rolled alloy-steel pipe flanges, forged fittings, and valves intended for service at metal temperatures from 750 to 1100 F. (Carbon-steel pipe flanges are covered by ASTM Specifications A105 and A181 abstracted on pages 529 and 530). It is not intended to limit these specifications to the 13 materials whose compositions are listed nor to imply that all of the alloy steels contained herein are suitable for the entire temperature range.

Certification of Test.—Where mutually agreed, a certification that the material conforms to the requirements of this specification as specified for a particular grade may be the basis of acceptance of the material.

[1] For complete specification, reference may be made to ASTM A182 (see note, p. 369).

Manufacturing Practice.—Forgings shall be brought as nearly as practicable to the finished shape by hot working and shall be so processed as to cause metal flow during the hot-working operation in the direction most favorable for resisting the stresses encountered in service. Plate flanges are excluded by this requirement.

Heat Treatment.—Procedures for heat treating by annealing or normalizing and drawing are covered in detail. A stabilizing treatment for austenitic steels also is described (see ASTM E44 for heat-treating procedures).

Chemical Composition.—Materials shall conform to chemical requirements of accompanying table.

Tension Tests.—Forgings shall conform to the following minimum tensile requirements at room temperature.

TENSILE REQUIREMENTS

Grade	Tensile strength, pounds per square inch	Yield point, pounds per square inch	Elongation in 2 in., per cent	Reduction of area, per cent
F1	70,000	45,000	25	35
F3	70,000	40,000	20	40
F4	90,000	70,000	18	50
F5	90,000	65,000	22	50
F6	85,000	55,000	25	60
F7	100,000	75,000	18	50
F11	100,000	70,000	17	30
F8	75,000	30,000	45	50
F10	80,000	35,000	40	60
F12	100,000	45,000	50	60

Hydrostatic Tests.—Valve bodies and fittings and other pressure-containing parts shall be tested after machining to a hydrostatic pressure prescribed in the following table. No hydrostatic test is required for welding-neck or other flanges.

HYDROSTATIC TEST PRESSURES

Primary Service Pressure Rating at 900 F, Psi	Standard Hydrostatic Test Pressure, Psi
300	900
400	1,200
600	1,800
900	2,400
1,500	4,200
2,500	7,200

Macro-etch Tests.—In case of question as to the soundness of material in any lot of forgings a macro-etch shall be made for each melt present in the lot. Etchings tests shall show sound and reasonably uniform material free from injurious lamination, cracks, segregations, and similar objectionable defects.

Chemical Requirements, ASTM A182

Type	Ferritic steels				
Identification symbol[1]	F1	F3	F4	F5	F6
Grade..	Carbon-molybdenum	Chromium-molybdenum	Nickel-chromium-molybdenum	4 to 6 per cent chromium	13 per cent chromium
Carbon, per cent......	0.35 max	0.15 to 0.25	0.35 to 0.45	0.25 max.	0.12 max
Manganese, per cent..	0.30 to 0.80	0.40 to 0.60	0.50 to 0.80	0.30 to 0.50	0.50 max
Phosphorus, per cent..	0.04 max	0.04 max	0.04 max	0.03 max	0.03 max
Sulphur, per cent.....	0.05 max	0.05 max	0.05 max	0.03 max	0.03 max
Silicon, per cent......	0.20 to 0.50	0 45 to 0.75	....	0.50 max	0.50 max
Nickel, per cent......			1.50 to 2.00		0.50 max
Chromium, per cent..	....	1.50 to 2.00	0.50 to 0.80	4.00 to 6.00	11.5 to 13.5
Molybdenum, per cent	0.40 to 0.60	0.60 to 0.80	0.30 to 0.40	0.45 to 0.65[2]	
Tungsten, per cent...				0.75 to 1.25[2]	
Titanium, per cent...				1.00 max[3]	
Columbium, per cent.					

Type	Ferritic steels		Austenitic steels		
Identification symbol[1]	F7	F11	F8	F8m	F8c
Grade..	Chromium-molybdenum	Chromium-manganese-molybdenum	18 chromium 8 nickel	18 chromium 8 nickel type with molybdenum	18 chromium 8 nickel type with columbium
Carbon, per cent......	0.25 to 0.35	0.25 to 0.40	0.08 max	0.08 max	0.08 max
Manganese, per cent..	0.40 to 0.60	1.20 to 1.50	2.50 max	2.50 max	2.50 max
Phosphorus, per cent..	0.04 max	0.04 max	0.035 max	0.035 max	0.035 max
Sulfur, per cent.......	0.05 max	0.05 max	0.030 max	0.030 max	0.030 max
Silicon, per cent......	0.15 to 0.45	0.40 to 0.60	0.85 max	0.85 max	0.85 max
Nickel, per cent......	....	....	8.00 min	10.00 min	9.50 min
Chromium, per cent...	0.80 to 1.10	0.60 to 0.90	18.00 min	17.00 min	17.00 min
Molybdenum, per cent	0.15 to 0.25	0.25 to 0.40		2.00 min	
Tungsten, per cent....					
Titanium, per cent....					
Columbium, per cent..					

Type	Austenitic steels		
Identification symbol[1]	F8t	F10	F12
Grade..	18 chromium 8 nickel type with titanium	20 nickel 8 chromium	20 nickel 8 chromium
Carbon, per cent.......	0.08 max	0.10 to 0.20	0.45 max
Manganese, per cent....	2.50 max	0.50 to 0.70	0.50 to 0.70
Phosphorus, per cent...	0.035 max	0.03 max	0.03 max
Sulfur, per cent........	0.030 max	0.03 max	0.03 max
Silicon, per cent........	0.85 max	1.00 to 1.40	0.90 to 1.25
Nickel, per cent........	9.00 min	19.0 to 22.0	19.0 to 22.0
Chromium, per cent....	17.00 min	7.00 to 9.00	7.00 to 9.00
Molybdenum, per cent..			
Tungsten, per cent.....			
Titanium, per cent.....			
Columbium, per cent...			

[1] Alloys have been numbered so that similar alloys in the corresponding specifications for pipe, castings, and bolting will bear the same number for alloys numbered 10 or below. No alloys F2 or F9 are included herein as the particular compositions corresponding to these numbers are not included in these specifications.

[2] Either molybdenum or tungsten shall be used.

[3] May be added.

[4] Grade F8c shall have a columbium content of not less than ten times the carbon content and not more than 1.00 per cent.

[5] Grade F8t shall have a titanium content of not less than five times the carbon content and not more than 0.60 per cent.

Marking.—Identification marks showing the manufacturer's symbol or name, designation of service rating, the grade as F1, F3, etc., and the size shall be legibly stamped on each forging in accordance with the requirements of MSS Standard Practice SP 25 (see abstract, page 687).

Ladle Analysis.—A ladle analysis shall be made of each melt of steel to determine the percentages of the elements specified and the results reported to the purchaser. A check analysis may be made by the forgings manufacturer and reported where the ladle analysis is not available.

Check Analysis.—Check analysis may be made by the purchaser from the finished product or from broken tensile-test specimens.

BOLTING FOR FLANGED JOINTS

Bolting Materials.—Ordinarily carbon-steel bolts are suitable for use with cast-iron flanges where their relatively low tensile strength affords some protection to the flanges against overstress and consequent breakage. While carbon-steel bolts will resist the calculated stresses, the Code for Pressure Piping recommends that alloy bolt studs be used with steel flanges for power piping and district-heating steam-service pressures in excess of 160 psi and/or temperatures over 450 F. For gas and air piping in power, industrial, and gas-manufacturing plants wherever located, or anywhere within the boundaries of cities and villages, the code requires alloy-steel studs if the pressure exceeds 300 psi, except that high-strength carbon steel bolts may be used above 300 psi at ordinary temperatures. The Code permits use of commercial-steel or wrought-iron machine bolts for pressures up to 250 psi and/or temperatures up to 450 F for oil piping and up to 800 psi for cold-oil trunk lines. For refrigeration piping, carbon-steel bolts are permitted for pressures of 300 psi and lower.

Working stresses in bolting materials for temperatures up to 750 F have been successfully established as a fraction of the room-temperature tensile strength. The bolting in the line flanges of the ASA steel flange standards is designed to give a stress not exceeding 7,000 psi, considering internal working pressure only. The internal pressure is assumed to act upon an area circumscribed by the periphery of the contact surface. In the case of valve-bonnet joints, cleanout flanges, etc., an allowable working stress of 9,000 was established on the same basis. The working stresses in the bolting must, of course, be higher than just sufficient to balance the internal pressure. An initial bolt stress of 30,000 to 60,000 psi to set the gasket in place is common practice (see discussion of gasket compression, page 511, and turning effort to tighten bolts, page 535).

The selection of working stresses for bolting at temperatures of 850 to 1100 F requires knowledge of the creep characteristics of the particular bolting material. A special type of creep test, which has been developed for the study of flange bolting to give more direct information than the usual creep test employing constant loading, is termed a "relaxation" test.[1] It simulates conditions as they actually exist in a bolted flanged connection, where the initial elastic strain is more or less gradually reduced by plastic deformation to a stress which the material is capable of supporting at the operating temperature. For further discussion of working stresses at high temperatures, see Chap. III, page 336. For allowable bolting stresses in accordance with the ASME Boiler Code, see pages 44 and 513.

Bolting Design.—The initial tension secured in the bolting depends on the method of control,[2] if any, used in tightening the bolts. Experienced pipe fitters usually tighten the bolts in high-pressure flanges to give a bolt tension of approximately 60,000 psi with raised-face joints. In the case of ring-type joints, a bolt tension of 30,000 psi has been found adequate. It is entirely practicable to control the initial bolt tension by measuring the elastic elongation of the bolts, but this refinement is not needed in the majority of installations. Bolt tension also may be controlled to some extent by use of torque wrenches adjusted to permit application of only sufficient turning effort to secure the desired bolt stress.

Turning Effort.—The turning effort required to secure bolt stresses of 30,000 and 60,000 psi with different size alloy-steel bolts threaded in accordance with the American Standard Screw Threads for High-strength Bolting, ASA B1.4, is given in Table XLIX. This table is based on tests conducted by Crane Company and represents results obtained with well-lubricated threads and bearing surfaces. Tests with no lubricant on threads and bearing surfaces showed an increase of almost 100 per cent in the torque required to secure a given bolt stress.

Special thread lubricants are available both for temperatures below 500 F and from 500 to 1000 F. Such lubricants not only

[1] "Interpretation and Use of Creep Results," by J. J. Kanter, *Trans. ASM*, December, 1936, p. 898.

[2] "Flanged Pipe Joints—Methods for Applying and Determining Bolt Stresses," by E. C. Petrie, *Heating, Piping and Air Conditioning*, June, 1936, p. 303.

facilitate initial tightening but permit easier disassembly after service.

TABLE XLIX.—TURNING EFFORT TO TIGHTEN 8-PITCH-THREAD BOLTS

Nominal diameter of bolt, inches	Number of threads per inch	Area at root of thread, square inches	Stress: 30,000 lb per sq in. Torque, foot-pounds	Stress: 30,000 lb per sq in. Force per bolt, pounds	Stress: 60,000 lb per sq in. Torque, foot-pounds	Stress: 60,000 lb per sq in. Force per bolt, pounds
1/2	13	0.126	30	3,780	60	7,560
5/8	11	0.202	60	6,060	120	12,120
3/4	10	0.302	100	9,060	200	18,120
7/8	9	0.419	160	12,570	320	25,140
1	8	0.551	245	16,530	490	33,060
1 1/8	8	0.728	355	21,840	710	43,680
1 1/4	8	0.929	500	27,870	1,000	55,740
1 3/8	8	1.155	680	34,650	1,360	69,300
1 1/2	8	1.405	800	42,150	1,600	84,300
1 5/8	8	1.680	1100	50,400	2,200	100,800
1 3/4	8	1.980	1500	59,400	3,000	118,800
1 7/8	8	2.304	2000	69,120	4,000	138,240
2	8	2.652	2200	79,560	4,400	159,120
2 1/4	8	3.423	3180	102,690	6,360	205,380
2 1/2	8	4.292	4400	128,760	8,800	257,520
2 3/4	8	5.259	5920	157,770	11,840	315,540
3	8	6.324	7720	189,720	15,440	379,440

Use of 8-pitch-thread Series.—It is customary to use bolt studs with two nuts in high-pressure flanged joints. Bolt studs are threaded in accordance with the American Standard Screw Threads for High-strength Bolting, ASA B1.4, abstracted on page 548. This standard provides eight threads per inch for sizes 1 in. and larger. A comparison of the bolt-root areas given in Table L, page 547, shows that in the larger bolt sizes the use of the 8-pitch-thread series gives an increase in cross-sectional area at the root of threads of approximately 15 per cent over the coarse thread series. Hence the force exerted by the bolt when tightened to give a certain stress is correspondingly greater for the bolts with 8 threads per inch. The greater mechanical advantage obtained with 8-pitch threads over coarser threads is deemed to be of considerable assistance in making up joints having large-diameter bolts. Although 12 threads per inch have been used

for high-temperature stud bolts in some cases, the 8-pitch-thread series is considered to represent a good design satisfactory for most high-pressure and high-temperature work.

ASTM Standard Specifications for ALLOY-STEEL BOLTING MATERIAL FOR HIGH-TEMPERATURE SERVICE

Serial Designation A96-44 (ASA G17.2)

Abstracted[1]

These specifications cover alloy-steel bolting material for pressure vessels, valves, flanges, and fittings for high-temperature service. (For allowable stresses at various temperatures, reference may be made to Table VI on page 44).

The term "bolting material" as used in these specifications covers rolled, forged, or cold-drawn bars and bolts, screws, studs and bolt-studs. Unless otherwise specified in the order, a certification that bolts, screws, and studs for valves and other fittings intended for stock and other purposes requiring assembly in the manufacturer's plant conform to these specifications shall be accepted in lieu of tests herein specified.

Nuts.—Carbon-steel nuts suitable for use with bolting material covered by these specifications are described in ASTM Specification A 194, abstracted on

PERMISSIBLE VARIATIONS IN SIZE OF HOT-ROLLED BARS

Specified size, in.	Permissible variations from specified size, in.		Out-of-round, in
	Over	Under	
5/16 and under	0.005	0.005	0.008
Over 5/16 to 7/16, incl	0.006	0.006	0.009
Over 7/16 to 5/8, incl	0.007	0.007	0.010
Over 5/8 to 7/8, incl	0.008	0.008	0.012
Over 7/8 to 1, incl	0.009	0.009	0.013
Over 1 to 1 1/8, incl	0.010	0.010	0.015
Over 1 1/8 to 1 1/4, incl	0.011	0.011	0.016
Over 1 1/4 to 1 3/8, incl	0.012	0.012	0.018
Over 1 3/8 to 1 1/2, incl	0.014	0.014	0.021
Over 1 1/2 to 2, incl	1/64	1/64	0.023
Over 2 to 2 1/2, incl	1/32	0	0.023
Over 2 1/2 to 3 1/2, incl	3/64	0	0.035
Over 3 1/2 to 4, incl	1/16	0	0.046

[1] For complete specification, reference may be made to ASTM A96 (see note,

Threads.—Unless otherwise specified, threads shall be in accordance with the American Standard Screw Threads for High-strength Bolting, ASA B1.4, see abstract on page 548. Where practicable, threads shall be formed after heat treatment.

Finish.—Bolts, screws, studs, and stud bolts shall be pointed and shall have a workmanlike finish.

Standard permissible variations for dimensions of bars shall be as set forth in the table on page 537.

Headed bolts shall be semifinished, hexagonal or square in shape, and in accordance with the dimensions for the regular or heavy series, as required, of the American Standard for Wrench-head Bolts and Nuts and Wrench Openings (ASA No. B18.2). Unless otherwise specified, the American Standard heavy hexagon series shall be used.

Composition.—The composition of the bolting material, other than phosphorus and sulphur, shall be agreed upon by the manufacturer and the purchaser. Nickel, chrome-nickel, chrome-vanadium, chrome-manganese, or any other type of alloy steel may be submitted under these specifications. It is recommended that the carbon content shall not be less than 0.20 or more than 0.45 per cent, and that the carbon ranges shall be 0.10 per cent of carbon.

Phosphorus, per cent, not over	0.045
Sulphur, per cent, not over	0.05

Physical Properties.—The bolting material shall conform to the following tensile properties for the class specified:

Diameter, inches		Class A	Class B	Class C
2½ and under	Tensile strength, min, psi	95,000	105,000	125,000
	Yield point, min, psi	70,000	80,000	105,000
	Elongation in 2 in., min, per cent	20	20	16
	Reduction of area, min, per cent	50	50	50
Over 2½ to 4, incl.	Tensile strength, min, psi	90,000	100,000	115,000
	Yield point, min, psi	65,000	75,000	95,000
	Elongation in 2 in., min, per cent	20	20	16
	Reduction of area, min, per cent	50	50	45
Over 4 to 7, incl.	Tensile strength, min, psi	90,000	100,000	110,000
	Yield point, min, psi	65,000	75,000	85,000
	Elongation in 2 in., min, per cent	20	20	16
	Reduction of area, min, per cent	50	50	45

Heat Treatment.—Heat treatment shall consist of quenching and tempering.

Tension Tests.—For bars, one tension test shall be made from each tempering charge representing each heat-treatment charge and melt. For bolts, screws, studs, and bolt-studs, one tension test shall be made on each lot of 300 pieces or fraction thereof for sizes 2½ in. and under in diameter; or for each lot of 100 pieces or fraction thereof for sizes over 2½ in. Unless required on the order, tension tests shall not be made on an order of less than 300 or 100 pieces of 2½ in. and under, or over 2½ in., respectively. In which case, acceptance shall be based on Brinell hardness tests or certification that material conforms

to these specifications. Where the section will permit, a standard 2-in.-gage-length, ½-in.-diameter test specimen shall be used.

Brinell Hardness Tests.—When agreed upon between the manufacturer and the purchaser, Brinell hardness tests may be made to determine the acceptance of bolting material in lieu of tension tests but shall not be used as a basis for rejection without confirming tension tests being made.

The bolting material, after final heat treatment, shall conform to the following requirements:

Class	Brinell Hardness Number
A	190 to 250
B	210 to 270
C	260 to 320

ASTM Tentative Specifications for ALLOY-STEEL BOLTING MATERIALS FOR HIGH-TEMPERATURE SERVICE FROM 750 to 1100 F METAL TEMPERATURES

Serial Designation A193-44T

Abstracted[1]

These specifications cover several varieties of ferritic and one austenitic alloy steel that have been used rather extensively as bolting material in the temperature range 750 to 1100 F. The term "bolting material" as used in these specifications covers rolled, forged, or cold-drawn bars and bolts, screws, studs, and stud-bolts. (Bolting material for more moderate high-temperature service is covered by ASTM Specification A96, abstracted on page 537). Selection from the respective steels, or from additional compositions which may be proposed, should be made on the basis of the requirements of design, service conditions, physical properties, and high-temperature charactéristics. The particular draw temperature and/or physical properties desired shall be specified on the purchase order.

Certification.—Where mutually agreed upon in writing between the manufacturer and the purchaser, a certification that the material conforms to the requirements of these specifications shall be the basis of acceptance of the material. Otherwise, the manufacturer shall report to the purchaser or his representative the results of the chemical analyses and physical tests made in accordance with these specifications.

Nuts.—Carbon- and alloy-steel nuts for use with these bolting materials are covered in ASTM Specification 194, abstracted on page 544.

Threads.—Unless otherwise specified, threads shall be in accordance with the American Standard Screw Threads for High-strength Bolting, ASA B1.4, see abstract on page 548. Where practicable, threads shall be formed after heat treatment.

Finish.—See similar requirements for ASTM A96 abstracted on page 537.

[1] For complete specification, reference may be made to ASTM A193 (see note, p. 369).

Physical Properties of ASTM Specification A193 Bolting Material.—Materials shall conform to the following requirements as to minimum tensile properties at room temperature after oil quenching and drawing as noted in the table. Physical properties of materials as stocked are given in boldface type.

Grade	Diameter, inches	Minimum-draw temperature, Fahrenheit	Tensile strength, pounds per square inch	Yield point, pounds per square inch	Elongation in 2 in., per cent	
Ferritic steels						
B4 Nickel-chromium-molybdenum	2½ and under	1000	160,000	135,000	14	45
		1100	140,000	120,000	15	50
		1200	**125,000**	**105,000**	**16**	**50**
	Over 2½ to 4, incl.	1000	155,000	130,000	14	45
		1100	135,000	115,000	15	50
		1200	120,000	100,000	16	50
B5 4 to 6 per cent chromium	2½ and under	1000	120,000	95,000	15	45
		1100	110,000	90,000	16	50
		1200	100,000	85,000	17	55
	Over 2½ to 4, incl.	1000	105,000	85,000	15	45
		1100	100,000	80,000	16	50
		1200	95,000	75,000	17	55
B6 13 per cent chromium	2½ and under	1000	150,000	125,000	13	45
		1100	120,000	100,000	15	50
		1200	105,000	85,000	17	55
	Over 2½ to 4, incl.	1000	140,000	120,000	13	45
		1100	115,000	95,000	15	50
		1200	100,000	80,000	17	55
B7 Chromium-molybdenum	2½ and under	1000	135,000	115,000	15	50
		1100	**125,000**	**105,000**	**16**	**50**
		1200	105,000	90,000	17	55
	Over 2½ to 4, incl.	1000	125,000	105,000	15	50
		1100	115,000	95,000	16	55
		1200	105,000	85,000	17	55
B7a Chromium-high-molybdenum	2½ and under	1000	135,000	115,000	15	50
		1100	**125,000**	**105,000**	**16**	**50**
		1200	105,000	90,000	17	55
	Over 2½ to 4, incl.	1000	125,000	105,000	15	50
		1100	115,000	95,000	16	55
		1200	105,000	85,000	17	55
B11 Tungsten-chromium-vanadium	2½ and under	1000	210,000	190,000	11	35
		1100	205,000	180,000	12	35
		1200	185,000	165,000	13	40
	Over 2½ to 4, incl.	1000	205,000	185,000	10	35
		1100	200,000	175,000	11	35
		1200	180,000	160,000	12	40
B12 Nickel-chromium	2½ and under	**1000**	**125,000**	**105,000**	**16**	**50**
		1100	115,000	95,000	17	50
		1200	105,000	85,000	18	55
	Over 2½ to 4, incl.	1000	120,000	100,000	16	50
		1100	110,000	90,000	17	50
		1200	100,000	80,000	18	55

Physical Properties of ASTM Specification A193 Bolting Material.—*(Concluded)*

Grade	Diameter, inches	Minimum-draw temperature, Fahrenheit	Tensile strength, pounds per square inch	Yield point, pounds per square inch	Elongation in 2 in., per cent	Reduction of area, per cent
Ferritic steels *(Continued)*						
B13 Tungsten-molybdenum-chromium	2½ and under	1000 1100 1200	155,000 135,000 120,000	130,000 115,000 100,000	14 15 16	45 50 55
	Over 2½ to 4, incl.	1000 1100 1200	145,000 125,000 115,000	120,000 105,000 95,000	14 15 16	45 50 55
B14[1] Chromium-molybdenum-vanadium	2½ and under	1000 1100 1200	145,000 135,000 125,000	120,000 115,000 105,000	14 15 16	45 45 50
	Over 2½ to 4, incl.	1000 1100 1200				
B15 Silicon-chromium-molybdenum	2½ and under	1000 1100 1200	160,000 140,000 125,000	135,000 120,000 105,000	14 15 16	45 50 50
	Over 2½ to 4, incl.	1000 1100 1200	155,000 135,000 120,000	130,000 115,000 100,000	14 15 16	45 45 50
Austenitic steels						
B8 18 chromium 8 nickel	All diameters	2000 (water quench)	75,000	30,000	35	50

[1] This material has been given a special normalizing treatment.

Tension Tests.—For bars, one tension test shall be made from each tempering charge. If more than one quenching charge is represented in a tempering charge, one tension test shall be made from each quenching charge. If more than one melt is represented in a quenching charge, one tension test shall be made for each melt.

Except as specified in the following paragraph, one tension test shall be made for each lot of 300 pieces or fraction thereof of bolts, screws, and studs for sizes up to and including 2½ in. diameter; or for each lot of 100 pieces or fraction thereof for sizes over 2½ in.

Unless required on the order the tension tests of the preceding paragraph shall not be made on an order of less quantity than specified; in which cases acceptance shall be based upon certification of the manufacturer.

Marking.—The serial marking for identification of material and the manufacturer's identification mark shall be stamped on the top of the head of bolts

Chemical Composition of ASTM Specification A193 Bolting Material.—Each alloy* shall conform to the following grades of chemical composition, or to other chemical composition as specified on the purchase order:

Type	Ferritic steels										Austenitic steel
Identification symbol	B4	B5	B6	B7	B7a	B11	B12	B13	B14	B15	B8
Grade	Nickel-chromium-molybdenum	4 to 6 per cent chromium	13 per cent chromium	Chromium-molybdenum	Chromium-high molybdenum	Tungsten-chromium-vanadium	Nickel-chromium	Tungsten-molybdenum-chromium	Chromium molybdenum-vanadium	Silicon chromium-molybdenum	18 chromium-8 nickel
Carbon, per cent	0.35 to 0.45	0.35 max.	0.12 max.	0.35 to 0.45	0.35 to 0.45	0.40 to 0.50	0.35 to 0.45	0.30 to 0.40	0.35 to 0.50	0.40 to 0.50	0.07 max.
Manganese, per cent	0.50 to 0.80	0.30 to 0.50	0.60 max.	0.60 to 0.90	0.60 to 0.90	0.20 to 0.40	0.60 to 0.90	0.60 to 0.90	0.40 to 0.70	0.40 to 0.70	0.20 to 0.70
Phosphorus, max, per cent	0.04	0.03	0.140[2]	0.04	0.04	0.04	0.04	0.04	0.04	0.03	0.140[2]
Sulphur, max, per cent	0.05	0.03	0.500[2]	0.05	0.05	0.04	0.05	0.04	0.05	0.03	0.500[2]
Silicon, per cent	....	0.50 max	0.50 max	0.15 to 0.30	0.15 to 0.30	0.15 to 0.30	....	0.15 to 0.30	0.15 to 0.30	0.50 to 0.80	0.75 max.
Nickel, per cent	1.50 to 2.00	....	....	....	....	....	1.00 to 1.50	....	....	....	7.00 to 10.0
Chromium, per cent	0.50 to 0.80	4.00 to 6.00	11.5 to 13.0	0.80 to 1.10	0.80 to 1.10	1.00 to 1.50	0.45 to 0.75	0.45 to 0.75	0.80 to 1.10	1.00 to 1.50	17.0 to 20.0
Molybdenum, per cent	0.30 to 0.40	0.40 to 0.60[1]	....	0.15 to 0.25	0.45 to 0.65	....	....	0.40 to 0.65	0.30 to 0.40	0.40 to 0.60	
Tungsten, per cent	....	0.57 to 1.25[1]	....	....	....	1.70 to 2.30	....	0.85 to 1.35			
Vanadium, per cent	....	....	....	....	....	0.29 to 0.30	....	....	0.20 to 0.30		

[1] Either molybdenum or tungsten shall be used as specified.

[2] Where phosphorus exceeds 0.045 per cent, sulphur shall not exceed 0.050 per cent. Where sulphur exceeds 0.050 per cent, phosphorus shall not exceed 0.045 per cent. These limits of phosphorus and sulphur shall apply unless otherwise specified.

* As these steels are not necessarily all suitable for the full range of temperature up to 1100 F., care should be exercised in making the selection.

and screws, and on one end of studs. The identification symbol shall be as shown in the table of chemical compositions.

ASTM Tentative Specifications for HEAT-TREATED CARBON-STEEL BOLTING MATERIAL

Serial Designation A261-44T

Abstracted[1]

These specifications cover one grade of heat-treated carbon-steel bolting material 2 in. and under in diameter for pressure vessels, valves, flanges, and fittings. Material is identified by the symbol BO. The term "bolting material" covers bars, headed bolts, screws, studs, and stud-bolts.

Certification.—When agreed upon in writing, a certification that the material conforms to these specifications shall be the basis of acceptance.

Nuts.—Carbon-steel nuts for use with this bolting material are covered in ASTM Specification A194, abstracted on page 544.

Heat Treatment.—The heat treatment shall consist of quenching in a liquid medium from a temperature above the critical range and tempering at a temperature of at least 950 F.

Chemical Composition.—The material shall conform to the following chemical requirements:

	Ladle analysis	Check analysis
Carbon, max, per cent.....	0.55	0.57
Manganese, max, per cent.	1.00	1.06
Phosphorus, max, per cent.	0.04	0.048
Sulphur, max, per cent....	0.05	0.058
Silicon, min, per cent......	0.15	0.13

Tensile Properties.—The material after heat treatment shall conform to the following requirements:

	Grade BO
Tensile strength, min, psi..............................	100,000
Yield point, min, psi..................................	75,000
Elongation in 2 in., min, per cent.....................	16
Reduction of area, min, per cent......................	45

Hardness Test.—The bolting material after heat treatment shall have a Brinell hardness of 200 to 260. When agreed upon by the manufacturer and the purchaser, Brinell hardness tests may be made to determine the acceptance of material in lieu of tension tests, but they shall not be used as a basis of rejection without confirming tension tests. At least four times as many pieces shall be tested for hardness as required below for tension tests.

[1] For complete specification, reference may be made to ASTM A261 (see note,

Tension Tests.—For bars, one tension test shall be made from each tempering charge representing each quenching charge and melt. For bolts, screws, studs, and stud-bolts of sizes 1 in. and under in diameter, one tension test shall be made on each lot of 2,000 pieces, or fraction thereof, if required on the order. For sizes over 1 in. the tension test shall be made for each lot of 1,000 pieces, or fraction thereof, if required on the order.

Finish.—See similar requirements for ASTM A96, abstracted on page 537.

Threads.—Threads shall be in accordance with the American Standard for Screw Threads for High-strength Bolting, ASA B1.4 (see abstract on page 548). Where practicable, threads shall be formed after heat treatment.

Marking.—The material identification marking BO and the manufacturer's identification mark shall be stamped on the top of head of bolts and screws and on one end of studs and stud-bolts.

ASTM Standard Specifications for CARBON- AND ALLOY-STEEL NUTS FOR BOLTS FOR HIGH-PRESSURE AND HIGH-TEMPERATURE SERVICE TO 1100 F

Serial Designation A194-40

Abstracted[1]

These specifications cover carbon- and alloy-steel nuts for bolts used in either or both high-pressure and high-temperature service at temperatures up to 1100 F. Class 0 is intended for use under the least exacting conditions, Classes 3 and 4 for the most severe conditions, and Classes 1, 2, and *2H* for service between these extremes.

NOTE.—In the event that forged nuts in sizes ½ in. and under are unobtainable, bar nuts will be acceptable.[2]

Certification.—Where mutually agreed, certification that the material conforms to these requirements may be the basis of acceptance of the material in lieu of the physical tests specified. When specified, the drawing temperature used with Classes 2H, 3, and 4 shall be furnished to the purchaser.

Fabrication.—Class 0 nuts shall be made by the hot-forged process, cold process, or machined from bar stock. Classes 1 and 2 nuts shall be made by the hot-forged process, cold process, or machined from hot-forged or hot-rolled bars. Classes 2H, 3, and 4 nuts shall be made by the hot-forged or cold processes, or machined from hot-forged or hot-rolled bars and shall be heat-treated to meet the required physical properties. These classes of nuts shall be quenched and drawn at a temperature at least 100 F above the service temperature, but in no case less than 850 F for Classes 2H and 3. The minimum drawing temperature for Class 4 shall be 1100 F.

Stress-relieving.—Before tapping, all nuts, except Class 0, made by the cold process shall be heated in the process of manufacturing to a temperature of at least 1000 F. Nuts made by the hot process or from hot-forged or hot-rolled bars need not be subjected to this stress-relieving.

[1] For complete specification, reference may be made to ASTM A194 (see note, p. 369).

[2] Emergency Alternate Provision, issued Apr. 6, 1942.

Chemical Composition.—Class 0, 1, 2, 2H, 3, and 4 analyses will be furnished the purchaser only when specified on the order, and shall conform to the following requirements as to chemical composition for the respective classes:

	Class 0	Class 1	Classes 2 and 2H	Class 3[1]	Class 4[3]
Carbon, per cent	0.25 max	0.15 min	0.40 min	0.35 max	0.40 to 0.50
Chromium, per cent				4.0 to 6.0	
Molybdenum, per cent				0.40 to 0.60[2]	
Tungsten, per cent				0.75 to 1.25[2]	0.20 min
Phosphorus, max, per cent	0.10	0.05	0.05	0.05	0.04 max
Sulphur, max, per cent	0.15	0.05	0.05	0.05	0.05 max
Silicon, per cent				0.50 max	0.15 min

[1] Other types of alloy steel, with their appropriate heat treatments, approved by the purchaser may be submitted under these specifications. Silicon content of Class 3 nuts, 0.50 max.
[2] Either molybdenum or tungsten may be used, if desired.

Brinell Hardness Test.—Class 0 nuts in the condition as manufactured shall show a minimum Brinell hardness of 120. Samples of each class of nuts, except Class 0, shall show the Brinell hardness specified below: (1) in the finished condition; (2) after the sample has been subjected for 24 hr to a temperature of 850 F for Class 1, 1000 F for Classes 2 and 2H, and 1100 F for Classes 3 and 4, and then cooled slowly.

HARDNESS REQUIREMENTS[1]

Class	Sample nut as finished			Sample nut after treatment as in above paragraph	
	Brinell hardness	Rockwell hardness		Brinell hardness	Rockwell hardness, B scale
		C scale	B scale		
1	120 min	....	70 min	120 min	70 min
2	160 min	....	84 min	160 min	84 min
2H	248 to 352	24 to 37	......	180 min	89 min
3	248 to 352	24 to 37	......	200 min	94 min
4	248 to 352	24 to 37	......	200 min	94 min

[1] Rockwell hardness values were added to this standard in 1942 as an editorial change.

The Brinell hardness test shall be made on the side of the nut, not on the top or bottom space.

Drift Test.—Class 2H, 3, and 4 nuts when machined from bar stock and all other classes of nuts shall be capable of meeting the following drift test: A conical mandrel, part of which has a diameter equal to the nominal nut size, shall be forced through the tapped hole to the nominal nut size, cold, without cracking the body of the nut. The test may be continued until the nut is broken for examination of the structure.

Stripping Test.—Nuts of Classes 1, 2, 2H, 3, and 4 shall be capable of meeting the following stripping test: A nut shall be assembled on a piece of bolting

material held in a tension-testing machine so that a load is applied to the nut. The threads in the nut shall not strip when subjected to a stress equal to 130,000 psi for Class 1, and 150,000 psi for Classes 2, 2H, 3, and 4 figured from the mean diameter of the bolt. (Mean diameter is the average of the root and pitch diameters.)

Number of Tests.—When so specified on the purchase order, the purchaser's representative may select two nuts per keg for sizes ⅝ in. and smaller, one nut per keg for sizes over ⅝ in. up to and including 1½ in., and one nut per every two kegs for sizes larger than 1½ in., and they shall be subjected to the tests specified. Periodic control tests when made by the manufacturer on each lot of nuts may be considered sufficient for certification purposes.

Finish.[1]—Nuts shall be semifinished, hexagonal in shape, and in accordance with the dimensions for the Regular or Heavy Series, as required, of the American Standard for Wrench-Head Bolts and Nuts and Wrench Openings (ASA B18.2-1941). Unless otherwise specified, the American Standard Heavy Series shall be used.

Threads.—Nuts for use with heat-treated carbon-steel and alloy-steel bolts or bolt-studs shall be threaded, unless otherwise specified, in accordance with the American Standard for Screw Threads for High-strength Bolting (ASA B1.4), sizes 1 in. and smaller in diameter with the coarse-thread series and 1⅛ in. and larger in diameter with the 8-pitch-thread series (see page 548).

Marking.—Nuts shall bear the manufacturer's identification mark and shall be legibly stamped to indicate the class and process of manufacture as follows:

Class	Marking	
	Nuts hot forged or cold punched	Nuts machined from bar stock
0	No marking	0
1	1	1*B*
2	2	2*B*
2*H*	2*H*[1]	2*HB*[1]
3	3	3*B*
4	4	4*B*

[1] The letter *H* indicates heat-treated nuts.

American (National) Standard SCREW THREADS

ASA B1.1-1935

Abstracted[2]

This standard covers the dimensional specifications for American Standard Screw Threads applicable to bolts, machine screws, nuts, and other threaded

[1] Editorially revised in 1943.

[2] For complete standard, reference may be made to ASA B1.1 (see note, page 430).

Table L.—Coarse-thread Series and 8-pitch-thread Series*—General Dimensions, see Fig. 28
(Table 3, ASA B1.1-1935)
(All dimensions in inches)

Identification		Basic diameters			Thread data				Basic[2] area of section at root of thread	
Size	Threads per inch	Major diameter	Pitch diameter	Minor diameter	Pitch[1]	Basic depth of thread	Basic width	Minimum width of flat at major diameter of nut	Coarse-thread series, square inches	8-pitch-thread series, square inches
	n	D	E	K	p	h	$p/8$	$p/24$		
¼	20	0.2500	0.2175	0.1850	0.0500000	0.03248	0.00625	0.00208	0.0269	
5/16	18	0.3125	0.2764	0.2403	0.0555556	0.03608	0.00694	0.00231	0.0454	
⅜	16	0.3750	0.3344	0.2938	0.0625000	0.04059	0.00781	0.00260	0.0678	
7/16	14	0.4375	0.3911	0.3447	0.0714286	0.04639	0.00893	0.00298	0.0933	
½	13	0.5000	0.4500	0.4001	0.0769231	0.04996	0.00962	0.00321	0.1257	
9/16	12	0.5625	0.5084	0.4542	0.0833333	0.05413	0.01042	0.00347	0.1620	
⅝	11	0.6250	0.5660	0.5069	0.0909091	0.05905	0.01136	0.00379	0.2018	
¾	10	0.7500	0.6850	0.6201	0.1000000	0.06495	0.01250	0.00417	0.3020	
⅞	9	0.8750	0.8028	0.7307	0	0.07217	0.01389	0.00463	0.4193	
1	8	.0000	0.9188	0.8376	0.1250000	0.08119	0.01562	0.00521	0.5510	0.551
1⅛	7	.1250	.0322	0.9394	0.1428571	0.09279	0.01786	0.00595	0.6931	0.729
1¼	7	.2500	.1572	1.0644	0.1428571	0.09279	0.01786	0.00595	0.8898	0.930
1⅜	6	.3750	.2667	1.1585	0.1666667	0.10825	0.02083	0.00694	1.0541	1.155
1½	6	.5000	.3917	1.2835	0.1666667	0.10825	0.02083	0.00694	1.2938	1.405
1¾	5	.7500	.6201	1.4902	0.2000000	0.12990	0.02500	0.00833	1.7441	1.980
2	4½	2.0000	1.8557	1.7113	0.2222222	0.14434	0.02778	0.00926	2.3001	2.655
2¼	4½	2.2500	2.1057	1.9613	0.2222222	0.14434	0.02778	0.00926	3.0212	3.430
2½	4	2.5000	2.3376	2.1752	0.2500000	0.16238	0.03125	0.01042	3.7161	4.290
2¾	4	2.7500	2.5876	2.4252	0.2500000	0.16238	0.03125	0.01042	4.6194	5.250
3	4	3.0000	2.8376	2.6752	0.2500000	0.16238	0.03125	0.01042	5.6209	6.330
3¼	4	3.2500	3.0876	2.9252	0.2500000	0.16238	0.03125	0.01042	6.7205	
3½	4	3.5000	3.3376	3.1752	0.2500000	0.16238	0.03125	0.01042	7.9183	
3¾	4	3.7500	3.5876	3.4252	0.2500000	0.16238	0.03125	0.01042	9.2143	
4	4	4.0000	3.8376	3.6752	0.2500000	0.16238	0.03125	0.01042	10.6084	

* The 8-pitch thread series refers to bolt diameters 1 in. and greater when 8 threads per inch are used. Basic thread dimensions throughout the 8-pitch-thread series are the same as for the 1 in. diameter of the coarse-thread series. The coarse-thread series is recommended for general use; the 8-pitch-thread series for high-strength bolting.

[1] This column is given to seven decimal places for computation purposes only.

[2] These basic areas are given as information to be used in roughly computing the ultimate strength of the bolt. The actual strength, computed from the ultimate strength of the material, will depend on the actual cross-sectional area at the root of the thread and an increment contributed by the thread itself, depending on the material and process of manufacture.

parts. The form of thread profile now designated the American National formerly was known as "United States Standard" or "Sellers' Profile." The *basic angle* of thread between the sides of the thread measured in an axial plane is 60 deg. The line bisecting this 60-deg angle is perpendicular to the axis of the screw thread. The *basic width* of flat at the root and crest is given by $F = 0.125p$. The *basic depth* is found from: $h = 0.\ 0.649519 \times p = 0.649519/n$, where p = pitch in inches and n = number of threads per inch.

In order to provide clearance in nut at the minor diameter, the basic depth is reduced by one-sixth or more depending upon size and pitch by truncating the crest of the thread. Clearance at the major diameter of the nut is provided by decreasing the depth of the truncated triangle any desired amount down to one-third of its theoretical value.

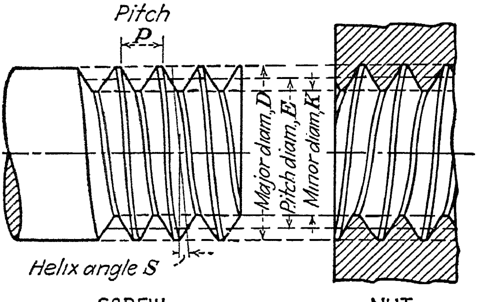


FIG. 28.—Screw threads for bolts and nuts, ASA B1.1.

Five series of screw threads are covered by this standard differing from one another in the diameter-pitch relations of the threads but all employing the American National form of thread. General dimensions of the coarse thread series, former "United States Standard," are reproduced in Table L. The basic areas at root of thread are given for both the coarse-thread series and the 8-pitch-thread series customarily used for bolt studs in high-pressure pipe flanges. For threads for high-temperature bolting, see abstract of ASA B1.4. For discussion of bolting practice, see pages 534 to 537; for bolting materials, see pages 537 to 546.

American Standard
SCREW THREADS FOR HIGH-STRENGTH BOLTING

ASA B1.4-1945

Abstracted[1]

This standard covers the limiting dimensions and tolerances for screw threads on high-strength bolting for use with pressure vessels and steel flanges,

[1] For complete specification reference may be made to ASA B1.4 (see note, p. 430).

fittings, and valves. This standard is based on a combination of screws made to Class 3 tolerances and nuts made to Class 2 tolerances according to the American Standard for Screw Threads, ASA B1.1.

For sizes 1 in. and smaller the coarse-thread series of that standard is used, while for larger sizes the 8-pitch-thread series is used.

Table LI.—Limiting Dimensions and Tolerances for Threads on Screws
(Table 1, ASA B1.4-1945)
(All dimensions in inches)

Sizes	Threads per inch	Basic major diameter	Allowance (minus)[1]	Major diameter Max	Major diameter Min	Major diameter Toler-	Pitch diameter Max	Pitch diameter Min	Pitch diameter Tolerance[2]	Minor diameter
1/4	20	0.2500	0.0010	0.2490	0.2418	0.0072	0.2165	0.2139	0.0026	0.1877
5/16	18	0.3125	0.0011	0.3114	0.3032	0.0082	0.2753	0.2723	0.0030	0.2432
3/8	16	0.3750	0.0013	0.3737	0.3647	0.0090	0.3331	0.3299	0.0032	0.2970
7/16	14	0.4375	0.0013	0.4362	0.4264	0.0098	0.3898	0.3862	0.0036	0.3486
1/2	13	0.5000	0.0015	0.4985	0.4881	0.0104	0.4485	0.4448	0.0037	0.4041
9/16	12	0.5625	0.0016	0.5609	0.5497	0.0112	0.5068	0.5028	0.0040	0.4587
5/8	11	0.6250	0.0017	0.6233	0.6115	0.0118	0.5643	0.5601	0.0042	0.5118
3/4	10	0.7500	0.0019	0.7481	0.7353	0.0128	0.6831	0.6786	0.0045	0.6254
7/8	9	0.8750	0.0021	0.8729	0.8589	0.0140	0.8007	0.7958	0.0049	0.7366
1		1.0000	0.0022	0.9978	0.9826	0.0152	0.9166	0.9112	0.0054	0.8444
1 1/8		.1250	0.0024	.1226	1.1074	0.0152	1.0414	1.0359	0.0055	0.9692
1 1/4		.2500	0.0025	.2475	1.2323	0.0152	1.1663	1.1605	0.0058	.0941
1 3/8		.3750	0.0025	.3725	1.3573	0.0152	1.2913	1.2852	0.0061	.2191
1 1/2		.5000	0.0027	.4973	1.4821	0.0152	1.4161	1.4098	0.0063	.3439
1 5/8		.6250	0.0028	.6222	1.6070	0.0152	1.5410	1.5345	0.0065	.4688
1 3/4		.7500	0.0029	.7471	1.7319	0.0152	1.6659	1.6591	0.0068	.5937
1 7/8		.8750	0.0030	.8720	1.8568	0.0152	1.7905	1.7838	0.0070	.7186
2		2.0000	0.0031	.9969	1.9817	0.0152	1.9157	1.9084	0.0073	.8435
2 1/8		2.1250	0.0032	2.1218	2.1066	0.0152	2.0406	2.0331	0.0075	.9684
2 1/4		2.2500	0.0033	2.2467	2.2315	0.0152	2.1655	2.1578	0.0077	2.0933
2 1/2		2.5000	0.0035	2.4965	2.4813	0.0152	2.4153	2.4071	0.0082	2.3431
2 3/4		2.7500	0.0037	2.7463	2.7311	0.0152	2.6651			2.5929
3		3.0000	0.0038	2.9962	2.9810	0.0152	2.9150	2.9058	0.0092	2.8428
3 1/4		3.2500	0.0039	3.2461	3.2309	0.0152	3.1649	3.1556	0.0093	3.0927
3 1/2		3.5000	0.0040	3.4960	3.4808	0.0152	3.4148	3.4055	0.0093	3.3426

The American National Form of thread shall be used, see ASA B1.1.

[1] The maximum pitch diameters of screws are smaller than the minimum pitch diameters of nuts by these amounts.

[2] Pitch diameter tolerances include errors of lead and angle.

To facilitate assembly and reduce the possibility of seizure of the threads, an allowance or neutral zone between the minimum nut and the maximum screw is provided by making the maximum pitch diameter of the screw smaller than the minimum pitch diameter of the nut. This allowance is equal to the difference between Class 2 and 3 tolerances of American Standard B1.1.

In conformance with American Standard practice, the basic nut has been retained, the allowance being taken exclusively from the screw. All nut dimen-

sions given in this standard are in agreement with those given in American Standard B1.1 for Class 2 fit. The maximum major diameter of the screw is below basic by an amount equal to the allowance. The tolerance on the major diameter is the same as the Class 3 tolerance specified in American Standard B1.1. The maximum minor diameter of the screw is smaller than the Class 2 dimension specified in B1.1 by an amount equal to the allowance. See Table LI for limiting dimensions and tolerances for threads on screws.

American Standard
WRENCH-HEAD BOLTS AND NUTS AND WRENCH OPENINGS

ASA B18.2-1941

Abstracted[1]

This standard covers regular, heavy, and light bolts, heads, and nuts both hexagon and square for general use by all industry. Dimensions of semifinished heavy nuts and heavy bolt heads are reproduced in Table LII since the greater bearing surface of the heavy series bolt heads and nuts customarily is specified for piping work. For dimensions of regular series bolt heads and nuts and light series nuts, reference may be made to ASA B18.2.

NOTES AND FORMULAS

Heavy nuts and heavy bolt heads are for use where greater bearing surface is desired.

Semifinished nuts are threaded, and machined on bearing surface only.

Semifinished bolt heads are machined under head only.

Width across flats equals $(1\frac{1}{2}D + \frac{1}{8})$, adjusted to sixteenths.

Taper of the sides of nuts and bolt heads shall not exceed 2 deg, and the width across flats shall be measured at the largest width.

Tolerance for width across flats is minus $0.050D$ from basic.

Minimum width across rounded corners of square equals 1.373 times minimum width across flats.

Minimum width across rounded corners of hexagon equals 1.14 times minimum width across flats.

Nominal thickness of unfinished square nuts equals D.

Nominal thickness of heavy semifinished hexagonal nuts is the over-all distance from the top to the bearing surface and for sizes $\frac{1}{2}$ to $1\frac{1}{8}$ in. equals $(D - \frac{1}{64})$; for sizes $1\frac{1}{4}$ to 2 in. equals $(D - \frac{1}{32})$; for sizes $2\frac{1}{2}$ to 3 in. equals $(D - \frac{3}{64})$; and for sizes $3\frac{1}{4}$ to 4 in. equals $(D - \frac{1}{16})$.

Nominal height of head is the distance from the top to the bearing surface and for sizes $\frac{1}{2}$ to $\frac{7}{8}$ in. equals $(\frac{3}{4}D - \frac{1}{32})$; for sizes 1 to $1\frac{7}{8}$ in. equals $(\frac{3}{4}D)$; for sizes 2 to 3 in. equals $(\frac{3}{4}D - \frac{1}{16})$.

Tolerance for thickness of nut and for height of head is plus or minus $(0.016D + 0.012)$ from nominal.

Tops of square and hexagon nuts and bolt heads shall be flat and chamfered; the angle of chamfer with the top surface shall be 25 deg for square and 30 deg

[1] For complete standard, reference may be made to ASA B18.2 (see note, p. 430).

for hexagon; diameter of top circle shall be the maximum width across flats within a tolerance of minus 15 per cent.

Bearing surface of semifinished heads and nuts shall be washer faced; thickness of the washer face shall be 1/64 in.; diameter of the washer face shall be the maximum width across flats within a tolerance of −5 per cent.

Bearing surface shall be at right angles to the axis of the threaded hole or bolt within a tolerance of 2 deg for 1-in. nuts and bolts or smaller, and 1 deg for nuts and bolts larger than 1 in.

Bearing surface of heads of bolts larger than 1 in. is concentric with the axis of the body within a tolerance of 3 per cent of the maximum width across flats.

Maximum radius under head of bolts for 1/2-in. size shall be 1/32; for sizes 9/16 to 1 in. shall be 1/16; for sizes 1 1/8 to 2 in. shall be 1/8; for sizes 2 1/4 to 3 in. shall be 3/16.

Table LII.—American Standard Heavy Nuts and Heavy Bolt Heads

(Compiled from Tables 3, 4, 11, and 12, ASA B18.2-1941)
(All dimensions in inches)

Diameter of bolt D		Width across flats, Max (basic)		Width across flats, Min	Width across corners, min, Sq	Width across corners, min, Hex	Height heavy bolt heads, Unfinished, Sq	Height heavy bolt heads, Unfinished, Hex	Height heavy bolt heads, Semi-finished hex	Thickness heavy nuts, Unfinished, Sq	Thickness heavy nuts, Unfinished, Hex	Thickness heavy nuts, Semi-finished hex
1/2	0.5000	7/8	0.8750	0.850	1.167	0.969	7/16	7/16	13/32	1/2	1/2	31/64
9/16	0.5625	15/16	0.9375	0.906	1.244	1.033	15/32	15/32	7/16	9/16	9/16	35/64
5/8	0.6250	1 1/16	1.0625	1.031	1.416	1.175	17/32	17/32	1/2	5/8	5/8	39/64
3/4	0.7500	1 1/4	1.2500	1.213	1.665	1.383	5/8	5/8	19/32	3/4	3/4	47/64
7/8	0.8750	1 7/16	1.4375	1.394	1.914	1.589	23/32	23/32	11/16	7/8	7/8	55/64
1	1.0000	1 5/8	1.6250	1.575	2.162	1.796	13/16	13/16	3/4	1	1	63/64
1 1/8	1.1250	1 13/16	1.8125	1.756	2.411	2.002	29/32	29/32	27/32	1 1/8	1 1/8	1 7/64
1 1/4	1.2500	2	2.0000	1.938	2.661	2.209	1	1	15/16	1 1/4	1 1/4	1 7/32
1 3/8	1.3750	2 3/16	2.1875	2.119	2.909	2.416	1 3/32	1 3/32	1 1/32	1 3/8	1 3/8	1 11/32
1 1/2	1.5000	2 3/8	2.3750	2.300	3.158	2.622	1 3/16	1 3/16	1 1/8	1 1/2	1 1/2	1 15/32
1 5/8	1.6250	2 9/16	2.5625	2.481	3.406	2.828	1 9/32	1 9/32	1 7/32	1 5/8	1 5/8	1 19/32
1 3/4	1.7500	2 3/4	2.7500	2.663	3.656	3.036	1 3/8	1 3/8	1 5/16	1 3/4	1 3/4	1 23/32
1 7/8	1.8750	2 15/16	2.9375	2.844	3.905	3.242	[illegible]	[illegible]	1 13/32	1 7/8	1 7/8	1 27/32
2	2.0000	3 1/8	3.1250	3.025	4.153	3.449	[illegible]	[illegible]	1 7/16	2	2	1 31/32
2 1/4	2.2500	3 1/2	3.5000	3.388	4.652	3.862	[illegible]	[illegible]	1 5/8	2 1/4	2 1/4	2 13/64
2 1/2	2.5000	3 7/8	3.8750	3.750	5.149	4.275	1 15/16	1 15/16	1 13/16	2 1/2	2 1/2	2 29/64
2 3/4	2.7500	4 1/4	4.2500	4.113	5.647	4.689	2 1/8	2 1/8	2	2 3/4	2 3/4	2 45/64
3	3.0000	4 5/8	4.6250	4.475	6.144	5.102	2 5/16	2 5/16	2 3/16	3	3	2 61/64
3 1/4	3.2500	5	5.0000	4.838	6.643	5.515	.....	.....	.....	3 1/4	3 1/4	3 3/16
3 1/2	3.5000	5 5/8	5.3750	5.200	7.140	5.928	.....	.....	.....	3 1/2	3 1/2	3 7/16
3 3/4	3.7500	5 3/4	5.7500	5.563	7.638	6.342	.....	.....	.....	3 3/4	3 3/4	3 11/16
4	4.0000	6 1/8	6.1250	5.925	8.135	6.755	.....	.....	.....	4	4	3 15/16

Wrenches.—For piping work, especially on high-pressure joints, it is desirable to use box or closed-end wrenches rather than the

open type, since the latter are likely to become sprung open. Ample clearance must, therefore, be provided between the bolts and the base of the flange hub. Table LIII represents good practice in this respect.

TABLE LIII.—DIMENSIONS AND CLEARANCE FOR BOX WRENCHES FOR HEAVY SERIES HEXAGONAL NUTS OR BOLT HEADS

Size of bolt	Diameter of wrench head	Thickness of wrench head	Minimum radial distance from center of bolt to base of flange hub
1/4	29/32	1/4	1/2
5/16	1 3/32	9/32	5/8
3/8	1 1/4	5/16	11/16
7/16	1 3/8	11/32	3/4
1/2	1 1/2	25/64	13/16
9/16	1 5/8	7/16	7/8
5/8	1 3/4	31/64	15/16
3/4	2 1/16	9/16	1 1/8
7/8	2 3/8	21/32	1 1/4
1	2 5/8	3/4	1 3/8
1 1/8	2 7/8	13/16	1 1/2
1 1/4	3 1/4	15/16	1 3/4
1 3/8	3 1/2	1	1 7/8
1 1/2	3 3/4	1 1/16	2
1 5/8	4	1 1/8	2 1/8
1 3/4	4 1/4	1 1/4	2 1/4
2	4 3/4	1 3/8	2 1/2
2 1/4	5 1/4	1 1/2	2 3/4
2 1/2	5 7/8	1 5/8	3
2 3/4	6 1/2	1 3/4	3 1/2
3	7	1 7/8	4

VALVES

Gate Valves.—Figure 29 illustrates typical gate valves of the rising- and nonrising-steam types, for 250 lb working steam pressure. Gate valves up to this working pressure are usually made with an oval bonnet flange to reduce the face-to-face dimension. In the valves designed for higher pressures, the bonnet flange is made circular to permit a recessed gasket joint. Such a valve designed for 600 lb working steam pressure is illustrated in Fig. 30.

In the rising-stem type of valve the upper part of the stem is threaded, and a nut, fastened solidly to the handwheel and held in the yoke by thrust collars, serves to move the stem as the handwheel is turned. In the nonrising-stem valve the lower end of the stem is threaded and screws into the disk, being restrained by a thrust collar. The rising-stem valve obviously requires a greater amount of space when opened. It is generally to be

preferred, however, because the position of the stem indicates at once whether the valve is open or closed. Nonrising-stem valves are sometimes provided with an indicator for this purpose.

There are three types of seating employed in gate valves. The wedge gate shown in Fig. 30 is commonly used. This is a solid-wedge type. There are also split-wedge or *double-disk* gate valves in which the disks are forced against the seats by the wedging

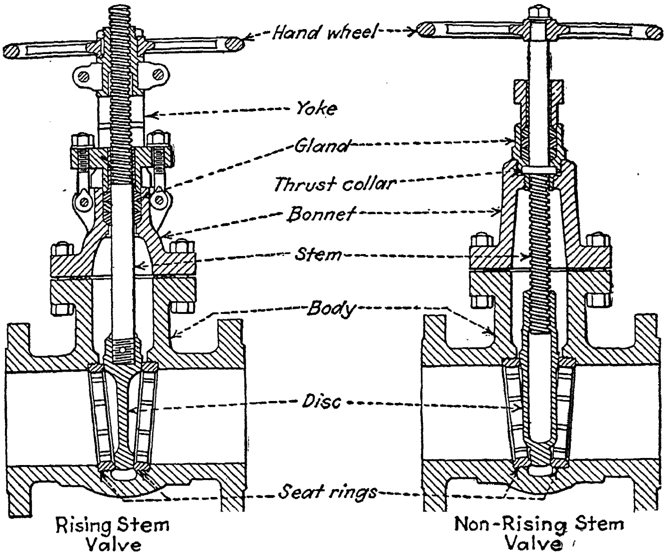


FIG. 29.—Wedge gate valves for 250 lb. working steam pressure.

action of the stem as it is screwed home (see Fig. 3, page 1042 and Fig. 3*b*, page 1122).

Some engineers favor the *parallel-slide* valve (Fig. 31), which depends for its tightness entirely upon the fluid pressure exerted against one side or the other of the disk. The chief advantage of this type is that the disk cannot be jammed into the body, thereby making the valve difficult to open. This is particularly important where motors are used for opening and closing. Actually, it is probable that the tightness of the solid-wedge gate valve is also really produced by the fluid pressure acting against one side or the other so that the two types are similar in this respect. The

disk in the parallel-slide type slides against the seat while the valve is being opened or closed. Consequently, these parts must be made of metals which have no tendency to gall or tear each other. Being held away by guides, the wedge gate does not come into contact with its seat until almost at the closing point. The parallel-slide gate valve is especially favored for high-temperature steam service by some engineers on the supposition that it is less

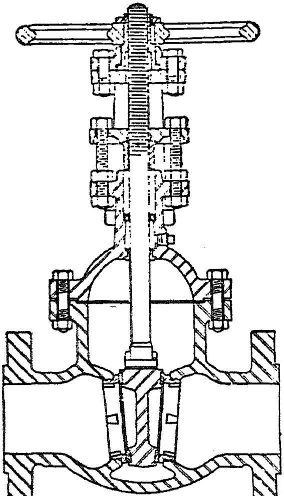


FIG. 30.—Wedge gate valve for 600 lb. working steam pressure.

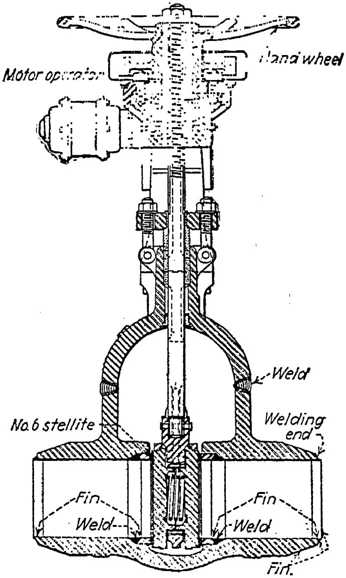


FIG. 31.—Parallel-slide gate valve showing welded construction.

apt to stick in the closed position as a result of change in temperature. Gate valves are used where a straight-through flow is desired with a minimum amount of pressure loss.

A parallel-slide gate valve for welded construction is shown in Fig. 31. This valve, with welding ends, a welded bonnet joint, and welded-in seat rings, is particularly suited for high-temperature service where trouble might be experienced in keeping flanged joints tight. The use of welded-in seat rings with stellite facing is deemed to ensure sufficient durability to warrant welding the bonnet joint as well as the line joints.

TABLE LIV.—TYPICAL DIMENSIONS OF 125-LB STANDARD GATE VALVES

Pipe size	2	2 1/2	3	3 1/2	4	4 1/2	5	6	7	8	10	12	14	16	18	20	24
Face to face, screw end valves, inches	5 1/4	5 3/4	6 5/16	6 5/8	7 1/16	7 3/8	8	8 3/4									
Face to face, flange end valves, inches	7	7 1/2	8	8 1/2	9	9 1/2	10	10 1/2	11	11 1/2	13	14	15	16	17	18	20
Diameter of flanges, inches	6	7	7 1/2	8 1/2	9	9 1/4	10	11	12 1/2	13 1/2	16	19	21	23 1/2	25	27 1/2	32
Rising stem valves:																	
Center to top of stem, open, inches	13 1/2	15 11/16	17 7/16	19 15/16	22 13/16	24 5/8	26 7/8	30 5/8	34 3/16	37 5/8	45 3/4	53 9/16	58 7/8	67	74 11/16	82 7/16	97 15/16
Center to top of stem, closed, inches	11 1/4	12 7/8	14 1/8	16 1/8	18 7/16	19 3/4	21 9/16	24 1/4	26 15/16	29 9/16	35 7/16	41 1/4	45 1/4	51 5/16	57 1/8	62 7/8	74 5/16
Non-rising stem valves:																	
Center to top of stem, inches	11 5/16	13	13 5/16	15 3/4	18	18 13/16	20 13/16	22 1/2	25 3/8	27 1/16	31 11/16	36 3/16					

TABLE LV.—AMERICAN STANDARD FOR CONTACT-SURFACE-TO-CONTACT-SURFACE DIMENSIONS OF CAST-IRON DOUBLE-DISK FLANGED GATE VALVES
(Table 2, ASA B16.10)
(All dimensions in inches)

Nominal pipe size	Contact surface to contact surface dimensions			
	125 lb	175 lb	250 lb	800 lb hydraulic
2	7	7¼	8½	11½
2½	7½	8	9½	13
3	8	9¼	11⅛	14
3½	8½			
4	9	10½	12	17
5	10			
6	10½	13	15⅞	22
8	11½	14¼	16½	26
10	13	16¾	18	31
12	14	17½	19¾	33
14 O.D.	....	....	22½	
16 O.D.	....	....	24	
	....	....		
18 O.D.	....	....	26	
20 O.D.	....	....	28	
24 O.D.	....	....	31	

Pressure designations are the basic steam-service ratings of the valves, except the 800 lb hydraulic.
Where dimensions are not given, either the sizes are not made, or there is insufficient demand to warrant the expense of unification.
The connecting-end flanges of 175-lb valves are the same as those on 250-lb valves.

Pressure Standards.—There are a large number of weights of valves for various pressures. Gate valves in sizes 2 in. and upward are commonly manufactured for working steam pressures of 25, 125, and 250 lb with bodies of cast iron and with steel bodies for the ASA working-steam-pressure ratings of 150, 300, 400, 600, 900, 1,500, and 2,500 lb, with higher pressure ratings for use with water (see page 626). Some manufacturers offer additional lines of cast-iron valves for intermediate pressures such as 150 or 175 lb working steam pressure.

Dimensions.—Typical dimensions of screwed- and flanged-end gate valves suitable for 125 psi steam pressure are given in Table LIV. The American Standard for Face-to-face Dimensions of Ferrous Flanged- and Welding-end Valves, ASA B16.10, is based on the dimensions of MSS Standard Practice, SP 32, formulated by the Manufacturers' Standardization Society of the Valve and

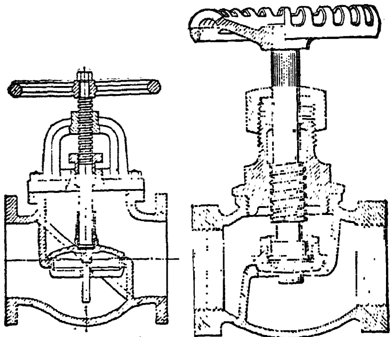
Fittings Industry. Standard contact-surface-to-contact-surface dimensions of cast-iron double-disk flanged gate valves are given in Table LV. Dimensions of cast-iron and steel flanged wedge gate valves are given in Table LVI. Welding-end valves 8 in. and smaller have the same face-to-face dimensions as flanged valves.

TABLE LVI.—AMERICAN STANDARD CONTACT-SURFACE-TO-CONTACT-SURFACE DIMENSIONS OF CAST-IRON AND STEEL FLANGED WEDGE GATE VALVES
(Table 1, ASA B16.10-1939)
(All dimensions in inches)

Nominal pipe size	Contact surface to contact surface dimensions									
	Cast iron				Steel					
	125 lb	175 lb	250 lb	800 lb hydraulic	150 lb	300 lb	400 lb	600 lb	900 lb	1,500 lb
1	....	....	....	....	....	....	8½	8½	10	10
1¼	....	....	....	....	....	....	9	9	11	11
1½	....	....	....	....	....	7½	9½	9½	12	12
2	7	7¼	8½	11½	7	8½	11½	11½	14½	14½
2½	7½	8	9½	13	7½	9½	13	13	16½	16½
3	8	9¼	11⅛	14	8	11⅛	14	14	15	18½
3½	8½	10	11⅞	....	8½	11⅞				
4	9	10½	12	17	9	12	16	17	18	21½
5	10	11½	15	....	10	15	18	20	22	26½
6	10½	13	15⅞	22	10½	15⅞	19½	22	24	27¾
8	11½	14¼	16½	26	11½	16½	23½	26	29	32¾
10	13	16¾	18	31	13	18	26½	31	33	39
12	14	17½	19¾	33	14	19¾	30	33	38	44½
14 O.D.	15	....	22½	....	15	30	32½	35	40½	
16 O.D.	16	....	24	....	16	33	35½	39	44½	
18 O.D.	17	....	26	....	17	36	38½	43	48	
20 C.D.	18	....	28	....	18	39	41½	47	52	
24 O.D.	20	....	31	....	20	45	48½	55	61	

Pressure designations are the basic steam-service ratings of the valves, except the 800 lb hydraulic.
Where the dimensions are not given, either the sizes are not made, or there is insufficient demand to warrant the expense of unification.
The connecting-end flanges of 175-lb valves are the same as those on 250-lb valves.

Globe Valves.—The globe valve has the advantage of being less costly to manufacture than the gate valve. Its disadvantages are that it interposes a greater resistance to the flow; it is not



Iron body globe valve rising stem.

All brass globe valve.

FIG. 32.—Globe valves.

TABLE LVII.—AMERICAN STANDARD CONTACT-SURFACE-TO-CONTACT-SURFACE DIMENSIONS OF CAST-IRON AND STEEL FLANGED GLOBE AND ANGLE VALVES
(Table 3, ASA B16.10-1939)
(All dimensions in inches)

Nominal pipe size	Contact surface to contact surface dimensions							
	Cast iron		Steel					
	125 lb	250 lb	150 lb	300 lb	400 lb	600 lb	900 lb	1,500 lb
¾	....	....	....	....	7½	7½		
1	....	....	....	....	8½	8½	10	10
1¼	....	....	....	....	9	9	11	11
1½	....	....	....	....	9½	9½	12	12
2	8	10½	8	10½	11½	11½	14½	14½
2½	8½	11½	8½	11½	13	13	16½	16½
3	9½	12½	9½	12½	14	14	15	18½
3½	10½	13¼	10½	13¼				
4	11½	14	11½	14	16	17	18	21½
5	13	15¾	14	15¾	18	20	22	26½
6	14	17½	16	17½	19½	22	24	27¾
8	19½	21	19½	22	23½	26	29	32¾

.o warrant expense of unification.
Center to contact surface dimensions of angle valves are one-half of the contact surface to contact surface dimensions of corresponding globe valves. Tolerances on center to contact surface of angle valves are one-half those for contact surface to contact surface dimensions.

balanced, making it sometimes necessary to overcome the full effect of the fluid pressure against the disk area when closing or opening (depending upon the direction of flow); and it may prevent the complete drainage of the pipe line because of its peculiar construction. A globe valve is sometimes of advantage where the flow is to be throttled and for this purpose is superior to the gate valve. Globe valves are not ordinarily used for working steam pressures of over 250 lb except in the smaller sizes or for special throttling purposes. Figure 32 shows a large valve with flanged ends and a small screwed-end valve. Globe valves invariably have rising stems, and the larger sizes are of the outside-screw-and-yoke construction.

Contact-surface-to-contact-surface dimensions of cast-iron and steel flanged globe and angle valves are given in Table LVII.

Check Valves.—There are two principal kinds of check valves shown in Fig. 33, the swing check and the lift check. The

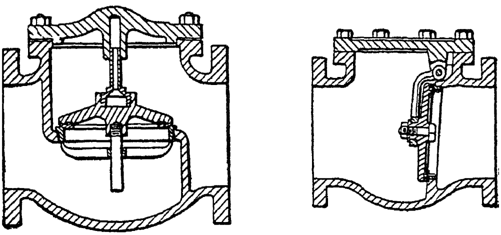


FIG. 33.—Check valves.

swing check is the more commonly used. Check valves are installed to prevent reversal of the flow of the fluid. Lift or poppet checks are frequently desirable in vertical pipe lines. The force of gravity obviously plays some part in the functioning of a check valve, and the position of the valve must always be given consideration. Lift checks must always be placed so that the direction of lift is exactly vertical; and swing checks must be placed so that the flapper will always be closed freely and positively by gravity. For balanced swing checks, see pages 903 and 1035.

Contact-surface-to-contact-surface dimensions of cast-iron and steel flanged swing-check valves are given in Table LVIII.

Valve Materials.—Valve bodies are made of brass or bronze in the smaller sizes and for moderate pressures and temperatures. Cast iron is used in large sizes up to working steam pressures of

250 lb and temperatures of 450 F. Cast steel and sometimes alloy cast steel are used for more severe service. Forged steel is used in small valve bodies and the parts drilled out, but this is hardly a practicable method in the larger sizes, although valves up to 8 in. have been made.

Table LVIII.—American Standard Contact-surface-to-contact-surface Dimensions of Cast-iron and Steel Flanged Swing-check Valves
(Table 4, ASA B16.19-1939)
(All dimensions in inches)

Nominal pipe size	Contact surface to contact surface dimensions						
	Cast iron			Steel			
	125 lb	250 lb	800 lb hydraulic	150 lb	300 lb	400 lb	600 lb
2	8	10½	11½	8	10½	11½	11½
2½	8½	11½	13	8½	11½	13	13
3	9½	12½	14	9½	12½	14	14
3½	10½	13¼	....	10½	13¼		
4	11½	14	17	11½	14	16	17
5	13	15¾	....	13	15¾		
6	14	17½	22	14	17½	19½	22
8	....	21	26	....	21	23½	26
10	....	24½	31	....	24½	26½	31
12	....	28	33	....	28	30	33

These dimensions are not intended to cover the type of check valve having the seat angle at approximately 45 deg to the run of the valve or the "Underwriters' Pattern" where large clearances are required.

Pressure designations are the primary steam-service ratings of the connecting end flanges, except the 800 lb hydraulic.

Where dimensions are not given, either the sizes are not made, or there is insufficient demand to warrant expense of unification.

Some small valves are made of monel metal or stainless steel throughout to suit pressure and temperature conditions obtaining. A considerable number of small valves for high-pressure instrument piping and similar services are manufactured out of solid-bar stock which is turned and bored to suit.

The materials used for valve seats and disks for high-pressure steam service are very important, and this subject is being given much attention at the present time. What is desired is a material which will withstand the scouring and cutting action of the fluid, which tends to enlarge minute leaks between the seat and disk to

serious leaks. Also the rubbing of one part over the other, under heavy pressure exerted by the fluid, requires metals which will slide smoothly over one another, otherwise severe galling and tearing will take place. Mere hardness alone will not prevent cutting, nor can freedom from galling be secured simply by a smooth finish of the bearing surfaces. These are qualities which are apparently not entirely measurable by ordinary means and must be determined by actual service tests. Recently, good results have been secured by using different materials in the disk and seats, there being apparently less tendency for galling to take place.

Straight 12 to 14 per cent chromium stainless steel has been used with success. Welded-on hard surfaces of cobalt-iron-chromium alloys are used for extreme temperature conditions (see also pages 331 to 335, and 346, 347, Chap. III).

Pressure-reducing Valves.—Among the numerous special valves, the pressure-reducing valve is important. Its purpose is to maintain a constant and reduced pressure on a system of piping supplied from a higher pressure source. In the simple rubber-diaphragm type (Fig. 34*a*) the amount of opening of the valve is controlled by the pressure on the reduced-pressure side, conducted through a pilot line to a rubber diaphragm attached to the valve stem. The pressure exerted on the diaphragm is opposed by a weighted lever (or sometimes a spring), and the pressure to be maintained is adjusted by shifting the weight on the lever or adding more weight.

Where the reduced pressure is more than 10 lb or when the service is severe, the pilot type of reducing valve, shown in Fig. 34*b*, is used. In this type of valve the reduced pressure brought back through a pilot line to the diaphragm *B* is opposed by the spring *C*, and the small pilot valve *E* is thus controlled so as to admit just enough high-pressure steam above the piston *F* to maintain the necessary opening of the main valve. There are many variations of this design, but all work on the same basic principle, using a pilot valve which is controlled by the reduced pressure and which operates the main valve.

The pilot type of reducing valve is obviously more complicated than the rubber-diaphragm type and is more difficult to keep in good working order, particularly when the steam contains some solid matter. It should not be used when the simpler valve will serve the purpose. Any type of reducing valve tends to wear at

the disk and seat because of its continual throttling action and cannot be depended upon to close tightly after being in service.* This cutting is often made worse because of a general tendency to select a valve which is too large for the service.

Size of Reducing Valves.—One method of choosing the proper size of reducing valve is to determine the pipe size which will give a reasonable velocity for the amount of steam flowing and then select a reducing valve of the same size. A velocity of 6,000 ft per min on the inlet side is good practice for this purpose.

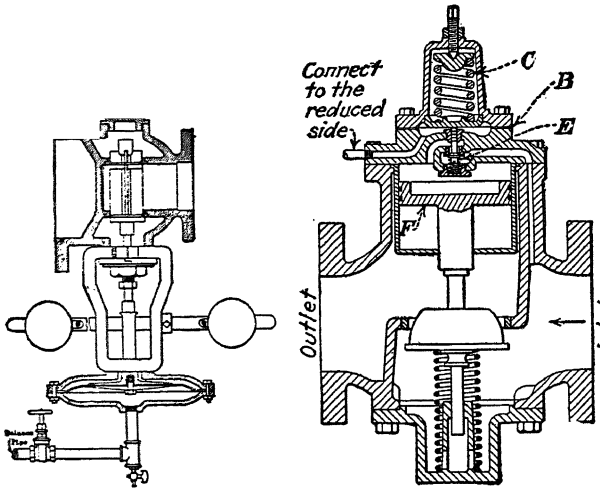


FIG. 34.—Reducing valves, *a*, rubber-diaphragm type; *b*, pilot type.

This method, though rather arbitrary, is recommended by some manufacturers and is very simple. In any case the pressure and temperature conditions and the desired quantity of steam should be specified.

The chart in Fig. 35, page 244 is of assistance in finding the pipe size for a given pressure, velocity, and quantity.

In the case of a reducing valve having a larger outlet than inlet, the inlet size is the nominal size.

Installation.—Reducing valves should invariably be installed with a gate valve on either side and a by-pass which permits steam to be supplied while the valve is out of service for any reason. It

is good practice to put a globe valve and a gate valve in the by-pass, the former for hand throttling, and the latter for tight closing (see Fig. 2 on page 875). A valve that is used for throttling soon becomes worn and will not close tightly.

When the building up of the high pressure on the low-pressure side, owing to improper operation of the reducing valve, would be dangerous, a relief valve should be installed. The ASA Code for Pressure Piping requires that, except in a steam plant used for district heating, one or more relief valves shall be provided in case the piping or equipment on the low-pressure side does not meet the requirements for the full initial pressure. It is mandatory that a pressure gage be installed on the low-pressure side of a reducing valve.

Traps.—The function of a steam trap is to discharge the water of condensation from steam piping without permitting steam to escape. There are three main types. The float trap (Fig. 35*a*) has a hollow float which rises as water enters and, through a system of levers, opens a valve through which the water is discharged. In the bucket type of trap (Fig. 35*b*), water spills over the top of the bucket, causing it to sink and to open the valve. The float trap gives a more or less continuous discharge if the flow of water to it is steady, whereas the bucket trap always discharges intermittently. An inverted bucket-type trap is illustrated in Fig. 35*c*. The size of a trap should be chosen on the basis of its effective valve area or its actual discharge capacity, rather than by the size of the inlet and outlet connections.

A type of trap described as an *impulse* trap is illustrated in Fig. 35*d*. In this type, flashing of the hot condensate tends to force a small piston into the discharge opening when the temperature of the condensate approaches within about 30 F of the saturation temperature. As soon as condensate collected in the drain system cools sufficiently below the flash temperature, the trap opens and discharges accumulated water until the temperature of the condensate once more approaches the saturation temperature and flashing closes the trap, again repeating the cycle. A small orifice permits a continuous discharge of steam or flashed vapor when the trap is closed. Except in the case of draining highly superheated steam lines, it is vapor from the flashed condensate rather than steam which issues from the closed trap. For further explanation of the operation of impulse traps, see *Mechanical Engineering* for August, 1936.

Single orifices are sometimes used to remove condensate from high-pressure high-temperature steam lines. Where the drains are required only in bringing the line up to temperature, the use of orifices is particularly desirable.

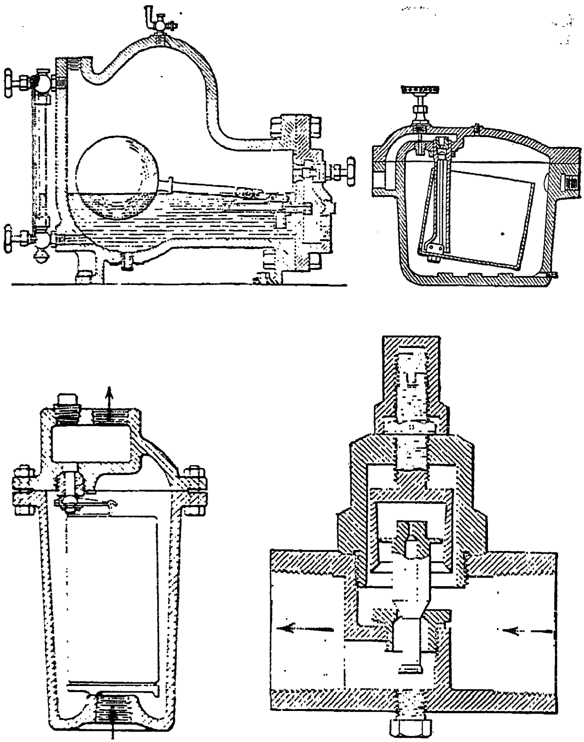


FIG. 35.—Types of steam traps: *a*, float; *b*, bucket; *c*, inverted bucket; *d*, impulse.

Safety Valves.—A common type of safety or relief valve is shown in Fig. 36. Its essential elements are a disk, which is held against a seat by a heavy spring, and a so-called "huddling chamber." A safety valve should pop open quickly and close quickly. The huddling chamber is so constructed that when the valve opens slightly a static pressure is built up in the chamber which immediately

forces the valve wide open. The pressure below the valve must drop a few pounds below the opening pressure before the valve will close. This is known as the "blowdown" and is from 2 to 4 per cent of the set pressure, but not less than 2 lb in any case. The amount of blowdown can be changed by means of the adjusting ring of the huddling chamber. Since reducing the blowdown affects the capacity and operation of the valve, it is mandatory that the blowdown adjustment be made and sealed by the manufacturer.

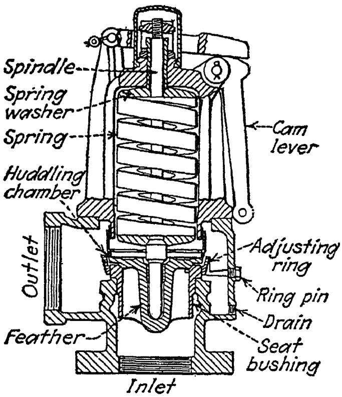


FIG. 36.—Safety valve.

The approximate capacities of typical high-lift safety valves for saturated steam are given in Table LIX. Because of inability to predict the relieving capacity of valves by formulas, the ASME Boiler Construction Code requires that the maximum rated capacity of a safety valve shall be 90 per cent of the flow determined by actual steam-flow tests. The lift in inches and the popping and blowdown pressures, together with the relieving capacity at a pressure 3 per cent over the set pressure, all as determined by tests with steam on valves of corresponding design and construction, must be plainly stamped on each valve.

Further information on Boiler Code requirements for safety valves will be found on pages 877–880. A table of relieving capacities for *air*, expressed in cubic feet per minute of *free* air, is given on page 880.

TABLE LIX.—MAXIMUM RELIEVING CAPACITIES OF POP SAFETY VALVES, POUNDS OF SATURATED STEAM PER HOUR[1]

Valve size, inches	Gage pressures, pounds per square inch														
	5	10	15	20	25	30	40	50	60	75	100	125	150	175	200
1½	750	1,000	1,200	1,300	1,400	1,600	2,000	2,400	2,800	3,500	4,500	5,000	6,000	7,000	8,000
2	1,200	1,500	1,750	2,000	2,000	2,200	2,700	3,200	3,700	4,100	5,000	6,000	7,000	8,000	9,000
2½	1,650	2,100	2,550	2,850	3,000	3,300	3,900	4,500	5,100	6,000	7,500	8,500	9,500	10,500	12,000
3	2,300	2,850	3,400	3,950	4,000	4,400	5,200	6,000	6,800	8,000	10,000	11,500	13,000	14,500	16,000
3½	2,850	3,500	4,200	4,850	5,000	5,500	6,500	7,500	8,500	10,000	12,000	14,000	16,000	18,000	20,000
4	3,500	4,400	5,300	6,050	6,500	7,200	8,600	10,000	11,400	13,000	16,000	18,500	21,000	23,500	26,000
4½	4,300	5,300	6,250	7,250	8,000	8,800	10,400	12,000	13,600	15,500	20,000	23,000	26,000	29,000	32,000
6			9,800	11,750	22,000	25,000	31,000	37,000	44,000	51,000	65,000	79,000	94,000	108,000	122,000

Valve size, inches	Gage pressures, pounds per square inch													
	250	300	350	400	450	500	550	600	650	700	750	800	850	900
1½	9,000	10,000	11,000	12,000	13,000	14,000	15,000	16,000	17,000	18,000	19,000	20,000	22,000	24,000
2	10,500	12,000	14,500	17,000	19,000	21,000	23,000	25,000	27,000	29,000	31,000	33,000	35,000	37,000
2½	14,000	16,000	22,000	28,000	31,000	34,000	37,000	40,000	44,000	48,000	52,000	56,000	71,000	87,000
3	20,000	24,000	31,000	38,000	43,000	48,000	53,000	58,000	63,000	68,000	73,000	78,000	106,000	135,000
3½	25,000	30,000	40,000	50,000	60,000	70,000	80,000	90,000	97,500	105,000	112,500	120,000	162,000	204,000
4	32,000	38,000	51,000	64,000	85,000	106,000	127,000	148,000	178,000	208,000	238,000	268,000	284,000	300,000
4½	39,000	50,000	63,000	80,000	110,000	140,000	170,000	200,000						
6	150,000	179,000	207,000	236,000										

[1] The maximum relieving capacities given are typical of the results guaranteed by several manufacturers for their high-capacity external-spring-type valves at 3 per cent overpressure and where the back pressure does not exceed 60 per cent of the initial pressure. These values represent about the maximum discharge obtainable with the best design of valves made at the present time. In old-style valves the relieving capacities may not exceed one-third to one-half of those given above. Where accurate values are desired, reference should be made to the manufacturer's catalogue. For relieving capacities of safety valves for compressed air, see Table IV, p. 880.

While the relieving capacities given in Table LIX are useful in making preliminary estimates of the size and number of valves required, final selection must be based on guaranteed capacities.

FITTINGS

Designation of Sizes.—In the case of reducing tees, crosses, and Y branches (laterals), the size of the largest run opening shall be given first, followed by the size of the opening at the opposite end of the run. Where the fitting is a tee or Y branch (lateral), the size of the outlet is given next. Where the fitting is a cross,

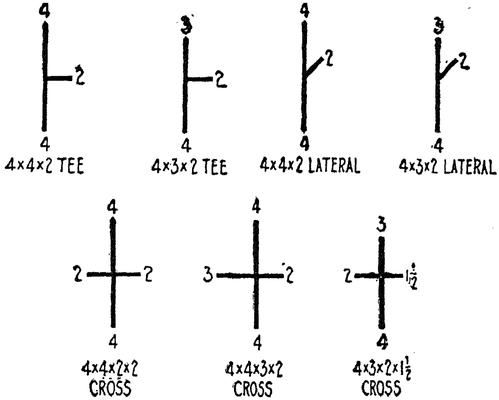


FIG. 37.—Designation of reducing fittings.

the largest opening of outlet is the third dimension given followed by the opening opposite. The straight-line sketches (Fig. 37) illustrate how the reducing fittings are read.

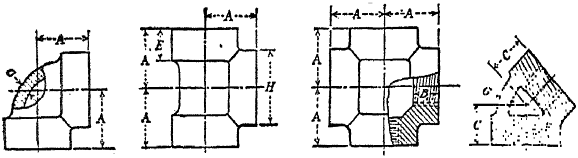
Where an external thread is wanted, the word "male" shall follow the size of that opening.

Cast-iron Screwed Fittings.—Tables LX to LXIV, inclusive, give the American standard dimensions for cast-iron screwed fittings for maximum working saturated steam pressures of 125 and 250 psi gage, ASA B16d-1941. The maximum hydraulic service pressure ratings, including shock, given in ASA B16d are 175 and 400 psi gage, respectively, at or near the ordinary range of air temperatures. Material for cast-iron screwed fittings is required to conform to Class A of ASTM A126. Body thicknesses at no point shall be less than 90 per cent of the specified minimum

TABLE LX.—AMERICAN STANDARD 125-LB CAST-IRON SCREWED FITTINGS

Dimensions of 90- and 45-deg elbows, tees, and crosses (straight

(Table 1, ASA B16d)
(All dimensions in inches)



ELBOW TEE CROSS 45° ELBOW

	A	C	B	E	F		G	H
Nominal pipe size	Center to end, elbows, tees and crosses	Center to end, 45-deg. elbows	Length of thread, minimum	Width of band, minimum	Inside diameter of fitting: Minimum	Inside diameter of fitting: Maximum	Metal thickness, minimum	Outside diameter of band, minimum
1/4	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0.93
3/8	0.95	0.80	0.36	0.44	0.675	0.719	0.120	1.12
1/2	1.12	0.88	0.43	0.50	0.840	0.897	0.130	1.34
3/4	1.31	0.98	0.50	0.56	1.050	1.107	0.155	1.63
1	1.50	1.12	0.58	0.62	1.315	1.385	0.170	1.95
1 1/4	1.75	1.29	0.67	0.69	1.660	1.730	0.185	2.39
1 1/2	1.94	1.43	0.70	0.75	1.900	1.970	0.200	2.68
2	2.25	1.68	0.75	0.84	2.375	2.445	0.220	3.28
2 1/2	2.70	1.95	0.92	0.94	2.875	2.975	0.240	3.86
3	3.08	2.17	0.98	1.00	3.500	3.600	0.260	4.62
3 1/2	3.42	2.39	1.03	1.06	4.000	4.100	0.280	5.20
4	3.79	2.61	1.08	1.12	4.500	4.600	0.310	5.79
5	4.50	3.05	1.18	1.18	5.563	5.663	0.380	7.05
6	5.13	3.46	1.28	1.28	6.625	6.725	0.430	8.28
8	6.56	4.28	1.47	1.47	8.625	8.725	0.550	10.63
10	8.08[1]	5.16	1.68	1.68	10.750	10.850	0.690	13.12
12	9.50[1]	5.97	1.88	1.88	12.750	12.850	0.800	15.47

[1] This applies to elbows and tees only.

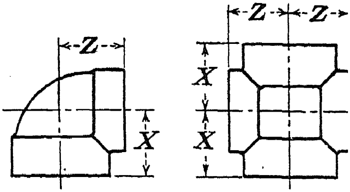
Fittings having right- and left-hand threads shall have four or more ribs or the letter *L* cast o the band at end with left-hand threads.

TABLE LXI.—AMERICAN STANDARD 125-LB CAST-IRON SCREWED FITTINGS

Dimensions of elbows and crosses (reducing sizes)[1]

(Table 2, ASA B16d)

(All dimensions in inches)



Elbows			Crosses		
Nominal pipe size	Center to end		Nominal pipe size	Center to end	
	X	Z		X	Z
¾ × ½	1.20	1.22	¾ × ½	1.20	1.22
1 × ¾	1.37	1.45	1 × ¾	1.37	1.45
1 × ½	1.26	1.36	1¼ × 1	1.58	1.67
1¼ × 1	1.58	1.67	1¼ × ¾	1.45	1.62
1¼ × ¾	1.45	1.62	1½ × 1¼	1.82	1.88
1½ × 1¼	1.82	1.88	1½ × 1	1.65	1.80
1½ × 1	1.65	1.80	1½ × ¾	1.52	1.75
1½ × ¾	1.52	1.75	2 × 1½	2.02	2.16
2 × 1½	2.02	2.16	2 × 1¼	1.90	2.10
2 × 1¼	1.90	2.10	2 × 1	1.73	2.02
2 × 1	1.73	2.02	2 × ¾	1.60	1.97
2 × ¾	1.60	1.97	2½ × 2	2.39	2.60
2½ × 2	2.39	2.60	3 × 2	2.52	2.89
2½ × 1½	2.16	2.51	4 × 3	3.30	3.60
3 × 2½	2.83	2.99	4 × 2	2.74	3.41
3 × 2	2.52	2.89	5 × 4	4.00	4.41
4 × 3	3.30	3.60	6 × 4	4.13	4.94
4 × 2	2.74	3.41			
5 × 4	4.00	4.41			
6 × 5	4.63	5.03			
6 × 4	4.13	4.94			
8 × 6	5.56	6.37			

[1] For dimensions not given, see Table LX.

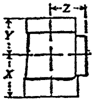


TABLE LXII.—AMERICAN STANDARD 125-LB CAST-IRON SCREWED FITTINGS

Dimensions of tees (reducing sizes)[1]

(Table 3, ASA B16d)

(All dimensions in inches)

Nominal pipe sizes	Center to end			Nominal pipe sizes	Center to end		
	X	Y	Z		X	Y	Z
½ × ½ × ¾	1.22	1.22	1.20	1½ × ¾ × 1½	1.94	1.75	1.94
½ × ½ × ⅜	1.04	1.04	1.03	1½ × ¾ × 1¼	1.82	1.62	1.88
¾ × ¾ × 1	1.45	1.45	1.37	1½ × ½ × 1½	1.94	1.66	1.94
¾ × ¾ × ½	1.20	1.20	1.22	2 × 2 × 3	2.89	2.89	2.52
¾ × ¾ × ⅜	1.12	1.12	1.13	2 × 2 × 2½	2.60	2.60	2.39
¾ × ½ × ¾	1.31	1.22	1.31	2 × 2 × 1½	2.02	2.02	2.16
¾ × ½ × ½	1.20	1.12	1.22	2 × 2 × 1¼	1.90	1.90	2.10
1 × 1 × 1½	1.80	1.80	1.65	2 × 2 × 1	1.73	1.73	2.02
1 × 1 × 1¼	2.10	2.10	1.90	2 × 2 × ¾	1.60	1.60	1.97
1 × 1 × ¾	1.37	1.37	1.45	2 × 2 × ½	1.49	1.49	1.88
1 × 1 × ½	1.26	1.26	1.36	2 × 1½ × 2½	2.60	2.51	2.39
1 × 1 × ⅜	1.18	1.18	1.27	2 × 1½ × 2	2.25	2.16	2.25
1 × ¾ × 1	1.50	1.45	1.50	2 × 1½ × 1½	2.02	1.94	2.16
1 × ¾ × ¾	1.37	1.31	1.45	2 × 1½ × 1¼	1.90	1.82	2.10
1 × ¾ × ½	1.26	1.20	1.36	2 × 1½ × 1	1.73	1.65	2.02
1 × ½ × 1	1.50	1.36	1.50	2 × 1½ × ¾	1.60	1.52	1.97
1 × ½ × ¾	1.37	1.22	1.45	2 × 1½ × ½	1.49	1.41	1.88
1 × ⅜ × 1	1.50	1.27	1.50	2 × 1¼ × 2	2.25	2.10	2.25
1¼ × 1¼ × 2	1.67	1.67	1.58	2 × 1¼ × 1½	2.02	1.88	2.16
1¼ × 1¼ × 1½	1.88	1.88	1.82	2 × 1¼ × 1¼	1.90	1.75	2.10
1¼ × 1¼ × 1	1.58	1.58	1.67	2 × 1¼ × 1	1.73	1.58	2.02
1¼ × 1¼ × ¾	1.45	1.45	1.62	2 × 1 × 2	2.25	2.02	2.25
1¼ × 1¼ × ½	1.34	1.34	1.53	2 × 1 × 1½	2.02	1.80	2.16
1¼ × 1 × 1½	1.88	1.80	1.82	2 × ¾ × 2	2.25	1.97	2.25
1¼ × 1 × 1¼	1.75	1.67	1.75	2½ × 2½ × 4	3.51	3.51	3.05
1¼ × 1 × 1	1.58	1.50	1.67	2½ × 2½ × 3	2.99	2.99	2.83
1¼ × 1 × ¾	1.45	1.37	1.62	2½ × 2½ × 2	2.39	2.39	2.60
1¼ × 1 × ½	1.34	1.25	1.53	2½ × 2½ × 1½	2.16	2.16	2.51
1¼ × ¾ × 1¼	1.75	1.62	1.75	2½ × 2½ × 1¼	2.04	2.04	2.45
1¼ × ¾ × 1	1.58	1.45	1.67	2½ × 2½ × 1	1.87	1.87	2.37
1¼ × ½ × 1¼	1.75	1.53	1.75	2½ × 2½ × ¾	1.74	1.74	2.32
1½ × 1½ × 2	2.16	2.16	2.02	2½ × 2 × 2½	2.70	2.60	2.70
1½ × 1½ × 1¼	1.82	1.82	1.88	2½ × 2 × 2	2.39	2.25	2.60
1½ × 1½ × 1	1.65	1.65	1.80	2½ × 2 × 1½	2.16	2.02	2.51
1½ × 1½ × ¾	1.52	1.52	1.75	2½ × 2 × 1¼	2.04	1.90	2.45
1½ × 1½ × ½	1.41	1.41	1.66	2½ × 2 × 1	1.87	1.73	2.37
1½ × 1¼ × 1½	1.94	1.88	1.94	2½ × 2 × ¾	1.74	1.60	2.32
1½ × 1¼ × 1¼	1.82	1.75	1.88	2½ × 2 × ½	1.63	1.49	2.23
1½ × 1¼ × 1	1.65	1.58	1.80	2½ × 1½ × 2½	2.70	2.51	2.70
1½ × 1¼ × ¾	1.52	1.45	1.75	2½ × 1½ × 2	2.39	2.16	2.60
1½ × 1¼ × ½	1.41	1.34	1.66	2½ × 1½ × 1½	2.16	1.94	2.51
1½ × 1 × 2	2.16	2.02	2.02	2½ × 1 × 2½	2.70	2.37	2.70
1½ × 1 × 1½	1.94	1.80	1.94	3 × 3 × 4	3.60	3.60	3.30
1½ × 1 × 1¼	1.82	1.67	1.88	3 × 3 × 3½	3.33	3.33	3.18
1½ × 1 × 1	1.65	1.50	1.80	3 × 3 × 2½	2.83	2.83	2.99

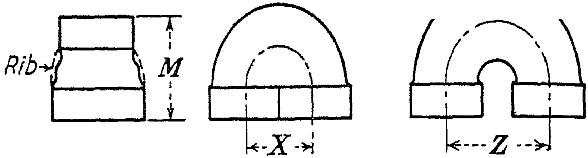
[1] For dimensions not given, see Table LX.

TABLE LXII.—(*Concluded*)

Nominal pipe sizes	Center to end			Nominal pipe sizes	Center to end		
	X	Y	Z		X	Y	Z
3 × 3 × 2	2.52	2.52	2.89	4 × 2 × 4	3.79	3.41	3.79
3 × 3 × 1½	2.29	2.29	2.80	4 × 1½ × 4	3.79	3.32	3.79
3 × 3 × 1¼	2.17	2.17	2.74	4 × 1¼ × 4	3.79	3.26	3.79
3 × 3 × 1	2.00	2.00	2.66	5 × 5 × 6	5.03	5.03	4.63
3 × 3 × ¾	1.87	1.87	2.61	5 × 5 × 4	4.00	4.00	4.41
3 × 2½ × 3	3.08	2.99	3.08	5 × 5 × 3½	3.75	3.75	4.31
3 × 2½ × 2½	2.83	2.70	2.99	5 × 5 × 3	3.51	3.51	4.22
3 × 2½ × 2	2.52	2.39	2.89	5 × 5 × 2½	3.26	3.26	4.13
3 × 2½ × 1½	2.29	2.16	2.80	5 × 5 × 2	2.95	2.94	4.03
3 × 2½ × 1¼	2.17	2.04	2.74	5 × 5 × 1½	2.72	2.72	3.94
3 × 2½ × 1	2.00	1.87	2.66	5 × 4 × 5	4.50	4.41	4.50
3 × 2 × 3	3.08	2.89	3.08	5 × 4 × 4	4.00	3.79	4.41
3 × 2 × 2½	2.83	2.60	2.99	5 × 4 × 3	3.51	3.30	4.22
3 × 2 × 2	2.52	2.25	2.90	5 × 4 × 2	2.95	2.74	4.03
3 × 2 × 1½	2.29	2.02	2.80	5 × 3 × 5	4.50	4.22	4.50
3 × 1½ × 3	3.08	2.80	3.08	5 × 3 × 4	4.00	3.60	4.41
3 × 1 × 3	3.08	2.66	3.08	5 × 3 × 3	3.51	3.08	4.22
3½ × 3½ × 3	3.18	3.18	3.33	5 × 2 × 5	4.50	4.03	4.50
3½ × 3½ × 2½	2.93	2.93	3.24	6 × 6 × 8	6.37	6.37	5.56
3½ × 3½ × 2	2.62	2.62	3.14	6 × 6 × 5	4.63	4.63	5.03
3½ × 3½ × 1½	2.39	2.39	3.05	6 × 6 × 4	4.13	4.13	4.94
3½ × 2½ × 1¼	2.27	2.72	2.99	6 × 6 × 3	3.64	3.64	4.75
3½ × 3 × 3	3.18	3.08	3.33	6 × 6 × 2½	3.39	3.39	4.66
3½ × 3 × 2½	2.93	2.83	3.24	6 × 6 × 2	3.08	3.08	4.56
3½ × 3 × 2	2.62	2.52	3.14	6 × 5 × 5	4.63	4.50	5.03
3½ × 3 × 1½	2.39	2.29	3.05	6 × 5 × 4	4.13	4.00	4.94
3½ × 2 × 3½	3.42	3.14	3.42	6 × 4 × 6	5.13	4.94	5.13
3½ × 1½ × 3½	3.42	3.05	3.42	6 × 4 × 4	4.13	3.79	4.94
4 × 4 × 6	4.94	4.94	4.13	6 × 3 × 6	5.13	4.75	5.13
4 × 4 × 5	4.41	4.41	4.00	6 × 2 × 6	5.13	4.56	5.13
4 × 4 × 3½	3.54	3.54	3.69	8 × 8 × 6	5.56	5.56	6.37
4 × 4 × 3	3.30	3.30	3.60	8 × 8 × 5	5.03	5.03	6.27
4 × 4 × 2½	3.05	3.05	3.51	8 × 8 × 4	4.50	4.50	6.17
4 × 4 × 2	2.74	2.74	3.41	8 × 8 × 3	4.00	4.00	6.07
4 × 4 × 1½	2.51	2.51	3.32	8 × 8 × 2½	3.69	3.69	6.01
4 × 4 × 1½	2.39	2.39	3.26	8 × 8 × 2	3.44	3.44	5.84
4 × 4 × 1	2.22	2.22	3.18	8 × 6 × 8	6.56	6.37	6.56
4 × 3½ × 3	3.30	3.18	3.60	8 × 6 × 6	5.56	5.13	6.37
4 × 3½ × 2½	3.05	2.93	3.51				
4 × 3½ × 2	2.74	2.62	3.41				
4 × 3 × 4	3.79	3.60	3.79				
4 × 3 × 3	3.30	3.08	3.60				
4 × 3 × 2	2.74	2.52	3.41				
4 × 2½ × 4	3.79	3.51	3.79				
4 × 2½ × 2½	3.05	2.70	3.51				

TABLE LXIII.—AMERICAN STANDARD 125-LB CAST-IRON SCREWED FITTINGS

Dimensions of reducing couplings and return bends
(Tables 4 and 6, ASA B16d)
(All dimensions in inches)



Nominal pipe size	Length of thread, min	Width of band, min	Inside diameter of fitting F		Metal thickness[1]	Outside diameter of band, min	Length of reducing couplings	Center to center of return bends	
			Max	Min				Close	Open
	B	E			G	H	M	X	Z
½	0.43	0.50	0.897	0.840	0.130	1.34	1.38	1.25	1.75
¾	0.50	0.56	1.107	1.050	0.155	1.63	1.50	1.50	1.88
1	0.58	0.62	1.385	1.315	0.170	1.95	1.70	1.75	2.50
1¼	0.67	0.69	1.730	1.660	0.185	2.39	2.13	2.25	3.00
1½	0.70	0.75	1.970	1.900	0.200	2.68	2.25	2.50	3.50
2	0.75	0.84	2.445	2.375	0.220	3.28	2.32	3.25	4.50
2½	0.92	0.94	2.975	2.875	0.240	3.86	2.63	3.75	5.50
3	0.98	1.00	3.600	3.500	0.260	4.62	2.88	4.50	6.50
3½	1.03	1.06	4.100	4.000	0.280	5.20	3.13		
4	1.08	1.12	4.600	4.500	0.310	5.79	3.38	6.00	7.50
5	1.18	1.18	5.663	5.563	0.380	7.05	3.57		
6	1.28	1.28	6.725	6.625	0.430	8.28	3.81		
8	1.47	1.47	8.725	8.625	0.550	10.63	5.25		
10	1.68	1.68	10.850	.....	0.690	13.12			
12	1.88	1.88	12.850	.....	0.800	15.47			

The use of ribs is optional with the manufacturer. For caps, see abstract of ASA B16.14, pp. 584, 585.

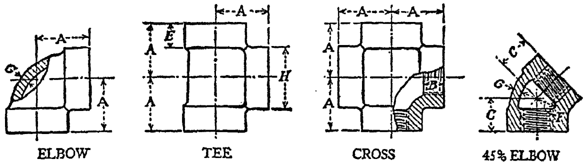
[1] Patterns shall be designed to produce castings of metal thicknesses given in the table. Metal thickness at no point shall be less than 90 per cent of thickness given in the table.

TABLE LXIV.—AMERICAN STANDARD 250-LB CAST-IRON SCREWED FITTINGS

Dimensions of elbows, 45-deg elbows, tees, and crosses, (straight sizes)

(Table 1, ASA B16d)

(All dimensions in inches)



Nominal pipe size	*A* Center to end, elbows, tees, and crosses	*C* Center to end, 45-deg. elbows	*B* Length of thread, minimum	*E* Width of band, minimum	*F* Inside diameter of fitting, Minimum	*F* Inside diameter of fitting, Maximum	*G* Metal thickness, minimum	*H* Outside diameter of band, minimum
¼	0.94	0.81	0.43	0.49	0.540	0.584	0.18	1.17
⅜	1.06	0.88	0.47	0.55	0.675	0.719	0.18	1.36
½	1.25	1.00	0.57	0.60	0.840	0.897	0.20	1.59
¾	1.44	1.13	0.64	0.68	1.050	1.107	0.23	1.88
1	1.63	1.31	0.75	0.76	1.315	1.385	0.28	2.24
1¼	1.94	1.50	0.84	0.88	1.660	1.730	0.33	2.73
1½	2.13	1.69	0.87	0.97	1.900	1.970	0.35	3.07
2	2.50	2.00	1.00	1.12	2.375	2.445	0.39	3.74
2½	2.94	2.25	1.17	1.30	2.875	2.975	0.43	4.60
3	3.38	2.50	1.23	1.40	3.500	3.600	0.48	5.36
3½	3.75	2.63	1.28	1.49	4.000	4.100	0.52	5.98
4	4.13	2.81	1.33	1.57	4.500	4.600	0.56	6.61
5	4.88	3.19	1.43	1.74	5.563	5.663	0.66	7.92
6	5.63	3.50	1.53	1.91	6.625	6.725	0.74	9.24
8	7.00	4.31	1.72	2.24	8.625	8.725	0.90	11.73
10	8.63	5.19	1.93	2.58	10.750	10.850	1.08	14.37
12	10.00	6.00	2.13	2.91	12.750	12.850	1.24	16.84

The 250-lb standard for screwed fittings covers only the straight sizes of 90- and 45-deg elbows, tees, and crosses.

metal thickness. Cast-iron screwed fittings customarily are furnished in black only.

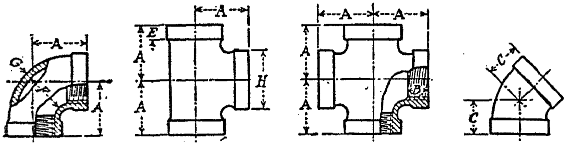
Malleable-iron Screwed Fittings.—Malleable-iron screwed fittings for 150-lb gage are given in ASA B16c. Malleable-iron fittings are furnished either black or galvanized. Dimensions are given in Tables LXV to LXXI. The galvanized screwed fittings

TABLE LXV.—AMERICAN STANDARD 150-LB MALLEABLE-IRON SCREWED FITTINGS

Dimensions of elbows, tees, crosses, and 45-deg elbows (straight sizes)

(Table 1, ASA B16c)

(All dimensions in inches)



ELBOW TEE CROSS 45° ELBOW

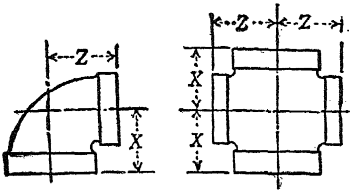
Nominal pipe size	A	C	B	E	F		G	H
	Center to end, elbows, tees, and crosses	Center to end, 45-deg. elbows	Length of thread, minimum	Width of band, minimum	Inside diameter of fitting		Metal thickness, minimum	Outside diameter of band, minimum
					Minimum	Maximum		
1/8	0.69	....	0.25	0.200	0.405	0.435	0.090	0.693
1/4	0.81	0.73	0.32	0.215	0.540	0.584	0.095	0.844
3/8	0.95	0.80	0.36	0.230	0.675	0.719	0.100	1.015
1/2	1.12	0.88	0.43	0.249	0.840	0.897	0.105	1.197
3/4	1.31	0.98	0.50	0.273	1.050	1.107	0.120	1.458
1	1.50	1.12	0.58	0.302	1.315	1.385	0.134	1.771
1 1/4	1.75	1.29	0.67	0.341	1.660	1.730	0.145	2.153
1 1/2	1.94	1.43	0.70	0.368	1.900	1.970	0.155	2.427
2	2.25	1.68	0.75	0.422	2.375	2.445	0.173	2.963
2 1/2	2.70	1.95	0.92	0.478	2.875	2.975	0.210	3.589
3	3.08	2.17	0.98	0.548	3.500	3.600	0.231	4.285
3 1/2	3.42	2.39	1.03	0.604	4.000	4.100	0.248	4.843
4	3.79	2.61	1.08	0.661	4.500	4.600	0.265	5.401
5	4.50	3.05	1.18	0.780	5.563	5.663	0.300	6.583
6	5.13	3.46	1.28	0.900	6.625	6.725	0.336	7.767

Dimensions for reducing elbows and reducing crosses are given in Table LXVI, and dimensions for reducing tees in Table LXVII.

commonly used in house water piping are 150-lb malleable iron. Malleable iron is required to conform to ASTM Specification A157 for Cupola Malleable Iron. Emergency specification ASTM ES20 covering malleable-iron flanges, pipe fittings, and valve parts permits conformance with either ASTM A47 or A197, unless the selection is made by the purchaser. See pages 344 and 345 for the physical properties of malleable iron.

TABLE LXVI.—AMERICAN STANDARD 150-LB MALLEABLE-IRON SCREWED FITTINGS

Dimensions of elbows and crosses (reducing sizes)
(Table 2, ASA B16c)
(All dimensions in inches)

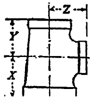


Elbows: Nominal pipe sizes	Elbows: Center to end *X*	Elbows: Center to end *Z*	Crosses: Nominal pipe sizes	Crosses: Center to end *X*	Crosses: Center to end *Z*
3/8 × 1/4	0.88	0.90	3/4 × 1/2	1.20	1.22
1/2 × 3/8	1.04	1.03	1 × 3/4	1.37	1.45
3/4 × 1/2	1.20	1.22	1 × 1/2	1.26	1.36
3/4 × 3/8	1.12	1.13	1 1/4 × 1	1.58	1.67
1 × 3/4	1.37	1.45	1 1/4 × 3/4	1.45	1.62
1 × 1/2	1.26	1.36	1 1/2 × 1 1/4	1.82	1.88
1 1/4 × 1	1.58	1.67	1 1/2 × 1	1.65	1.80
1 1/4 × 3/4	1.45	1.62	1 1/2 × 3/4	1.52	1.75
1 1/2 × 1 1/4	1.82	1.88	2 × 1 1/2	2.02	2.16
1 1/2 × 1	1.65	1.80	2 × 1 1/4	1.90	2.10
1 1/2 × 3/4	1.52	1.75	2 × 1	1.73	2.02
2 × 1 1/2	2.02	2.16	2 × 3/4	1.60	1.97
2 × 1 1/4	1.90	2.10	2 1/2 × 2	2.39	2.60
2 × 1	1.73	2.02	3 × 2	2.52	2.89
2 1/2 × 2	2.39	2.60			
3 × 2 1/2	2.83	2.99			
3 × 2	2.52	2.89			
4 × 3	3.30	3.60			

Malleable-iron screwed unions, both black and galvanized, are available for 150-, 250-, and 300-psi steam service. Ground-joint bronze or brass-to-iron and brass-to-brass types are made. Gasket-type screwed unions usually are limited to 150-lb steam and equivalent water service. Union fittings such as elbows and tees are available with either male or female threaded ends. For end-to-end and center-to-end dimensions of unions and union fittings

TABLE LXVII.—AMERICAN STANDARD 150-LB MALLEABLE-IRON SCREWED FITTINGS

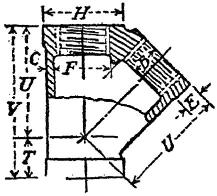
Dimensions of tees (reducing sizes)
(Table 3, ASA B16c)
(All dimensions in inches)



Nominal pipe sizes	Center to end		
	X	Y	Z
¼ × ¼ × ⅜	0.90	0.90	0.88
⅜ × ⅜ × ½	1.03	1.03	1.04
⅜ × ⅜ × ¼	0.88	0.88	0.90
⅜ × ⅜ × ⅛	0.81	0.81	0.85
⅜ × ¼ × ⅜	0.95	0.90	0.95
⅜ × ¼ × ¼	0.88	0.81	0.90
½ × ½ × ¾	1.22	1.22	1.20
½ × ½ × ⅜	1.04	1.04	1.03
½ × ½ × ¼	0.97	0.97	0.98
½ × ⅜ × ½	1.12	1.03	1.12
½ × ⅜ × ⅜	1.04	0.95	1.03
¾ × ¾ × 1	1.45	1.45	1.37
¾ × ¾ × ½	1.20	1.20	1.22
¾ × ¾ × ⅜	1.12	1.12	1.13
¾ × ¾ × ¼	1.05	1.05	1.08
¾ × ½ × ¾	1.31	1.22	1.31
¾ × ½ × ½	1.20	1.12	1.22
¾ × ½ × ⅜	1.12	1.04	1.13
1 × 1 × 1½	1.80	1.80	1.65
1 × 1 × 1¼	1.67	1.67	1.58
1 × 1 × ¾	1.37	1.37	1.45
1 × 1 × ½	1.26	1.26	1.36
1 × 1 × ⅜	1.18	1.18	1.27
1 × ¾ × 1	1.50	1.45	1.50
1 × ¾ × ¾	1.37	1.31	1.45
1 × ¾ × ½	1.26	1.20	1.36
1 × ½ × 1	1.50	1.36	1.50
1 × ½ × ¾	1.37	1.22	1.45
1 × ½ × ½	1.26	1.12	1.36
1¼ × 1¼ × 1½	1.88	1.88	1.82
1¼ × 1¼ × 1	1.58	1.58	1.67
1¼ × 1¼ × ¾	1.45	1.45	1.62
1¼ × 1¼ × ½	1.34	1.34	1.53
1¼ × 1¼ × ⅜	1.26	1.26	1.44
1¼ × 1 × 1¼	1.75	1.67	1.75
1¼ × 1 × 1	1.58	1.50	1.67
1¼ × 1 × ¾	1.45	1.37	1.62
1¼ × ¾ × 1¼	1.75	1.62	1.75
1¼ × ¾ × 1	1.58	1.45	1.67
1¼ × ¾ × ¾	1.45	1.31	1.62
1½ × 1½ × 2	2.16	2.16	2.02
1½ × 1½ × 1¼	1.82	1.82	1.88
1½ × 1½ × 1	1.65	1.65	1.80
1½ × 1½ × ¾	1.52	1.52	1.75
1½ × 1½ × ½	1.41	1.41	1.66
1½ × 1¼ × 1½	1.94	1.88	1.94
1½ × 1¼ × 1¼	1.82	1.75	1.88
1½ × 1¼ × 1	1.65	1.58	1.80
1½ × 1 × 1½	1.94	1.80	1.94
1½ × 1 × 1	1.65	1.50	1.80
1½ × ¾ × 1½	1.94	1.75	1.94
2 × 2 × 2½	2.60	2.60	2.39
2 × 2 × 1½	2.02	2.02	2.16
2 × 2 × 1¼	1.90	1.90	2.10
2 × 2 × 1	1.73	1.73	2.02
2 × 2 × ¾	1.60	1 60	1.97
2 × 2 × ½	1.49	1.49	1.88
2 × 1½ × 2	2.25	2.16	2.25
2 × 1½ × 1½	2.02	1.94	2.16
2 × 1½ × 1¼	1.90	1.82	2.10
2 × 1¼ × 2	2.25	2.10	2.25
2 × 1¼ × 1½	2.02	1.88	2.16
2 × 1 × 2	2.25	2.02	2.25
2 × 1 × 1½	2.02	1.80	2.16
2 × ¾ × 2	2.25	1.97	2.25
2½ × 2½ × 3	2.99	2.99	2.83
2½ × 2½ × 2	2.39	2.39	2.60
2½ × 2½ × 1½	2.16	2.16	2.51
2½ × 2½ × 1¼	2.04	2.04	2.45
2½ × 2½ × 1	1.87	1.87	2.37
2½ × 2½ × ¾	1.74	1.74	2.32
2½ × 2 × 2½	2.70	2.60	2.70
2½ × 2 × 2	2.39	2.25	2.60
2½ × 1½ × 2½	2.70	2.51	2.70
3 × 3 × 2½	2.83	2.83	2.99
3 × 3 × 2	2.52	2.52	2.89
3 × 3 × 1½	2.29	2.29	2.80
3 × 3 × 1¼	2.17	2.17	2.74
3 × 3 × 1	2.00	2.00	2.66
3 × 2½ × 3	3.08	2.99	3.08
3 × 2 × 3	3.08	2.89	3.08
3 × 2 × 2	2.52	2.25	2.89
3½ × 3½ × 2½	2.93	2.93	3.24
4 × 4 × 3	3.30	3.30	3.60
4 × 4 × 2½	3.05	3.05	3.51
4 × 4 × 2	2.74	2.74	3.41
5 × 5 × 3	3.51	3.51	4.22
6 × 6 × 4	4.13	4.13	4.94
6 × 6 × 3	3.64	3.64	4.75
6 × 6 × 2½	3.39	3.39	4.66
6 × 6 × 2	3.08	3.08	4.56

for domestic service and usual industrial purposes reference should be made to the manufacturer's catalogues. The U. S. Department of Commerce, Commercial Standard CS7-29 on Standard Weight Malleable Iron or Steel Screwed Unions covers requirements for

TABLE LXVIII.—AMERICAN STANDARD 150-LB MALLEABLE-IRON SCREWED FITTINGS
Dimensions of 45-deg Y branches (straight sizes)
(Table 4, ASA B16c)
(All dimensions in inches)



Nominal pipe size	B	E	F		G	H	T	U	V
	Length of thread, minimum	Width of band, minimum	Inside diameter of fittings		Metal thickness, minimum	Outside diameter of band, minimum	Center to end, inlet	Center to end, outlet	End to end
			Minimum	Maximum					
3/8	0.36	0.230	0.675	0.719	0.100	1.015	0.50	1.43	1.93
1/2	0.43	0.249	0.840	0.897	0.105	1.197	0.61	1.71	2.32
3/4	0.50	0.273	1.050	1.107	0.120	1.458	0.72	2.05	2.77
	0.58	0.302	1.315	1.385	0.134	1.771	0.85	2.43	3.28
1¼	0.67	0.341	1.660	1.730	0.145	2.153	1.02	2.92	3.94
1½	0.70	0.368	1.900	1.970	0.155	2.427	1.10	3.28	4.38
2	0.75	0.422	2.375	2.445	0.173	2.963	1.24	3.93	5.17
2½	0.92	0.478	2.875	2.975	0.210	3.589	1.52	4.73	6.25
3	0.98	0.548	3.500	3.600	0.231	4.285	1.71	5.55	7.26
4	1.08	0.661	4.500	4.600	0.265	5.401	2.01	6.97	8.98

Patterns shall be designed to produce castings to body thicknesses given in table. Body thicknesses at no point shall be less than 90 per cent of the thickness given in the table.

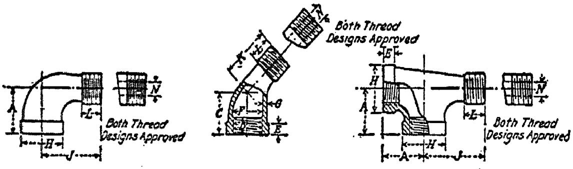
unions suitable for use with standard-weight pipe. For railroad and other exacting service, malleable-iron unions and union fittings conform to the specifications and recommended practice of the Association of American Railroads and are listed as AAR Malleable Iron Unions and Union Fittings.

Table LXIX.—American Standard 150-lb Malleable-iron Screwed Fittings

Dimensions of service or street tees and 90- and 45-deg elbows (straight sizes)

(Table 5, ASA B16c)

(All dimensions in inches)



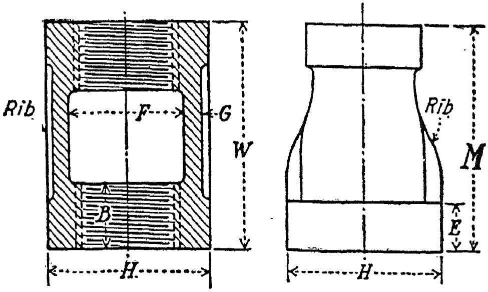
	A	C	B	E	F		G	H	J	K	L	N
	...enter ...lbows,	...enter ...45-deg	...gth of thre... ...nimum	...h of band, ...imum	Inside diameter of fittings					Cent... end elbo...		
1/8	0.69*	....	0.25	0.200	0.405	0.435	0.090	0.693	1.00*		0.2638	0.15
1/4	0.81	0.73	0.32	0.215	0.540	0.584	0.095	0.844	1.19	0.94	0.4018	0.26
3/8	0.95	0.80	0.36	0.230	0.675	0.719	0.100	1.015	1.44	.03	0.4078	0.37
1/2	1.12	0.88	0.43	0.249	0.840	0.897	0.105	1.197	1.63	.15	0.5337	0.51
3/4	1.31	0.98	0.50	0.273	1.050	1.107	0.120	1.458	1.89	.29	0.5457	0.69
1	1.50	1.12	0.58	0.302	1.315	1.385	0.134	1.771	2.14	.47	0.6828	0.91
1 1/4	1.75	1.29	0.67	0.341	1.660	1.730	0.145	2.153	2.45	.71	0.7068	1.19
1 1/2	1.94	1.43	0.70	0.368	1.900	1.970	0.155	2.427	2.69		0.7235	1.39
2	2.25	1.68	0.75	0.422	2.375	2.445	0.173	2.963	3.26	2.22	0.7565	1.79
2 1/2	2.70	1.95	0.92	0.478	2.875	2.975	0.210	3.589	3.86	2.57	.1375	2.20
3	3.08	2.17	0.98	0.548	3.500	3.600	0.231	4.285	4.51	3.00	.2000	2.78
3 1/2	3.42*	2.39	1.03	0.604	4.000	4.100	0.248	4.843	5.09*		.2500	3.24
	3.79	2.61	1.08	0.661	4.500	4.600	0.265	5.401	5.69	3.70	.3000	3.70
	4.50*	3.05	1.18	0.780	5.563	5.663	0.300	6.583	6.86*		.4063	4.69
	5.13*	3.46	1.28	0.900	6.625	6.725	0.336	7.767	8.03*		.5125	5.67

Patterns shall be designed to produce castings of body thicknesses given in table. Body thicknesses at no point shall be less than 90 per cent of the thickness given in the table.

* This dimension applies to street elbows only. Street tees are not made in these sizes.

TABLE LXX.—AMERICAN STANDARD 150-LB MALLEABLE-IRON SCREWED FITTINGS

Dimensions of couplings (straight and reducing sizes)
(Table 6, ASA B16c)
(All dimensions in inches)



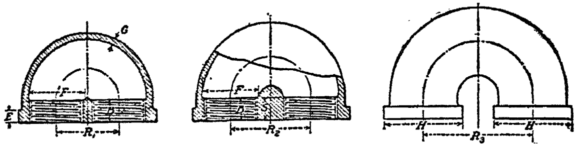
	B	E	F		G	H		W	M
Nominal pipe	Length of thread, mini-	Width of band, minimum	Inside diameter of fittings Minimum	Maxi-	Metal thickness, minimum	Outside diameter of band, minimum	Thick-ness of ribs[1]	Length of straight couplings	Length reducing coupl-
⅛	0.25	0.200	0.405	0.435	0.090	0.693	0.090	0.96	
¼	0.32	0.215	0.540	0.584	0.095	0.844	0.095	.06	1.00
⅜	0.36	0.230	0.675	0.719	0.100	1.015	0.100	.16	1.13
½	0.43	0.249	0.840	0.897	0.105	1.197	0.105	.34	1.25
¾	0.50	0.273	1.050	1.107	0.120	1.458	0.120	.52	1.44
1	0.58	0.302	1.315	1.385	0.134	1.771	0.134	.67	1.69
1¼	0.67	0.341	1.660	1.730	0.145	2.153	0.145	.93	2.06
1½	0.70	0.368	1.900	1.970	0.155	2.427	0.155	2.15	2.31
2	0.75	0.422	2.375	2.445	0.173	2.963	0.173	2.53	2.81
2½	0.92	0.478	2.875	2.975	0.210	3.589	0.210	2.88	3.25
3	0.98	0.548	3.500	3.600	0.231	4.285	0.231	3.18	3.69
3½	1.03	0.604	4.000	4.100	0.248	4.843	0.248	3.43	4.00
4	1.08	0.661	4.500	4.600	0.265	5.401	0.265	3.69	4.38

Patterns shall be designed to produce castings of body thicknesses given in table. Body thicknesses at no point shall be less than 90 per cent of the thickness given in the table.

[1] Right-hand couplings have two ribs and right- and left-hand couplings have four or more ribs.

Table LXXI.—American Standard 150-lb Malleable-iron Screwed Fittings

Dimensions of close, medium, and open pattern return bends
(Table 8, ASA B16c)
(All dimensions in inches)



CLOSE MEDIUM

	B[1]	E[2]	F		G	H	R_1	R_2	R_3
Nominal pipe	Length of thread, minimum	Width of band, minimum	Inside diameter of fittings		Metal thickness, minimum	Outside diameter of band, minimum	Center to center, close pattern	Center to center, medium pattern	Center to center, open pattern
			Minimum	Maximum					
½	0.43	0.249	0.840	0.897	0.116	1.197	1.000	1.25	1.50
¾	0.50	0.273	1.050	1.107	0.133	1.458	1.250	1.50	2.00
1	0.58	0.302	1.315	1.385	0.150	1.771	1.500	1.875	2.50
1¼	0.67	0.341	1.660	1.730	0.165	2.153	1.750	2.25	3.00
1½	0.70	0.368	1.900	1.970	0.178	2.427	2.188	2.50	3.50
2	0.75	0.422	2.375	2.445	0.201	2.963	2.625	3.000	4.00
2½	0.92	0.478	2.875	2.975	0.244	3.589	.....	.....	4.50
3	0.98	0.548	3.500	3.600	0.272	4.285	.....	.....	5.00

Patterns shall be designed to produce castings of body thicknesses given in table. Body thicknesses at no point shall be less than 90 per cent of the thickness given in the table.

[1] Dimension S in cut is "face to point of radius" dimension and is same as dimension B in table.

[2] It is permissible to furnish pattern return bends not banded. Close pattern return bends will not make up parallel coils, as the distance center to center of two adjacent bends is greater than the center-to-center of openings of a single bend.

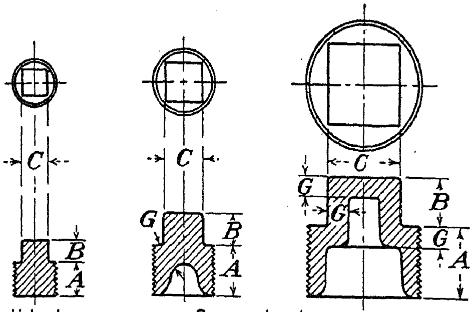
Note.—For 150-lb malleable-iron pipe bushings, see abstract of ASA B16.14, p. 582.

American Standard for
FERROUS PLUGS, BUSHINGS, LOCKNUTS, AND CAPS WITH PIPE THREADS
ASA B16.14-1943
Abstracted[1]

This standard brings together pipe plugs, pipe bushings and locknuts, and pipe caps, the dimensions of which originally were published in separate stand-

[1] For complete standard, reference may be made to ASA B16.14 (see note, p. 430).

TABLE LXXII.—DIMENSIONS OF SQUARE-HEAD PLUGS
(Table 1, ASA B16.14)
(All dimensions in inches)



Solid plugs   Cored plugs

Nominal[1,2] pipe size	Thread length, min *A*	Height of square, min *B*	Width across flats C[3] Nom	Width across flats C[3] Max	Metal thickness,[4] min *G*
⅛	0.37	0.24	9/32	0.281	
¼	0.44	0.28	⅜	0.375	
⅜	0.48	0.31	7/16	0.438	
½	0.56	0.38	9/16	0.563	0.16
¾	0.63	0.44	⅝	0.625	0.18
1	0.75	0.50	13/16	0.813	0.20
1¼	0.80	0.56	15/16	0.938	0.22
1½	0.83	0.62	1⅛	1.125	0.24
2	0.88	0.68	1 5/16	1.313	0.26
2½	1.07	0.74	1½	1.500	0.29
3	1.13	0.80	1 11/16	1.688	0.31
3½	1.18	0.86	1⅞	1.875	0.34

Material to be cast iron, malleable iron, or steel.

These plugs are threaded with American Standard tapered pipe threads, ASA B2.1, see abstract, p. 431.

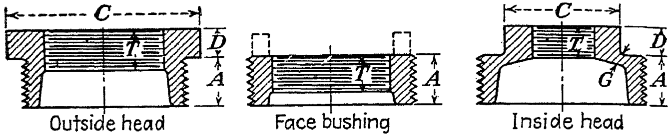
[1] Solid plugs are provided in sizes ⅛ to 3½ in., inclusive; cored plugs, ½ to 3½ in., inclusive.

[2] For sizes 4 in. and larger, slotted or bar pattern plugs are provided.

[3] These dimensions are the nominal size of wrench as given in Table 19, American Standard Wrench-Head Bolts and Nuts and Wrench Openings, ASA B18.2, see abstract, p. 550. Square head plugs are designed to fit these wrenches.

[4] Cored plugs have minimum metal thickness at all points, equal to dimension *G*, except at the end of the thread.

TABLE LXXIII.—DIMENSIONS OF OUTSIDE HEAD, INSIDE HEAD, AND FACE BUSHINGS*
(Table 4, ASA B16.14)
(All dimensions in inches)



Size[1,2,5]	Length of external thread, min A	Length of internal thread, min T	Height of head, min D	Width of head,[3] min C	Metal thickness,[4] min G
1/4 × 1/8	0.44	0.26†	0.14	0.64‡	
3/8 × 1/4	0.48	0.40†	0.16	0.68‡	
3/8 × 1/8	0.48	0.25	0.16	0.68‡	
1/2 × 3/8	0.56	0.41†	0.19	0.87‡	
1/2 × 1/4	0.56	0.32	0.19	0.87‡	
1/2 × 1/8	0.56	0.25	0.19	0.87‡	
3/4 × 1/2	0.63	0.53†	0.22	1.15	
3/4 × 3/8	0.63	0.36	0.22	1.15	
3/4 × 1/4	0.63	0.32	0.22	1.15	
1 × 3/4	0.75	0.50	0.25	1.42	
1 × 1/2	0.75	0.43	0.25	1.42	
1 × 3/8	0.75	0.36	0.30	1.12	
1 1/4 × 1	0.80	0.58	0.28	1.76	
1 1/4 × 3/4	0.80	0.50	0.28	1.76	
1 1/4 × 1/2	0.80	0.43	0.34	1.34	0.185
1 1/4 × 3/8	0.80	0.36	0.34	1.12	0.185
1 1/2 × 1 1/4	0.83	0.71†	0.31	2.00	
1 1/2 × 1	0.83	0.58	0.31	2.00	
1 1/2 × 3/4	0.83	0.50	0.37	1.63	0.200
1 1/2 × 1/2	0.83	0.43	0.37	1.34	0.200
2 × 1 1/2	0.88	0.70	0.34	2.48	
2 × 1 1/4	0.88	0.67	0.34	2.48	
2 × 1	0.88	0.58	0.41	1.95	0.220
2 × 3/4	0.88	0.50	0.41	1.63	0.220
2 1/2 × 2	1.07	0.75	0.37	2.98	
2 1/2 × 1 1/2	1.07	0.70	0.44	2.68	
2 1/2 × 1 1/4	1.07	0.67	0.44	2.39	0.240
2 1/2 × 1	1.07	0.58	0.44	1.95	0.240
3 × 2 1/2	1.13	0.92	0.40	3.86	
3 × 2	1.13	0.75	0.48	3.28	
3 × 1 1/2	1.13	0.70	0.48	2.68	0.260
3 × 1 1/4	1.13	0.67	0.48	2.39	0.260
3 1/2 × 3	1.18	0.98	0.43	4.62	
3 1/2 × 2 1/2	1.18	0.92	0.52	3.86	
3 1/2 × 2	1.18	0.75	0.52	3.28	0.280
3 1/2 × 1 1/2	1.18	0.70	0.52	2.68	0.280

TABLE LXXIII.—(*Concluded*)

Size[1,2,5]	Length of external thread, min *A*	Length of internal thread, min *T*	Height of head, min *D*	Width of head,[3] min *C*	Metal thickness,[4] min *G*
4 × 3½	1.22	1.03	0.50	5.20	
4 × 3	1.22	0.98	0.50	4.62	
4 × 2½	1.22	0.92	0.60	3.86	0.310
4 × 2	1.22	0.75	0.60	3.28	0.310
5 × 4	1.31	1.08	0.50	5.79	
5 × 3½	1.31	1.03	0.60	5.20	
5 × 3	1.31	0.98	0.60	4.62	0.380
5 × 2½	1.31	0.92	0.60	3.86	0.380
6 × 5	1.40	1.18	0.63	7.05	
6 × 4	1.40	1.08	0.75	5.79	0.430
6 × 3½	1.40	1.03	0.75	5.20	0.430
6 × 3	1.40	0.98	0.75	4.62	0.430
6 × 2½	1.40	0.92	0.75	3.86	0.430
8 × 6	1.57	1.28	0.83	8.28	
8 × 5	1.57	1.18	0.83	7.05	0.550
8 × 4	1.57	1.08	0.83	5.79	0.550

* Material to be cast iron, malleable iron, or steel. (See footnote 1.)

These bushings are threaded with American Standard tapered pipe threads (ASA B2.1. See abstract, p. 481).

The addition of lugs on face bushings shall not be prohibited.

Cored bushings have minimum metal thickness at all points, equal to dimension *G*, except at the end of the thread.

† To provide proper metal thickness these sizes shall not be cored out to diameters greater than the root diameter of the internal thread. The length of the internal thread may be equal to the minimum dimension, *T*, or greater up to the full length of bushing.

‡ When made of bar stock, the dimensions may be [illegible], [illegible], and [illegible] in., respectively, in order to use regular bar stock sizes.

[1] Hexagon head or octagon head bushings, sizes 2½ in. and smaller reducing one size, may be made either of malleable iron or steel. Other sizes may be made either of cast iron or malleable iron or steel. Face bushings, sizes 2½ in. and smaller, may be made either of malleable iron or steel. Face bushings, 3 in. and larger reducing one size, may be made either of malleable iron or steel. Face bushings, 3 in. and larger reducing two sizes or more, may be made either of cast iron or malleable iron or steel.

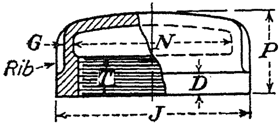
[2] The bushing of openings to provide greater reductions than those given in the table are made by using a bushing with the greater reduction obtainable, and reducing this again with a bushing giving the desired pipe sizes.

[3] Heads of bushings shall be hexagonal or octagonal except that on the larger sizes of outside head bushings the heads may be made round with lugs instead of hexagonal or octagonal.

[4] *G* same as metal thickness for 125-lb Cast-Iron Screwed Fittings (ASA B16d, see abstract, p. 567).

[5] The table of dimensions is for pattern sizes only and does not represent the list of reductions carried in regular stock.

TABLE LXXIV.—DIMENSIONS OF 150-LB MALLEABLE-IRON CAPS
(Table 6, ASA B16.14)
(All dimensions in inches)



Nominal pipe size	Length of thread, mi	Width of band,	Inside diameter of fittings N		Metal thickness of side[1]	Outside diameter of band, min	Height min	Thickness of ribs
	T	D	Min	Max	G	J	P	
½	0.43	0.249	0.840	0.897	0.105	1.197	0.87	0.105
¾	0.50	0.273	1.050	1.107	0.120	1.458	0.97	0.120
1	0.58	0.302	1.315	1.385	0.134	1.771	.16	0.134
1¼	0.67	0.341	1.660	1.730	0.145	2.153	.28	0.145
1½	0.70	0.368	1.900	1.970	0.155	2.427	.33	0.155
2	0.75	0.422	2.375	2.445	0.173	2.963	.45	0.173
2½	0.92	0.478	2.875	2.975	0.210	3.589	.70	0.210
3	0.98	0.548	3.500	3.600	0.231	4.285	.80	0.231
3½	1.03	0.604	4.000	4.100	0.248	4.843	.90	0.248
4	1.08	0.661	4.500	4.600	0.265	5.401	2.08	0.265
5	1.18	0.780	5.563	5.663	0.300	6.583	2.32	0.300
6	1.28	0.900	6.625	6.725	0.336	7.767	2.55	0.336

These caps are threaded with American Standard tapered pipe threads (ASA B2.1. See abstract, p. 481).

Reproduced from American Standard for Malleable-Iron Screwed Fittings, 150 Lb (ASA B16c. See abstract, p. 573).

Caps may be made flat or with the radius as shown in the illustration.

The outside radius of top is equal to $3 \times N$.

The use of ribs is optional with the manufacturer.

[1] Patterns shall be designed to produce castings of metal thicknesses given in the table. Metal thickness at no point shall be less than 90 per cent of the thickness given in the table.

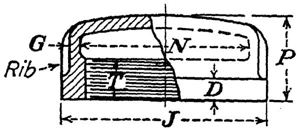
ards. These fittings are all threaded with American Standard Taper Pipe Threads (see abstract of ASA B2.1, page 481) and have a similar function in piping systems and other applications.

Plugs and bushings have no definite pressure ratings but are used with regular 125-lb cast-iron and 150-lb malleable-iron screwed fittings. For higher pressures solid plugs and face bushings are recommended. Malleable-iron caps are rated at 150 psi gage for saturated steam, and 300 psi gage for hydraulic pressure including shock at or near the ordinary range of air temperature. Cast-iron caps are rated at 125 psi gage for saturated steam, or 175 psi gage for hydraulic service.

These fittings are furnished in cast iron, malleable iron, or steel as indicated in the individual tables. The manufacturer shall be prepared to certify that his

product fulfills the requirements of ASTM Specifications A126 for cast iron and A197 for malleable iron, see abstracts on pages 625 and 573, respectively. Items made of carbon steel may be cast, forged, or machined from bar stock.

TABLE LXXV.—DIMENSIONS OF 125-LB CAST-IRON CAPS
(Table 7, ASA B16.14)
(All dimensions in inches)



Nominal pipe size	Length of thread, min *T*	Width of band, min *D*	Inside diameter of fitting, max *N*	Metal thickness[1] *G*	Outside diameter of band, min *J*	Height min *P*
2½	0.92	0.94	2.975	0.240	3.86	1.81
3	0.98	1.00	3.600	0.260	4.62	1.91
3½	1.03	1.06	4.100	0.280	5.20	2.03
4	1.08	1.12	4.600	0.310	5.79	2.22
5	1.18	1.18	5.663	0.380	7.05	2.38
6	1.28	1.28	6.725	0.430	8.28	2.63
8	1.47	1.47	8.725	0.550	10.63	2.88
10	1.68	1.68	10.850	0.690	13.12	3.50
12	1.88	1.88	12.850	0.800	15.47	3.88

These caps are threaded with American Standard tapered pipe threads, ASA B2.1, see abstract p. 481.

Reproduced from American Standard for Cast-Iron Screwed Fittings, 125 and 250 Lb (ASA, B16d). Abstracted on p. 567.

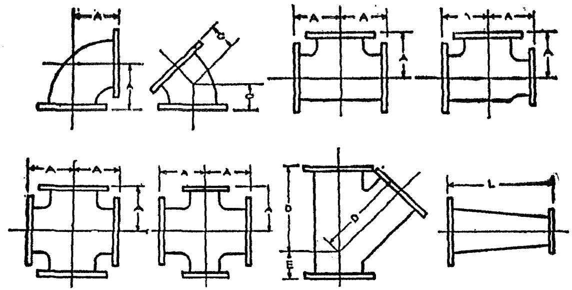
Caps may be made flat or with the radius as shown in the illustration.

The use of ribs is optional with the manufacturer.

[1] Patterns shall be designed to produce castings of metal thicknesses given in the tables. Metal thickness at no point shall be less than 90 per cent of thickness given in the table.

Flanges and Fittings for Lightweight Pipe.—Fittings for lightweight spiral-welded and spiral-riveted pipe have the same center-to-face dimensions as the 125-lb American Standard for Cast Iron. Fittings are furnished flanged to dimensions given in Table LXXVI, unless otherwise ordered. Fittings also are furnished plain end for welding or flanged to 125-lb American Standard diameter and drilling. Flanges are the same as the riveted pipe manufacturers' standard, except for minor changes in thickness and drilling. Hubs of flanges are designed for attaching by slipping over lightweight pipe and welding.

TABLE LXXVI.—FLANGES AND FITTINGS FOR LIGHTWEIGHT PIPE[1]
No. 10 Gage Thickness (0.1406 in.)



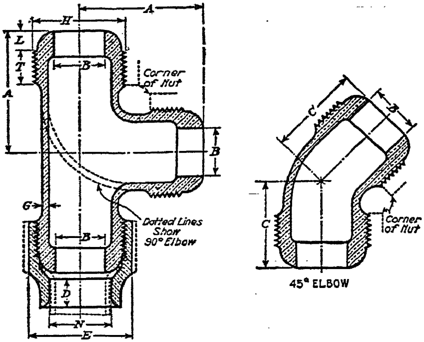
Nominal pipe size, inches	Flanges									Fittings					
	Outside-diameter flange	Thickness flange	Inside-diameter flange	Length of hub	Outside diameter of hub	Drilling template			Approx. weight each, pounds	Center-to-face short-radius ell	Center-to-face long-radius ell	Center to face	Center to face	Center to face	Face to face
						Number of bolts	Diameter and length of bolts	Diameter of bolt circle		A	A	C	D	E	L
3	6	3/8	3 1/4	3/4	3 9/16	4	1/2 × 1 1/2	4 3/4	2	5 1/2	7 3/4	3	10	3	6
4	7	3/8	4 1/4	3/4	4 11/16	8	1/2 × 1 1/2	5 15/16	3	6 1/2	9	4	12	3	7
5	8	3/8	5 1/4	3/4	5 3/4	8	1/2 × 1 1/2	6 15/16	4	7 1/2	10 1/4	4 1/2	13 1/2	3 1/2	8
6	9	1/2	6 1/4	7/8	6 3/4	8	1/2 × 1 3/4	7 7/8	6	8	11 1/2	5	14 1/2	3 1/2	9

TABLE LXXVI.—(*Concluded*)

Nominal pipe size, inches	Flanges: Outside-diameter flange *A*	Flanges: Thickness flange *T*	Flanges: Inside-diameter flange *B*	Flanges: Length of hub *L*	Flanges: Outside diameter of hub *E*	Flanges: Drilling template: Number of bolts	Flanges: Drilling template: Diameter and length of bolts	Flanges: Drilling template: Diameter of bolt circle	Flanges: Approx. weight each, pounds	Fittings: Center to face short-radius ell *A*	Fittings: Center to face long-radius ell *A*	Fittings: Center to face *C*	Fittings: Center to face *D*	Fittings: Center to face *E*	Fittings: Face to face *L*
8	11	½	8¼	⅞	8¾	8	½ × 1¾	10	7	9	14	5½	17½	4½	11
10	14	½	10 5/16	⅞	10⅞	12	⅝ × 2	12¼	12	11	16½	6½	20½	5	12
12	16	½	12 5/16	1	12⅞	12	⅝ × 2	14¼	14	12	19	7½	24½	5½	14
14	18	⅝	14 5/16	1⅛	14⅞	12	⅝ × 2¼	16¼	20	14	21½	7½	27	6	16
16	21¼	⅝	16 5/16	1⅛	17¼	16	⅝ × 2¼	19¼	31	15	24	8	30	6½	18
18	23¼	⅝	18⅜	1⅛	19¼	16	⅝ × 2¼	21¼	36	16½	26½	8½	32	7	19
20	25¼	⅝	20⅜	1⅛	21¼	20	⅝ × 2¼	23⅛	38	18	29	9½	35	8	20
22	28¼	⅝	22 7/16	1⅛	23¾	20	⅝ × 2¼	26	51	20	31½	10	37½	8½	22
24	30	¾	24 7/16	1⅜	25¾	20	⅝ × 2½	27¾	64	22	34	11	40½	9	24
26	32	¾	26 9/16	1⅜	27¾	24	¾ × 2½	29¾	66	23	36½	13	44	9	26
28	34	¾	28 9/16	1⅜	29¾	28	¾ × 2½	31¾	71	24	39	14	46½	9½	28
30	36	¾	30 9/16	1⅜	31¾	28	¾ × 2½	33¾	75	25	41½	15	49	10	30
32	38	¾	32 9/16	1⅜	33¾	28	¾ × 2½	35¾	80	26	44	16	....	....	32
34	40	¾	34 9/16	1⅜	35¾	32	¾ × 2½	37¾	85	27	46½	17	....	....	34
36	42	¾	36 9/16	1⅜	37¾	32	¾ × 2½	39¾	89	28	49	18	....	....	36
38	44	¾	38 9/16	1⅜	39¾	32	¾ × 2½	41¾	94	29	51½	19	....	....	38
40	46	¾	40 9/16	1⅜	41¾	36	¾ × 2½	43¾	99	30	54	20	....	....	40
42	48	¾	42 9/16	1⅜	43¾	36	¾ × 2½	45¾	103	31	56½	21	....	....	42
44	50	¾	44 9/16	1⅜	45¾	40	¾ × 2½	47¾	108						
46	52	¾	46 9/16	1⅜	47¾	40	¾ × 2½	49¾	112						
48	54	¾	48 9/16	1⅜	49¾	44	¾ × 2½	51¾	117						

[1] For data on spiral-welded pipe, see p. 353.

TABLE LXXVII.—GENERAL DIMENSIONS—BRASS FITTINGS FOR FLARED COPPER TUBES[1,2]
(Table 1, ASA A40.2)
(All dimensions in inches)



Size[5]	Outside diameter of tube	Center to face, minimum	Center to face of 45-deg elbow (minimum)	Maximum major[3,4] diameter of thread on fitting	Minimum minor[4] diameter of thread on nut	Length of thread	Length of seat projection	Metal thickness (min)	Diameter[5] of bore of nut	Length of bore in nut, (minimum)	Width across flats of nut	Number of flats on nut
B		A	C	H		T	L	G	N	D	E	
3/8	1/2	1.42	1.06	0.8750	0.7977	0.43	0.25	0.09	0.52	0.19	1.08	6
1/2	5/8	1.53	1.12	1.0000	0.9227	0.43	0.26	0.09	0.65	0.25	1.22	6
3/4	7/8	1.78	1.28	1.2500	1.1727	0.50	0.28	0.10	0.90	0.38	1.48	6
1	1 1/8	2.09	1.44	1.6250	1.5348	0.56	0.30	0.11	1.15	0.50	1.90	6
1 1/4	1 3/8	2.28	1.58	1.8750	1.7848	0.62	0.32	0.12	1.41	0.63	2.16	8
1 1/2	1 5/8	2.56	1.75	2.2500	2.1598	0.69	0.34	0.13	1.66	0.75	2.56	8
2	2 1/8	3.06	2.06	2.8750	2.7848	0.81	0.38	0.15	2.16	1.00	3.26	10

[1] These fittings are recommended for maximum cold-water working pressure of 175 psi gage.
[2] Nuts with 45-deg taper seat or convex-curved seat are interchangeable on ball joint fittings.
[3] Basic diameter.
[4] See Tables 2 and 3 of ASA A40.2 for detailed thread dimensions.
[5] Tolerances on diameter of bore of fitting and nut all sizes ±0.005.

American Standard for
BRASS FITTINGS FOR FLARED COPPER TUBES
ASA A40.2-1936

Abstracted[1]

The fittings covered by this standard are designed for use with copper water tubes corresponding to dimensions of ASTM B88 (see abstract, page 474). The

[1] For complete standard, reference may be made to ASA A40.2, (see note, p. 430).

fittings are suitable for a cold-water service pressure of 175 psi gage. The standard brass composition for these cast fittings is: 85 per cent copper, 5 per cent tin, 5 per cent lead, and 5 per cent zinc. The use of ASTM B62 Specification for material is recommended. Screw threads conform to ASA B1.1 for fine-thread series, with Class 2 fit on fitting and on nut. Several dimensions are given in Table LXXVII.

American Standard for SOLDERED-JOINT FITTINGS

ASA A40.3-1941

Abstracted[1]

This standard covers certain dimensions of soldered-joint wrought-metal and cast-brass fittings for use with copper water tubes corresponding to dimensions of ASTM B88 (see abstract, page 474). The laying length of cast-brass fittings (center-to-shoulder distance) is included, but because of the variety of forming methods, the laying length of wrought-metal fittings has not been standardized.

Fittings with this type of joint made with 50-50 tin-lead solder have the following pressure-temperature ratings. When authentic information on the performance of other types of solder is available, it is contemplated that higher pressure ratings will be issued.

Adjusted Pressure Ratings for Soldered Joints, ASA A40.3

Service temperature, F	Service pressure rating, psi gage		
	Water		Steam, all sizes
	2 in. and smaller	2½ in. and larger	
100	175	150	
150	125	100	
200	100	75	
250	75	50	15

Wrought fittings shall have a minimum copper content of 85 per cent. Cast-brass fittings shall conform to ASTM B62 and have the following composition: 85 per cent copper, 5 per cent tin, 5 per cent lead, and 5 per cent zinc. The minimum requirements for 50-50 tin-lead solder shall be in accordance with ASTM B32. Metal thickness tolerances and general dimensions of fittings are given in Table LXXVIII. For laying length dimensions of reducing elbows, tees, and crosses and couplings in both straight and reducing sizes, references should be made to ASA A40.3. Elbows and coupling adapters with male and female pipe thread ends also are available.

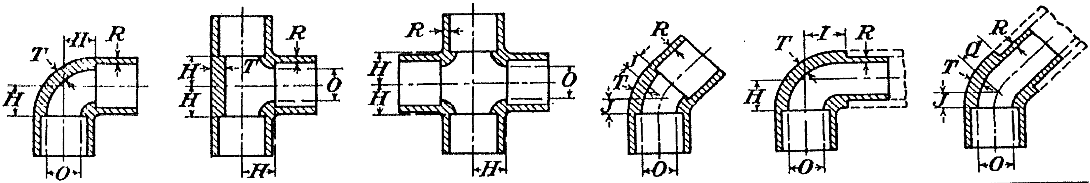
Note.—An estimate of the amount of solder required for 100 joints can be made on the basis of 1.25 to 1.50 lb of solder per inch of water tube diameter for

[1] For complete standard, reference may be made to ASA A40.3 (see note, p. 430).

TABLE LXXVIII.—SOLDERED-JOINT FITTINGS. DIMENSIONS OF TUBING AND FITTING ENDS AND ELBOWS, TEES, CROSSES, AND 45-DEG ELBOWS

(Tables 1 and 2, ASA A40.3-1941)

(All dimensions in inches)



Nominal size[1]	Outside diameter of tubing or male end of fittings[2]				Length of male end of fitting	Bore of fittings			Cast brass[4]							Wrought metal	Center to shoulder plus or minus
	Mean*	Max	Min	Tolerance,* plus or minus		Diameter Min	Diameter Max	Depth, min	Laying length, tee, ell, and cross[3]	Laying length, ell with external shoulder	Laying length, 45-deg ell	Laying length, 45-deg ell external shoulder	Inside diameter of fittings,[5] min	Metal thickness[6]		Metal thickness,[7] min[8]	
	A	B	C		K	F	E	G	H	I	J	Q	O	T	R	T and R	
1/4	0.375	0.376	0.374	0.001	7/16	0.378	0.380	5/16	1/4	3/8	3/16	1/4	0.31	0.08	0.048	0.030	0.04
3/8	0.500	0.501	0.499	0.001	1/2	0.503	0.505	3/8	5/16	7/16	3/16	5/16	0.43	0.08	0.048	0.035	0.05
1/2	0.625	0.626	0.624	0.001	11/16	0.628	0.630	1/2	7/16	9/16	3/16	5/16	0.54	0.09	0.054	0.040	0.06
3/4	0.875	0.876	0.874	0.001	15/16	0.878	0.880	3/4	9/16	11/16	1/4	3/8	0.78	0.10	0.060	0.045	0.06
1	1.125	1.1265	1.1235	0.0015	1 1/8	1.1285	1.1305	15/16	3/4	7/8	5/16	7/16	1.02	0.11	0.066	0.050	0.07
1 1/4	1.375	1.3765	1.3735	0.0015	1 3/16	1.3785	1.3805	1	7/8	1	7/16	9/16	1.26	0.12	0.072	0.055	0.07
1 1/2	1.625	1.627	1.623	0.002	1 5/16	1.629	1.6315	1 1/8	1	1 1/8	1/2	5/8	1.50	0.13	0.078	0.060	0.08
2	2.125	2.127	2.123	0.002	1 9/16	2.129	2.1315	1 3/8	1 1/4	1 3/8	9/16	3/4	1.98	0.15	0.090	0.070	0.08
2 1/2	2.625	2.627	2.623	0.002	1 3/4	2.629	2.6315	1 1/2	1 1/2	1 5/8	5/8	7/8	2.46	0.17	0.102	0.080	0.10
3	3.125	3.127	3.123	0.002	1 15/16	3.129	3.1515	1 11/16	1 3/4	1 7/8	3/4	1	2.94	0.19	0.114	0.090	0.10

TABLE LXXVIII.—(*Concluded*)

Nominal size[1]	Outside diameter of tubing or male end of fittings[2]				Length of male end of fitting	Bore of fittings			Cast brass[4]							Wrought metal	Center to shoulder, plus or minus
	Mean*	Max	Min	Tolerance,* plus or minus		Diameter Min	Diameter Max	Depth, min	Laying length, tee, ell, and cross[3]	Laying length, ell with external shoulder	Laying length, 45-deg ell	Laying length, 45-deg ell external shoulder	Inside diameter of fittings,[5] min	Metal thickness[6] T	Metal thickness[6] R	Metal thickness,[7] min[8] T and R	
	A	*B*	*C*		*K*	*F*	*E*	*G*	*H*	*I*	*J*	*Q*	*O*				
3½	3.625	3.627	3.623	0.002	2³⁄₁₆	3.629	3.632	1¹⁵⁄₁₆	2	2⅛	⅞	1⅛	3.42	0.20	0.120	0.100	0.10
4	4.125	4.127	4.123	0.002	2⁹⁄₁₆	4.129	4.132	2³⁄₁₆	2¼	2⅜	¹⁵⁄₁₆	1¼	3.90	0.22	0.132	0.110	0.12
5	5.125	5.127	5.123	0.002	3¹⁄₁₆	5.129	5.132	2¹¹⁄₁₆	3⅛	....	1⁷⁄₁₆	....	4.87	0.28	0.168	0.125	0.12
6	6.125	6.127	6.123	0.002	3½	6.129	6.132	3⅛	3⅝	....	1⅝	....	5.84	0.34	0.204	0.140	0.14

* Sizes ¼ to 6 in., inclusive, taken from American Standard Specifications for Copper Water Tube, ASA H23.1-1939 (ASTM B88, see abstract, p. 474).

[1] This size is the nominal bore of the tube.

[2] Dimensions in this table, applying to tubing diameter, are based on hard tubing. Soft tubing not conforming to these dimensions must be sized to conform.

[3] Dimensions for reducing elbows and reducing crosses are given in Table 3 and dimensions for reducing tees in Table 4 of ASA A40.3, not reproduced here.

[4] These dimensions may be used for wrought metal fittings as well as for cast brass fittings at manufacturer's option.

[5] This dimension is the same as the inside diameter Class L tubing of American Standard Specifications for Copper Water Tube, ASA H23.1 (ASTM B88).

[6] Patterns shall be designed to produce body thicknesses given in the table. Metal thickness at no point shall be less than 90 per cent of the thicknesses given in the table.

[7] This dimension has the same thickness as Class L tubing.

[8] These dimensions are minimum, but in every case the thickness of wrought fittings should be at least as heavy as the tubing with which it is to be used.

NOTE.—Wrought fittings, as well as cast fittings, must be provided with a shoulder or stop at the bottom end of socket.

sizes up to and including 3½ in., and 1.75 to 3 lb per in. for 4 to 6 in. inclusive. An allowance of 2.0 oz of flux per pound of solder is suggested by one manufacturer of soldered-joint fittings.

American Standard for 25-LB CAST-IRON PIPE FLANGES AND FLANGED FITTINGS

ASA B16 b2-1931

Abstracted[1]

This standard covers cast-iron pipe flanges and flanged fittings for maximum saturated steam-service pressure of 25 psi gage. Sizes 36 in. and smaller also may be used for maximum nonshock hydraulic working pressure of 43 lb gage or a maximum gas pressure of 25 lb gage, at or near the ordinary range of air temperatures.

All fittings conforming to this standard are marked to indicate the manufacturer and the pressure rating of 25 lb. The manufacturer is required to be in a position to certify that the chemical and physical properties, as proved by test specimens, are at least equal to the minimum requirements specified in ASTM Specification A126 for Class A regular gray iron (see abstract of A126 on page 625).

Dimensions.—The center-to-face and face-to-face dimensions, the flange diameters, bolt circles, and number of bolts are the same as for the 125-lb. American Standard (ASA B16a, abstracted on page 597), with a reduction in the thickness of flanges and bolt diameters, thereby maintaining interchangeability between the two standards.

Facing.—Flanges and flanged fittings shall be plain faced; *i.e.*, without projection or raised face. An inspection limit of $\pm \frac{1}{16}$ in. shall be allowed on all center-to-face dimensions and $\pm \frac{1}{8}$ in. on all face-to-face dimensions.

Bolting.—Drilling templates are in multiples of four, so that fittings may be made to face in any quarter. Bolt holes shall straddle the center line and are drilled ⅛ in. larger in diameter than the nominal size of the bolt. Bolts shall be of steel with square heads and hexagon nuts conforming to American Standard unfinished regular (or heavy) bolt heads and unfinished heavy nuts.

Spot Facing.—The bolt holes shall not be spot-faced for ordinary service. When required, the flanges and fittings in sizes 36 in. and larger may be spot-faced or back-faced to the minimum thickness of flange with a plus tolerance of ⅛ in.

Reducing Fittings.—Reducing elbows and side-outlet elbows carry same dimensions center to face as straight size elbows corresponding to the size of larger opening.

Tees, side-outlet tees, and crosses, 16 in. and smaller, reducing on the outlet or branch have the same dimensions center to face and face to face as straight-size fittings corresponding to size of the larger opening. Sizes 18 in. and larger, reducing on the outlet or branch, are made in two lengths depending on the size of the outlet as given in the tables of dimensions.

[1] For complete standard, reference may be made to ASA B16 b2 (see note, p. 430).

Tees and crosses, reducing on run only, have the same dimensions center to face and face to face as straight fittings corresponding to the size of the larger opening.

Reducing fittings listed in this standard shall be ordered by the designation of the outlets in their proper sequence, as indicated in the sketches on page 567.

Side-outlet elbows and side-outlet tees shall have all openings on intersecting center lines.

TABLE LXXIX.—25-LB CAST-IRON FLANGED FITTINGS
(Table 1, ASA B16 b2)
Templates for drilling
(All dimensions in inches)

Nominal pipe size	Diameter of flange	Minimum thickness of flange[1,2]	Diameter of bolt circle	Number of bolts[3]	Diameter of bolts	Diameter of bolt holes[4]	Length of bolts[5]	Total effective area, bolt metal	Stress, pounds per square inch, bolt metal[5,6]	Size of ring gasket
4	9	3/4	7 1/2	8	5/8	3/4	2 1/4	1.616	570	4 × 6 7/8
5	10	3/4	8 1/2	8	5/8	3/4	2 1/4	1.616	750	5 × 7 7/8
6	11	3/4	9 1/2	8	5/8	3/4	2 1/4	1.616	930	6 × 8 3/4
8	13 1/2	3/4	11 3/4	8	5/8	3/4	2 1/4	1.616	1470	8 × 11
10	16	7/8	14 1/4	12	5/8	3/4	2 1/2	2.424	1440	10 × 13 3/8
12	19	1	17	12	5/8	3/4	2 3/4	2.424	2195	12 × 16 1/8
14	21	1 1/8	18 3/4	12	3/4	7/8	3 1/4	3.62	1750	14 × 18
16	23 1/2	1 1/8	21 1/4	16	3/4	7/8	3 1/4	4.83	1710	16 × 20 1/2
18	25	1 1/4	22 3/4	16	3/4	7/8	3 1/2	4.83	1965	18 × 22
20	27 1/2	1 1/4	25	20	3/4	7/8	3 1/2	6.04	1920	20 × 24 1/4
24	32	1 3/8	29 1/2	20	3/4	7/8	3 3/4	6.04	2690	24 × 28 3/4
30	38 3/4	1 1/2	36	28	7/8	1	4 1/4	11.76	2030	30 × 35 1/8
36	46	1 5/8	42 3/4	32	7/8	1	5	13.44	2610	36 × 41 7/8
42	53	1 3/4	49 1/2	36	1	1 1/8	5 1/4	19.80	2315	42 × 48 1/2
48	59 1/2	2	56	44	1	1 1/8	5 1/2	24.20	2475	48 × 55
54	66 1/4	2 1/4	62 3/4	44	1	1 1/8	5 3/4	24.20	3195	54 × 61 3/4
60	73	2 1/4	69 1/4	52	1 1/8	1 1/4	6	36.02	2515	60 × 68 1/8
72	86 1/2	2 1/2	82 1/2	60	1 1/8	1 1/4	6 1/4	41.57	3120	72 × 81 3/8
84	99 3/4	2 3/4	95 1/2	64	1 1/4	1 3/8	7 1/4	57.14	3005	84 × 94 1/4
96	113 1/4	3	108 1/2	68	1 1/4	1 3/8	7 3/4	60.57	3705	96 × 107 1/4

[1] All 25-lb cast-iron standard flanges have plain faces.

[2] Screwed companion flanges should not be thinner than the 125-lb American Standard thickness on sizes 24 in. and smaller. Other types of flanges may have thicknesses as given in table above.

[3] Drilling templates are in multiples of four, so that fittings may be made to face in any quarter, and bolt holes straddle the center line. Bolt holes are drilled 1/8 in. larger in diameter than the nominal diameter of the bolt.

[4] The bolt holes on cast-iron flanged fittings are not spot-faced for ordinary service. When required, the fittings and flanges in sizes 36 in. and larger may be spot-faced or back-faced to minimum thickness of flange with a plus tolerance of 1/8 in., so that standard length bolts can be used.

[5] Bolts shall be steel with square heads and hexagonal nuts conforming to American Standard Unfinished Regular (or Heavy) Bolt Heads and Unfinished Heavy Nuts (see p. 550).

[6] The stress shown is that of internal pressure only, assumed to act on a circular area equal in diameter to the outside diameter of a ring gasket covering the flange to the inside of bolts.

Elbows.—Special degree elbows, ranging from 1 to 45 deg inclusive, shall have the same center-to-face dimension as given for 45-deg elbows, and those over 45 deg and up to 90 deg, inclusive, shall have the same center-to-face dimensions as given for 90-deg elbows. The angle designation of an elbow is its deflection from straight line flow and is the angle between the flange faces.

Where long-radius fittings are specified it has reference only to elbows, which are made in two center-to-face dimensions and known as elbows and long-radius elbows, the latter being used only when so specified.

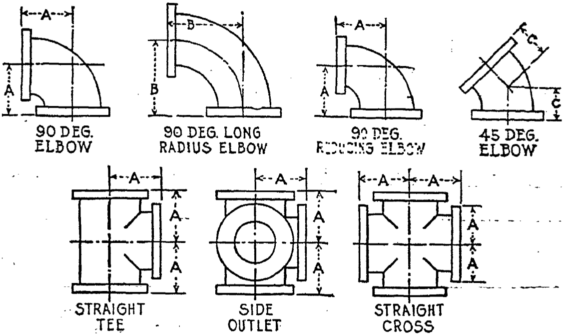
Screwed Companion Flanges.—Screwed companion flanges shall not be thinner than the 125-lb. American Standard thickness on sizes 24 in. and smaller. Other types of flanges may have thicknesses as in the Table LXXIX.

TABLE LXXX.—25-LB CAST-IRON FLANGED FITTINGS
(Table 5, ASA B16 b2)
Theoretical weights in pounds of elbows, tees, and crosses[1]

Nominal pipe size	90-deg elbow	45-deg elbow	90-deg long-radius elbow	Tees	Crosses
4	35	30	45	50	65
5					
6	55	50	65	85	100
8	80	65	105	120	150
10	135	100	160	185	225
12	185	160	245	270	325
14	250	195	330	370	450
16	340	240	425	450	550
18	385	285	530	550	670
20	465	350	665	660	780
24	695	490	950	960	1130
30	1050	840	1550	1500	1750
36	1620	1350	2480	2275	2600
42	2325	2000	3620	3200	3675
48	3205	2850	5300	4300	4880
54	4565	3970	7500	6250	6880
60	6000	5140	9675	8000	10250
72	9320	7525	14175	12150	13450

[1] All weights listed are for fittings faced and drilled, based upon minimum thicknesses and dimensions given in preceding table without allowances for variation. Cast iron is considered to weigh 0.26 lb per cu in.

TABLE LXXXI.—25-LB CAST-IRON FLANGED FITTINGS
Dimensions of elbows, tees, and crosses[1]
(Tables 2 and 3, ASA B16 b2)
(All dimensions in inches)

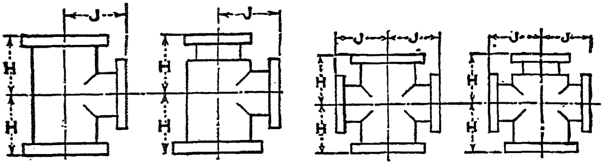


Nominal pipe size[1,2]	Center to face elbow[3,4,5] A	Center to face, long-radius elbow B	Center to face 45-deg elbow C	Center to face tee and crosses[3,6,7] A	Face to face tee and[1] crosses[2,3,6,7] AA	Diameter of flange	Thickness of flange,[8] minimum	Metal thickness of body, minimum
4	6½	9	4	6½	13	9	¾	0.42
5	7½	10¼	4½	7½	15	10	¾	0.44
6	8	11½	5	8	16	11	¾	0.4[illegible]
8	9	14	5½	9	18	13½	¾	0.4[illegible]
10	11	16½	6½	11	22	16	⅞	0.5[illegible]
12	12	19	7½	12	24	19	1	0.54
14	14	21½	7½	14	28	21	1⅛	0.57
16	15	24	8	15	30	23½	1⅛	0.60
18	16½	26½	8½	16½	33	25	1¼	0.64
20	18	29	9½	18	36	27½	1¼	0.67
24	22	34	11	22	44	32	1⅜	0.76
30	25	41½	15	25	50	38¾	1½	0.88
36	28	49	18	28	56	46	1⅝	0.99
42	31	56½	21	31	62	53	1¾	1.10
48	34	64	24	34	68	59½	2	1.26
54	39	71½	27	39	78	66¼	2¼	1.35
60	44	79	30	44	88	73	2¼	1.39
72	53	94	36	53	106	86½	2½	1.62

[1] Size of all fittings listed indicates nominal inside diameter of port.
[2] The flange diameters, bolt circles, and number of bolts are the same as the 125-lb American Standard (ASA B16...) with a reduction in the thickness of flanges and the bolt diameters as shown in Table LXXIX, thereby maintaining interchangeability with the 125-lb American Standard

[3] The center-to-face and face-to-face dimensions of fittings are the same as the 125-lb American Standard cast-iron flanged fittings.
[4] Special-degree elbows, ranging from 1 to 45 deg, inclusive, shall have the same center-to-face dimensions as 45-deg elbows, and those over 45 deg and up to 90 deg inclusive shall have the same

TABLE LXXXII.—25-LB CAST-IRON FLANGED FITTINGS
Dimensions of reducing tees and reducing crosses
(short-body patterns)[1,3,4,5]
(Table 4, ASA B16 b2)
(All dimensions in inches)



Size[1]	Size of outlet and smaller[2]	Center-to-face run *H*	Face-to-face run *HH*	Center-to-face outlet *J*
	All reducing fittings sizes 16 in. and smaller have same center-to-face dimensions as straight size fittings			
18	12	13	26	15½
20	14	14	28	17
24	16	15	30	19
30	20	18	36	23
36	24	20	40	26
42	24	23	46	30
48	30	26	52	34
54	36	29	58	37
60	40	33	66	41
72	48	40	80	48

[1] Short-body patterns are used for sizes 18 in. and larger.
[2] Long-body patterns are used when outlets are larger than given in above table, and, therefore have the same dimensions as straight-size fittings.
[3] Fittings reducing on the run only carry same dimensions center to face and face to face as straight-size fittings corresponding to size of the larger opening. Tees increasing on outlet, known as bull-head tees, will have same center-to-face and face-to-face dimensions as a straight fitting of the size of the outlet. For example: a 12 × 12 × 18 in. tee will be governed by the dimensions of the 18-in. long body tee, given in Table LXXXI; namely 16½ in. center to face of all openings and 33 in. face to face.
[4] In a side-outlet tee, the larger of the two side outlets governs the center-to-face dimension *J*.
[5] Side-outlet tees, with outlet at 90 deg or any other angle, straight or reducing, carry same dimensions center to face and face to face as regular tees having same reductions.

---

center-to-face dimensions as given for 90-deg elbows. The angle designation of an elbow is its deflection from straight-line flow and is the angle between the flange faces.
[5] Side-outlet elbows and side-outlet tees shall have all openings on intersecting center lines.
[6] Tees, side-outlet tees, and crosses, 16 in. and smaller, reducing on the outlet, have the same dimensions center to face and face to face as straight-size fittings, corresponding to the size of the larger opening. Sizes 18 in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet.
[7] Tees and crosses, reducing on run only, carry same dimensions center to face and face to face as a straight-size fitting of the larger opening.
[8] Screwed companion flanges should not be thinner than the 125-lb American Standard thickness on sizes 24 in. and smaller. Other types of flanges may have thicknesses as given in this table.

American Standard for
CAST-IRON PIPE FLANGES AND FLANGED FITTINGS, CLASS 125
ASA B16 a-1939[1] and B16 a.1-1943[2]
Abstracted

This standard has been revised to bring it into line with the practice followed in the manufacture and use of these flanges and fittings. Corrections have been made in wall thicknesses in sizes 3 in. and smaller to conform with actual practice. A wall-thickness tolerance of 87½ per cent of the dimensions given in the tables has been included to cover inspection requirements. The steam-service pressure ratings have been adjusted according to size to give more conservative stress conditions.

Pressure Ratings.[2]—The steam-service pressures at saturated-steam temperatures for the various sizes are as follows:

Sizes 1 to 12 in., inclusive........................ 125 lb SSP
Sizes 14 to 24 in., inclusive........................ 100 lb SSP
SSP = steam-service pressure

Water-service pressures at or near the ordinary range of air temperatures are:

Sizes 1 to 12 in., inclusive.......... 175 lb WWP
Sizes 14 to 48 in., inclusive.......... 150 lb WWP (flanges only)
WWP = water working pressure

NOTE.—Further research work is being conducted by ASA Sectional Committee A21 to determine water-service pressure ratings for fittings 14 in. and larger.

Marking.—All fittings shall have marks cast on them indicating the manufacturer and the steam-service pressure ratings given above in accordance with MSS Standard Practice SP-25.

Material.—Flanges and fittings 12 in. and smaller shall be made of material equal to the requirements of ASTM Specification A126, Class A, regular gray iron. Flanges and fittings 14 in. and larger shall be made of material equal to Class B, higher strength gray iron, of same specification (see abstract of ASTM Specification A126, page 625).

Wall-thickness Tolerance and Test Pressure.—The wall thickness of the castings at no point shall be less than 87½ per cent of the dimensions given in the tables. Fittings shall be designed to withstand a hydrostatic test pressure of twice the rated steam-service pressures. Unless specified by the purchaser, hydrostatic tests of cast-iron fittings covered by this standard are not required.

Facing.—These cast-iron flanges and flanged fittings shall be plain-faced, *i.e.*, without projection or raised face.

An inspection limit of ± 1/32 in. shall be allowed on all center-to-contact-surface dimensions for sizes up to and including 10 in. and ± 1/16 in. on sizes larger than 10 in. An inspection limit of ± 1/16 in. shall be allowed on all contact-surface-to-contact-surface dimensions for sizes up to and including 10 in. and ± ⅛ in. on sizes larger than 10 in.

[1] Copies of ASA B16a may be obtained from the American Standards Association, 29 West 39th St., New York 18, N.Y.

[2] American War Standard, approved Apr. 15, 1943.

Bolting.—Drilling templates are in multiples of four so that fittings may be made to face in any quarter.

Bolt holes shall straddle the center line. For bolts smaller than 1¾ in., the bolt holes shall be drilled ⅛ in. larger in diameter than the nominal size of the bolt. Holes for bolts 1¾ in. and larger shall be drilled ¼ in. larger than nominal diameter of bolts.

Bolts shall be of steel with American Standard Regular (or Heavy) Unfinished Square Heads and the nuts shall be of steel with American Standard Heavy Unfinished Hexagonal dimensions, all as specified in American Standard for Wrench Head Bolts and Nuts and Wrench Openings (ASA B18.2), abstracted on page 550. For bolts 1¾ in. in diameter and larger, bolt-studs with a nut on each end are recommended.

Hexagonal nuts for pipe sizes 1 to 48 in. can be conveniently pulled up with open wrenches of minimum design of heads. Hexagonal nuts for pipe sizes 48 to 96 in. can be conveniently pulled up with box wrenches.

All bolts, or bolt-studs if used, and all nuts shall be threaded in accordance with American Standard for Screw Threads (ASA B1.1) Coarse-thread Series, Class 2 Fit (abstracted on page 546).

Spot-facing Flanges.—The bolt holes of these cast-iron flanges need not be spot-faced for ordinary service except as follows: In sizes 12 in. and smaller when rough flanges, after facing, are oversize more than ⅛ in. in thickness, they shall be spot-faced to the specified thickness of flange (minimum) with a plus tolerance of ¹⁄₁₆ in. In sizes 14 to 24 in., inclusive, when rough flanges, after facing, are oversize more than ³⁄₁₆ in. in thickness, they shall be spot-faced to the specified thickness of flange (minimum) with a plus tolerance of ¹⁄₁₆ in. In sizes 30 in. and larger when rough flanges, after facing, are oversize more than ¼ in. in thickness, they shall be spot-faced to the specified thickness of flange (minimum) with a plus tolerance of ⅛ in.

Fittings.—The bolt holes of the flanges on these cast-iron fittings need not be spot-faced on sizes smaller than 18 in. for ordinary service, except as required for oversize thickness of flanges as indicated above. The bolt holes of *all* flanges on fittings 18 to 24 in., inclusive, shall be spot-faced to the specified thickness of the flange (minimum) with a plus tolerance of ¹⁄₁₆ in., and of *all* flanges on fittings sizes 30 to 48 in., inclusive, they shall be spot-faced to the specified thickness of the flange (minimum) with a plus tolerance of ⅛ in.

Reduced Fittings.—Reducing elbows and side-outlet elbows carry same dimensions center to face as straight-size elbows corresponding to the size of the larger opening.

Tees, side-outlet tees, crosses, and laterals, sizes 16 in. and smaller, reducing on the outlet or branch, have the same dimension, center to face and face to face as straight-size fittings corresponding to the size of the larger opening. Sizes 18 in. and larger, reducing on the outlet or branch, are made in two lengths depending on the size of the outlet as given in the tables of dimensions.

Tees, crosses, and laterals, reducing on the run only, have the same dimensions center to face and face to face as straight-size fittings corresponding to the size of the larger opening. On side-outlet tees having two different size reductions on the outlets, the larger of the two side outlets governs the center-to-face dimensions. Side outlet crosses are determined in the same manner as side outlet tees.

Tees increasing on the outlet are generally known as Bull Head Tees and have the same center-to-face and face-to-face dimensions as straight fittings the size of the outlet.

Reducers and eccentric reducers for all reductions have the same face-to-face dimensions as given in Table LXXXVIII for the larger opening. Reducing fittings listed in this standard shall be ordered by the designation of the outlets in their proper sequence as indicated in the sketches in Fig. 38.

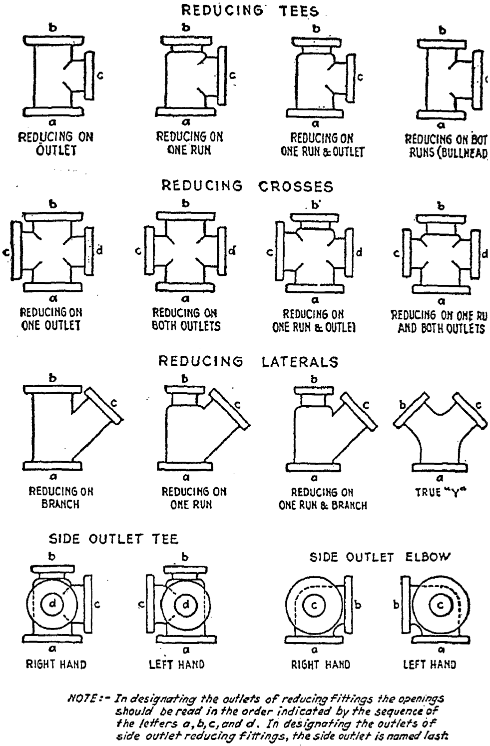
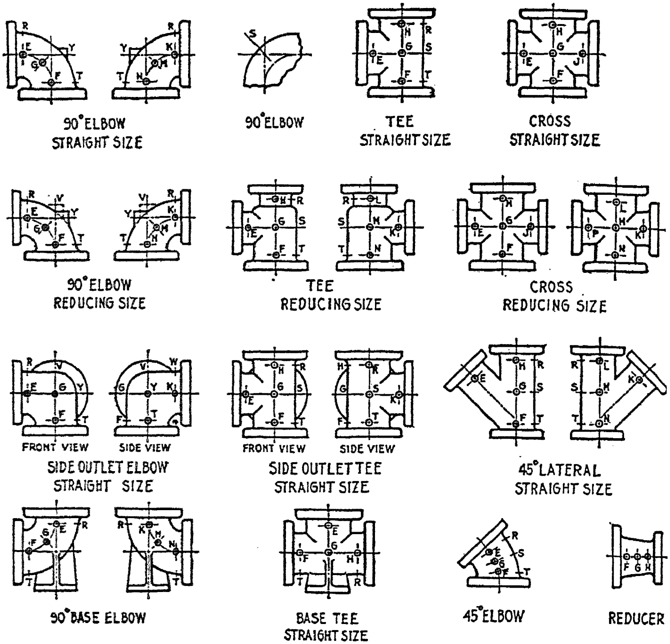


Fig. 38.—Method of designating outlets of reducing fittings when ordering.

Double-branch elbows whether straight or reducing have the same dimension center to face as straight-size elbows corresponding to the size of the larger opening.

Side-outlet elbows and side-outlet tees shall have all openings on intersecting center lines.

Elbows.—Special degree elbows ranging from 1 to 45 deg, inclusive, have the same center-to-face dimension given for 45-deg elbows and those over 45 deg and up to 90 deg, inclusive, shall have the same center-to-face dimensions given for 90-deg elbows. The angle designation of an elbow is its deflection from straight-line flow and is the angle between the flange faces.



NOTE:— In the above sketch all of the pairs of fittings, except the side outlet elbow and side outlet tee (straight sizes), represent fittings with symmetrical shapes. These latter represent two views of the same fitting.

FIG. 39.—Method of locating drain tappings.

Crosses.—Crosses both straight and reducing sizes 18 in. and larger shall be reinforced to compensate for the inherent weakness in the casting design.

True Y's.—The dimensions of true Y's, straight sizes, are given in Table LXXXVIII. Other forms are considered special and should be made to suit conditions.

Laterals.—Laterals (Y-branches) both straight and reducing sizes 8 in. and larger shall be reinforced to compensate for the inherent weakness in the casting design.

Drain Taps.—The maximum size of hole that can be tapped in the wall of the fitting without adding a boss is shown in the following table:

Maximum Size of Tapped Hole in Fitting without Adding Bosses

Size of Fitting (Inches).......	2 to 3	4 to 5			14 to 24
Size of Tapped Hole (Inches).			1¼	1½	

When bosses are required, the method of designating the locations of the tapped holes for drains is shown in Fig. 39. Each possible location is designated by a letter so that desired locations for the various types of fittings may be definitely specified without the use of further sketches or description.

TABLE LXXXIII.—AMERICAN STANDARD CLASS 125 CAST-IRON FLANGED FITTINGS

Templates for drilling[1]

(Table 2, ASA B16 a)

(All dimensions in inches)

Nominal pipe size	Diameter of flange	Thickness of flange (minimum)	Diameter of bolt circle	Number of bolts	Diameter of bolts	Diameter of drilled bolt holes[1]	Length of bolts	Length of bolt stud with two nuts	Size of ring gasket
1	4¼	7/16	3⅛	4	½	⅝	1¾	....	1 × 2⅝
1¼	4⅝	½	3½	4	½	⅝	2	....	1¼ × 3
1½	5	9/16	3⅞	4	½	⅝	2	....	1½ × 3⅜
2	6	⅝	4¾	4	⅝	¾	2¼	....	2 × 4⅛
2½	7	11/16	5½	4	⅝	¾	2½	....	2½ × 4⅞
3	7½	¾	6	4	⅝	¾	2½	....	3 × 5⅜
3½	8½	13/16	7	8	⅝	¾	2¾	....	3½ × 6⅜
4	9	15/16	7½	8	⅝	¾	3	....	4 × 6⅞
5	10	15/16	8½	8	¾	⅞	3	....	5 × 7¾
6	11	1	9½	8	¾	⅞	3¼	....	6 × 8¾
8	13½	1⅛	11¾	8	¾	⅞	3½	....	8 × 11
10	16	1 3/16	14¼	12	⅞	1	3¾	....	10 × 13⅜
12	19	1¼	17	12	⅞	1	3¾	....	12 × 16⅛
14 O.D.	21	1⅜	18¾	12	1	1⅛	4¼	....	14 × 17¾
16 O.D.	23½	1 7/16	21¼	16	1	1⅛	4½	....	16 × 20¼
18 O.D.	25	1 9/16	22¾	16	1⅛	1¼	4¾	....	18 × 21⅝
20 O.D.	27½	1 11/16	25	20	1⅛	1¼	5	....	20 × 23⅞
24 O.D.	32	1⅞	29½	20	1¼	1⅜	5½	....	24 × 28¼
30 O.D.	38¾	2⅛	36	28	1¼	1⅜	6¼	....	30 × 34¾
36 O.D.	46	2⅜	42¾	32	1½	1⅝	7	....	36 × 41¼
42 O.D.	53	2⅝	49½	36	1½	1⅝	7½	....	42 × 48
48 O.D.	59½	2¾	56	44	1½	1⅝	7¾	....	48 × 54½
54[2] O.D.	66¼	3	62¾	44	1¾	2	8½	10½	54 × 61
60[2] O.D.	73	3⅛	69¼	52	1¾	2	8¾	10¾	60 × 67½
72[2] O.D.	86½	3½	82½	60	1¾	2	9½	11½	72 × 80¾
84[2] O.D.	99¾	3⅞	95½	64	2	2¼	10½	12¾	84 × 93½
96[2] O.D.	113¼	4¼	108½	68	2¼	2½	11½	14	96 × 106¼

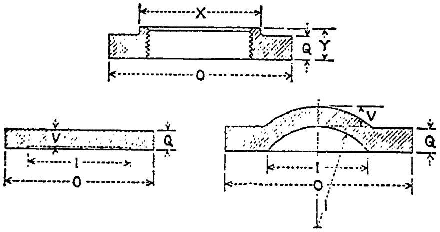
[1] Drilling templates are in multiples of four so that fittings may be made to face in any quarter, and bolt holes straddle the center line. For bolts smaller than 1¾ in. the bolt holes shall be drilled ⅛ in. larger in diameter than the nominal diameter of the bolt. Holes for bolts 1¾ in. and larger shall be drilled ¼ in. larger than nominal diameter of bolts. All Class 125 flanges have plain faces.

[2] These sizes are included for convenience where special fittings with larger flanges are required and do not necessarily carry a definite rating.

NOTE.—For further explanation, see text.

TABLE LXXXIV.—AMERICAN STANDARD CLASS 125 CAST-IRON FLANGED FITTINGS.

Dimensions of screwed companion and blind flanges[1,2]
(Table 3, ASA B16 a)
(All dimensions in inches)



10 IN. AND SMALLER 12 IN. AND LARGER

Nominal[3] pipe size	Diameter of flange	Thickness of flange, (minimum)	Wall[3] thickness	Diameter of hub, (minimum)	Length[4] of hub and threads, (minimum)
I	*O*	*Q*	*V*	*X*	*Y*
1	4¼	7/16	3/8	1 15/16	1 1/16
1¼	4 5/8	1/2	7/16	2 5/16	1 3/16
1½	5	9/16	1/2	2 9/16	7/8
2	6	5/8	9/16	3 1/16	1
2½	7	11/16	5/8	3 9/16	1 1/8
3	7½	3/4	11/16	4¼	1 3/16
3½	8½	13/16	3/4	4 13/16	1¼
4	9	15/16	7/8	5 5/16	1 5/16
5	10	15/16	7/8	6 7/16	1 7/16
6	11	1	15/16	7 9/16	1 9/16
8	13½	1 1/8	1 1/16	9 11/16	1¾
10	16	1 3/16	1 1/8	11 15/16	1 15/16
12	19	1¼	13/16	14 1/16	2 3/16
14 O.D.	21	1 3/8	7/8	15 3/8	2¼
16 O.D.	23½	1 7/16	1	17½	2½
18 O.D.	25	1 9/16	1 1/16	19 5/8	2 11/16
20 O.D.	27½	1 11/16	1 1/8	21¾	2 7/8
24 O.D.	32	1 7/8	1¼	25	3¼
30 O.D.	38¾	2 1/8	1 7/16		
36 O.D.	46	2 3/8	1 5/8		
42 O.D.	53	2 5/8	1 13/16		
48 O.D.	59½	2¾	2		

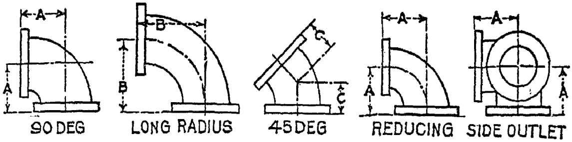
[1] For drilling templates refer to Table LXXXIII.
[2] All of these standard flanges have plain faces.
[3] All blind flanges for sizes 12 in. (19 in. O.D.) and larger must be dished with inside radius equal to the port diameter. The wall thickness at no point shall be less than 87½ per cent of the dimensions given in the table.
[4] For maximum service conditions, steel companion flanges in accordance with ASA-B-16 e are recommended.

Table LXXXV.—American Standard Class 125 Cast-iron Flanged Fittings

Dimensions of elbows[1]

(Table 4, ASA B16 a)

(All dimensions in inches)



Nominal pipe size	Inside diameter of fittings	Center[2,3,4] to face elbow *A*	Center to face, long-radius elbow[2,3,4] *B*	Center to face, 45 deg elbow *C*	Diameter of flange	Thickness of flange, (minimum)	Wall[5] thickness
1	1	3½	5	1¾	4¼	7/16	5/16
1¼	1¼	3¾	5½	2	4⅝	½	5/16
1½	1½	4	6	2¼	5	9/16	5/16
2	2	4½	6½	2½	6	⅝	5/16
2½	2½	5	7	3	7	11/16	5/16
3	3	5½	7¾	3	7½	¾	⅜
3½	3½	6	8½	3½	8½	13/16	7/16
4	4	6½	9	4	9	15/16	½
5	5	7½	10¼	4½	10	15/16	½
6	6	8	11½	5	11	1	9/16
8	8	9	14	5½	13½	1⅛	⅝
10	10	11	16½	6½	16	1 3/16	¾
12	12	12	19	7½	19	1¼	13/16
14 O.D.	14	14	21½	7½	21	1⅜	⅞
16 O.D.	16	15	24	8	23½	1 7/16	1
18 O.D.	18	16½	26½	8½	25	1 9/16	1 1/16
20 O.D.	20	18	29	9½	27½	1 11/16	1⅛
24 O.D.	24	22	34	11	32	1⅞	1¼
30 O.D.	30	25	41½	15	38¾	2⅛	1 7/16
36 O.D.	36	28	49	18	46	2⅜	1⅝
42 O.D.	42	31	56½	21	53	2⅝	1 13/16
48 O.D.	48	34	64	24	59½	2¾	2

[1] For drilling templates refer to Table LXXXIII.

[2] Reducing elbows and side-outlet elbows carry same dimensions center to face as straight-size elbows, corresponding to the size of the larger opening.

[3] Special-degree elbows, ranging from 1 to 45 deg, inclusive, have the same center-to-face dimensions given for 45-deg elbows, and those over 45 deg and up to 90 deg, inclusive, shall have the same center-to-face dimensions given for 90-deg elbows. The angle designation of an elbow is its deflection from straight-line flow and is the angle between the flange faces.

[4] Side-outlet elbows shall have all openings on intersecting center lines.

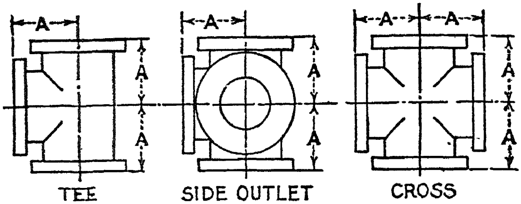
[5] Body thickness at no point shall be less than 87½ per cent of the dimensions given in the table.

TABLE LXXXVI.—AMERICAN STANDARD CLASS 125 CAST-IRON FLANGED FITTINGS

Dimensions of tees and crosses (straight sizes)[1,2]

(Table 5, ASA B16 a)

(All dimensions in inches)



Nominal[3] pipe size	Inside diameter of fittings	Center[3,4] to face tees and crosses A	Face[3,4] to face tees and crosses AA	Diameter of flange	Thickness of flange, minimum	Wall[5] thickness
1	1	3½	7	4¼	7/16	5/16
1¼	1¼	3¾	7½	4⅝	½	5/16
1½	1½	4	8	5	9/16	5/16
2	2	4½	9	6	⅝	5/16
2½	2½	5	10	7	11/16	5/16
3	3	5½	11	7½	¾	⅜
3½	3½	6	12	8½	13/16	7/16
4	4	6½	13	9	15/16	½
5	5	7½	15	10	15/16	½
6	6	8	16	11	1	9/16
8	8	9	18	13½	1⅛	⅝
10	10	11	22	16	1 3/16	¾
12	12	12	24	19	1¼	13/16
14 O.D.	14	14	28	21	1⅜	⅞
16 O.D.	16	15	30	23½	1 7/16	1
18 O.D.	18	16½	33	25	1 9/16	1 1/16
20 O.D.	20	18	36	27½	1 11/16	1⅛
24 O.D.	24	22	44	32	1⅞	1¼
30 O.D.	30	25	50	38¾	2⅛	1 7/16
36 O.D.	36	28	56	46	2⅜	1⅝
42 O.D.	42	31	62	53	2⅝	1 13/16
48 O.D.	48	34	68	59½	2¾	2

[1] For drilling templates refer to Table LXXXIII.

[2] Crosses, both straight and reducing sizes, 18 in. and larger shall be reinforced to compensate for the inherent weakness in the casting system.

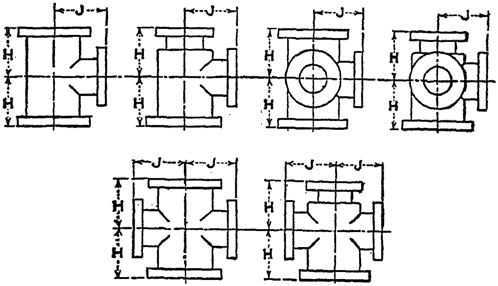
[3] Tees and crosses, reducing on run only, carry same dimensions center to face and face to face as a straight size fitting of the larger opening.

[4] Tees, side-outlet tees, and crosses, 16 in. and smaller, reducing on the outlet, have the same dimensions center to face and face to face as straight-size fittings, corresponding to the size of the larger opening. Sizes 18 in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet. For dimensions of short-body patterns, see Table LXXXVII.

[5] Body thickness at no point shall be less than 87½ per cent of the dimensions given in the table.

TABLE LXXXVII.—AMERICAN STANDARD CLASS 125 CAST-IRON FLANGED FITTINGS

Dimensions of reducing tees and reducing crosses[1,2,3,4,6,7]
(short-body patterns)
(Table 6, ASA B16 a)
(All dimensions in inches)



Nominal pipe size[1]	Size of outlet and smaller[2]	Center to face run *H*	Face to face run *HH*	Center to face outlet[5] *J*	Nominal pipe size[1]	Size of outlet and smaller[4]	Center to face run *H*	Face to face run *HH*	Center to face outlet[5] *J*
1	1	3½	7	3½	8	8	9	18	9
1¼	1¼	3¾	7½	3¾	10	10	11	22	11
1½	1½	4	8	4	12	12	12	24	12
2	2	4½	9	4½	14 O.D.	14	14	28	14
2½	2½	5	10	5	16 O.D.	16	15	30	15
3	3	5½	11	5½	18 O.D.	12	13	26	15½
3½	3½	6	12	6	20 O.D.	14	14	28	17
4	4	6½	13	6½	24 O.D.	16	15	30	19
5	5	7½	15	7½	30 O.D.	20	18	36	23
6	6	8	16	8	36 O.D.	24	20	40	26

[1] Short-body patterns are used for sizes 18 in. and larger.

[2] Fittings reducing on the run only carry same dimensions center to face and face to face as straight-size fittings corresponding to size of the larger opening. Tees increasing on outlet, known as "bull head tees," will have same center-to-face and face-to-face dimensions as a straight fitting of the size of the outlet. For example: A 12 × 12 × 18 in. tee will be governed by the dimensions of the 18-in. long-body tee given in Table LXXXVI; *viz.*, 16½ in. center to face of all openings and 33 in. face to face.

[3] Side-outlet tees, with outlet at 90 deg or any other angle, straight or reducing, carry same dimensions center to face and face to face as regular tees having same reductions. On side-outlet tees having two different size reductions on the outlets, the larger of the two side outlets governs the center-to-face dimensions.

[4] Long-body patterns are used when outlets are larger than given in the above table and, therefore, have the same dimensions as straight-size fittings.

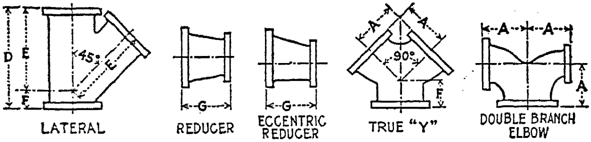
[5] Side-outlet crosses are determined in the same manner as side-outlet tees.

[6] Crosses, both straight and reducing, sizes 18 in. and larger, shall be reinforced to compensate for the inherent weakness in the casting design.

[7] For flange dimensions, wall thicknesses, and port diameters, see Table LXXXVI.

TABLE LXXXVIII.—AMERICAN STANDARD CLASS 125 CAST-IRON FLANGED FITTINGS

Dimensions of laterals, reducers, true Y's, double-branch elbows (straight sizes)[1]
(Table 7, ASA B16 a)
(All dimensions in inches)



Nominal[2,3] pipe size	Inside diameter of fittings	Center to face, true Y, and double branch elbow *A*	Face[3,4] to face lateral *D*	Center[3,4] to face lateral *E*	Center[3,4] to face, true Y, and lateral *F*	Face[5] to face reducer *G*	Diameter of flange	Thickness of flange, minimum	Wall[6] thickness
1	1	3½	7½	5¾	1¾	....	4¼	7⁄16	5⁄16
1¼	1¼	3¾	8	6¼	1¾	....	4⅝	½	5⁄16
1½	1½	4	9	7	2	....	5	9⁄16	5⁄16
2	2	4½	10½	8	2½	5	6	⅝	5⁄16
2½	2½	5	12	9½	2½	5½	7	11⁄16	5⁄16
3	3	5½	13	10	3	6	7½	¾	⅜
3½	3½	6	14½	11½	3	6½	8½	13⁄16	7⁄16
4	4	6½	15	12	3	7	9	15⁄16	½
5	5	7½	17	13½	3½	8	10	15⁄16	½
6	6	8	18	14½	3½	9	11	1	9⁄16
8	8	9	22	17½	4½	11	13½	1⅛	⅝
10	10	11	25½	20½	5	12	16	13⁄16	¾
12	12	12	30	24½	5½	14	19	1¼	13⁄16
14 O.D.	14	14	33	27	6	16	21	1⅜	⅞
16 O.D.	16	15	36½	30	6½	18	23½	17⁄16	1
18 O.D.	18	16½	39	32	7	19	25	19⁄16	11⁄16
20 O.D.	20	18	43	35	8	20	27½	111⁄16	1⅛
24 O.D.	24	22	49½	40½	9	24	32	1⅞	1¼
30 O.D.	30	25	59	49	10	30	38¾	2⅛	17⁄16
36 O.D.	36	....	....	....	....	36	46	2⅜	1⅝
42 O.D.	42	....	....	....	....	42	53	2⅝	113⁄16
48 O.D.	48	....	....	....	....	48	59½	2¾	2

[1] For drilling templates refer to Table LXXXIII.

[2] Laterals, both straight and reducing, sizes 8 in. and larger should be reinforced to compensate for the inherent weakness in the casting design.

[3] Laterals 16 in. and smaller, reducing on the outlet or branch, use the same dimensions center to face and face to face as straight-size fittings corresponding to size of the larger opening. Sizes 18 in. and larger, reducing on the outlet or branch, are made in two lengths depending on the size of the outlet. For dimensions of short-body patterns see Table LXXXIX.

[4] Laterals, reducing on the run only, carry same dimensions center to face and face to face as straight-size fittings corresponding to size of the larger opening.

[5] Reducers and eccentric reducers for all reductions use the same face-to-face dimensions given in the above table of dimensions for the larger opening.

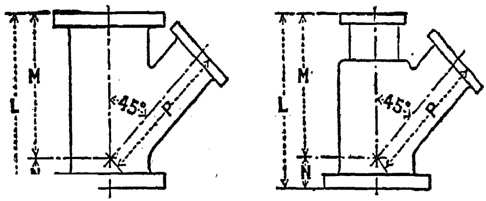
[6] Body thickness at no point shall be less than 87½ per cent of the dimensions given in the table.

Table LXXXIX.—American Standard Class 125 Cast-iron Flanged Fittings

Reducing laterals, short-body pattern[1,2,3,4]

(Table 8, ASA B16 a)

(All dimensions in inches)



Nominal pipe size[2,3,4,5]	Size of branch and smaller[1]	Face to face run L	Center to face run M	Center to face run N	Center to face branch P
1	1	7½			
1¼	1¼	8			
1½	1½	9	7		
2	2	10½	8		
2½	2½	12	9½		
3	3	13	10	3	
3½	3½	14½	11½	3	
4	4	15	12	3	
5	5	17	13½	3½	
6	6	18	14½	3½	
8	8	22	17½	4½	
10	10	25½	20½	5	
12	12	30		5½	
14 O.D.	14	33		6	
16 O.D.	16	36½	30	6½	
18 O.D.	8	26	25	1	
20 O.D.	10	28	27	1	
24 O.D.	12	32	31½		
30 O.D.	14	39	39		

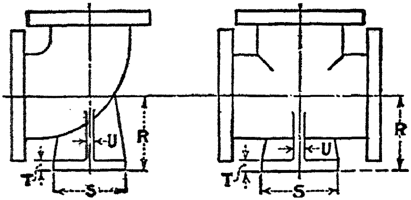
[1] Short-body patterns are used for sizes 18 in. and larger.
[2] Long-body patterns are used when branches are larger than given in the above table and therefore have same dimensions as straight size fittings.
[3] Long-body patterns shall be used for fittings which are reducing on the run only.
[4] Laterals, both straight and reducing, sizes 8 in. and larger shall be reinforced to compensate for the inherent weakness in the casting design.
[5] For flange dimensions, wall thicknesses, and port diameters, see Table LXXXVIII.

Table XC.—American Standard Class 125 Cast-iron Flanged Fittings

Dimensions of base elbows and base tees[2,4,6]

(Table 9, ASA B16 a)

(All dimensions in inches)



Nominal pipe size[5]	Center to base[1] R	Diameter of round base[2,3] S	Thickness of base T	Thickness of ribs U	Size of supporting pipe for base
1	3½	3½	7/16	3/8	¾
1¼	3⅝	3½	7/16	3/8	¾
1½	3¾	4¼	7/16	½	1
2	4⅛	4⅝	½	½	1¼
2½	4½	4⅝	½	½	1¼
3	4⅞	5	9/16	½	1½
3½	5¼	5	9/16	½	1½
4	5½	6	⅝	½	2
5	6¼	7	11/16	⅝	2½
6	7	7	11/16	⅝	2½
8	8⅜	9	15/16	⅞	4
10	9¾	9	15/16	⅞	4
12	11¼	11	1	1	6
14 O.D.	12½	11	1	1	6
16 O.D.	13¾	11	1	1	6
18 O.D.	15	13½	1⅛	1⅛	8
20 O.D.	16	13½	1⅛	1⅛	8
24 O.D.	18½	13½	1⅛	1⅛	8

[1] Dimension center to base is the same as the center to base for anchorage fittings for the same size fittings.

[2] Bases when drilled should be to the template of the flange of the supporting pipe size using only four holes in all cases so placed as to miss the ribs. For drilling templates refer to Table LXXXII. These bases are intended for supports in compression and are not to be used for anchors or supports in tension or shear.

[3] Size and center-to-face dimension of base are determined by the size of the largest opening of fitting.

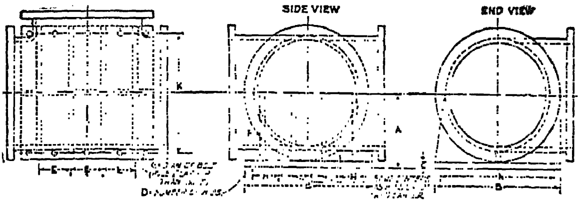
[4] Dimensions for base fittings apply to straight and reducing sizes and long- and short-body patterns.

[5] For sizes larger than 24 in., anchorage fittings are recommended.

Bases not finished unless so ordered.

TABLE XCI.—AMERICAN STANDARD CLASS 125 CAST-IRON FLANGED STANDARD

Dimensions of anchorage bases for tees, straight sizes[1,3]
(Table 10, ASA B16 a)
(All dimensions in inches)



Nominal pipe size[1]	Center to base *A*	Length of square base[2] *B*	Thickness of base *C*	Number of ribs *D*	Centers of ribs and inside bolts *E*	Thickness of ribs *F*	Diameter of bolts *G*	Longitudinal centers from end to second bolt *H*	Transverse bolt centers *K*
2½	4½	7	11/16	1	....	7/16	5/8	4½	4½
3	4⅞	7½	¾	1	....	7/16	5/8	5	5
3½	5¼	8½	13/16	1	....	7/16	5/8	6	6
4	5½	9	15/16	2	4¼	½	5/8	3¼	6½
5	6¼	10	15/16	2	5	½	7/8	3¾	7½
6	7	11	1	2	6	9/16	7/8	4⅜	8¾
8	8⅜	13½	1⅛	2	8	5/8	1	5½	11
10	9¾	16	1 3/16	3	4⅞	¾	1⅛	4¼	13⅜
12	11¼	19	1¼	3	5¾	13/16	1¼	4⅞	15½
14 O.D.	12½	21	1⅜	3	6¾	7/8	1¼	5½	17¾
16 O.D.	13¾	23½	1 7/16	3	7¾	1	1⅜	6	19¾
18 O.D.	15	25	1 9/16	3	8½	1 1/16	1⅜	6⅝	21¾
20 O.D.	16	27½	1 11/16	3	9½	1⅛	1½	7¼	24
24 O.D.	18½	32	1⅞	3	11⅜	1¼	1⅝	8½	28⅜
30 O.D.	22	38¾	2⅛	4	9⅜	1 7/16	1¾	7⅞	34½
36 O.D.	25½	46	2⅜	4	11¼	1⅝	1⅞	9⅛	40¾
42 O.D.	29¼	53	2⅝	4	13	1 13/16	2	10⅜	46¾
48 O.D.	32¾	59½	2¾	4	14⅞	2	2¼	11¾	53¼

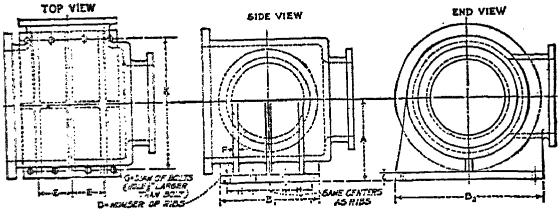
[1] All reducing fittings. 16 in. and smaller, with all reductions have same base as straight sizes.
[2] Size of base determined by size of largest opening fittings.
[3] Bases not finished unless so ordered.

Table XCII.—American Standard Class 125 Cast-iron Flanged Fittings

Dimensions of anchorage bases for reducing tees, short body pattern[1,3]

(Table 11, ASA B16 a)

(All dimensions in inches)



Nominal pipe size	Outlet sizes and smaller[1]	Center to base A	Length of square base[2] B_1	Width of base B_2	Thickness of base C	Number of ribs D	Centers of ribs and inside bolts E	Thickness of ribs F	Diameter of bolts G	Longitudinal center from end to second bolt H	Transverse bolt centers K
18 O.D.	12	15½	19	25	1 9/16	3	5¼	1 1/16	1¼	5⅛	21½
20 O.D.	14	16	21	27½	1 11/16	3	6	1⅛	1¼	5⅜	23¼
24 O.D.	16	18½	23½	32	1⅞	3	7	1¼	1⅜	6¼	28
30 O.D.	20	22	27½	38¾	2⅛	3	9	1 7/16	1½	7¼	34¾
36 O.D.	24	25½	32	46	2⅜	3	10¾	1⅝	1½	8⅜	41½
42 O.D.	24	29¼	36½	53	2⅝	4	8½	1 13/16	1⅝	7½	48½
48 O.D.	30	32¾	41¾	59½	2¾	4	9¾	2	1⅝	8¼	53¾

[1] All reducing fittings, 16 in. and smaller, with all reductions, have same base as straight sizes.
[2] Size of base determined by size of largest opening of fitting.
[3] Base is not finished unless so ordered.

TABLE XCIII.—AMERICAN STANDARD CLASS 125 CAST-IRON FLANGED FITTINGS

Theoretical weights of flanges, elbows, crosses, tees, side-outlet tees, and laterals[1]

(Table 1*A*, ASA B16 a)

Nominal pipe size	Companion flanges, pounds	Blind flanges, pounds	90-deg elbow, pounds	45-deg elbow, pounds	90-deg long-radius elbow, pounds	Side-outlet elbow, pounds	Tees, pounds	Cross and side-outlet tees not ribbed, pounds	Laterals[2] not ribbed, pounds
1	2	2	5	4	7	8	9	11	10
1¼	2	3	7	6	9	10	11	15	13
1½	3	3	9	8	11	13	15	19	17
2	5	5	14	12	16	20	21	28	25
2½	7	7	19	17	23	28	30	39	36
3	8	9	24	20	28	34	37	48	44
3½	11	12	31	27	37	46	49	63	59
4	14	16	41	36	48	59	64	82	75
5	17	20	52	45	62	74	81	105	96
6	22	25	68	60	85	96	105	135	125
8	31	42	110	94	145	150	165	210	210
10	45	63	175	145	230	240	270	330	340
12	63	88	250	220	350	340	380	470	520
14 O.D.	82	115	350	270	470	470	530	650	680
16 O.D.	105	160	470	360	670	620	700	850	950
18 O.D.	120	190	580	420	840	760	860	1,040	1,150
20 O.D.	150	250	740	540	1,080	970	1,100	1,330	1,480
24 O.D.	220	370	1,160	800	1,640	1,510	1,730	2,080	2,080
30 O.D.	...	620	1,850	1,430	2,800	2,350	2,710	3,210	3,680
36 O.D.	...	990	2,800	2,280	4,450	3,500	4,050	4,750	
42 O.D.	...	1,470	4,010	3,380	6,610	4,930	5,790	6,710	
48 O.D.	...	2,000	5,400	4,680	9,250	6,520	7,620	8,740	

NOTE.—Weights are for information only; not mandatory.

[1] All weights listed are for fittings faced and drilled, based on minimum thicknesses and dimensions given in preceding tables, without allowances for variation. Cast iron is assumed to weigh 0.26 lb per cu in.

[2] Weights of crosses and laterals do not include reinforcing ribs.

Table XCIV.—American Standard Class 125 Cast-iron Flanged Fittings

Theoretical weights in pounds of reducing elbows, reducers, and eccentric reducers[1]

(Table 2A, ASA B16 a)

Nominal pipe size	Reducing elbows	Reducers and eccentric reducers	Nominal pipe size	Reducing elbows	Reducers and eccentric reducers
3 × 2½	22	19	12 × 10	220	180
3 × 2	19	16	12 × 8	190	155
3 × 1½	17		12 × 6	165	140
3½ × 3	28	24	14 × 12	320	250
3½ × 2	24	20	14 × 10	280	220
4 × 3½	37	31	14 × 8	240	200
4 × 3	33	28	16 × 14	420	340
4 × 2½	31	26	16 × 12	380	310
4 × 2	29	24	16 × 10	340	280
5 × 4	48	39	16 × 8	300	250
5 × 3	40	32	18 × 16	540	430
5 × 2½	37	31	18 × 14	480	380
6 × 5	60	50	18 × 12	440	350
6 × 4	56	47	18 × 10	390	320
6 × 3½	51	43	20 × 18	680	520
6 × 3	47	39	20 × 16	640	490
8 × 6	90	77	20 × 14	570	450
8 × 5	82	71	20 × 12	520	410
8 × 4	77	66	24 × 20	1,010	760
10 × 8	150	120	24 × 18	930	700
10 × 6	125	100	24 × 16	880	670
10 × 5	115	95	24 × 12	740	580

[1] All weights listed are for fittings faced and drilled, based on minimum thicknesses and dimensions given in preceding tables, without allowances for variation. Cast iron is assumed to weigh 0.26 lb per cu in.

Table XCV.—American Standard Class 125 Cast-iron Flanged Fittings

Theoretical weights in pounds of reducing laterals[1]
(Table 3*A*, ASA B16 a)

Nominal pipe size	Lateral reducing outlet,[2] not ribbed	Nominal pipe size	Lateral reducing outlet,[2] not ribbed
3 × 3 × 2½	42	12 × 12 × 10	470
3 × 3 × 2	39	12 × 12 × 8	430
3 × 3 × 1½	36	12 × 12 × 6	400
3½ × 3½ × 3	55	14 × 14 × 12	640
3½ × 3½ × 2½	52	14 × 14 × 10	590
3½ × 3½ × 2	49	14 × 14 × 8	550
4 × 4 × 3½	70	16 × 16 × 14	880
4 × 4 × 3	66	16 × 16 × 12	830
4 × 4 × 2½	63	16 × 16 × 10	790
4 × 4 × 2	60	16 × 16 × 8	740
5 × 5 × 4	93	18 × 18 × 16	1,100
5 × 5 × 3½	86	18 × 18 × 14	1,030
5 × 5 × 3	82	18 × 18 × 12	980
5 × 5 × 2½	79	18 × 18 × 10	930
6 × 6 × 5	120	20 × 20 × 18	1,400
6 × 6 × 4	115	20 × 20 × 16	1,350
6 × 6 × 3½	105	20 × 20 × 14	1,270
6 × 6 × 3	105	20 × 20 × 12	1,220
8 × 8 × 6	195	20 × 20 × 10	*840
8 × 8 × 5	180	24 × 24 × 20	2,040
8 × 8 × 4	175	24 × 24 × 18	1,950
10 × 10 × 8	310	24 × 24 × 16	1,890
10 × 10 × 6	280	24 × 24 × 14	1,810
10 × 10 × 5	270	24 × 24 × 12	*1,250

[1] All weights listed are for fittings faced and drilled, based on minimum thicknesses and dimensions given in preceding tables, without allowances for variation. Cast iron is assumed to weigh 0.26 lb per cu in.
[2] Weights of laterals do not include reinforcing ribs.
* These sizes made in the short-body pattern only.

Table XCVI.—American Standard Class 125 Cast-iron Flanged Fittings

Theoretical weights in pounds of reducing tees[1]
(Table 4A, ASA B16 a)

Size	Weight	Size	Weight	Size	Weight
3 × 3 × 2½	36	6 × 6 × 3	89	14 × 14 × 10	480
3 × 3 × 2	33	6 × 5 × 6	100	14 × 14 × 8	460
3 × 3 × 1½	31	6 × 5 × 5	95	14 × 12 × 14	510
3 × 2½ × 3	35	6 × 5 × 4	92	14 × 12 × 12	490
3 × 2½ × 2½	34	6 × 5 × 3	84	14 × 12 × 10	460
3 × 2½ × 2	31	6 × 4 × 6	98	14 × 12 × 8	440
3 × 2½ × 1½	29	6 × 4 × 5	93	14 × 10 × 14	490
3 × 2 × 3	33	6 × 4 × 4	89	14 × 10 × 12	470
3 × 2 × 2½	32	6 × 4 × 3	82	14 × 10 × 10	450
3 × 2 × 2	29	6 × 3 × 6	92	14 × 10 × 8	420
3 × 2 × 1½	27	6 × 3 × 5	86	14 × 8 × 14	480
3 × 1½ × 3	32	6 × 3 × 4	83	14 × 8 × 12	460
3 × 1½ × 2½	30	6 × 3 × 3	76	14 × 8 × 10	430
3 × 1½ × 2	27	8 × 8 × 6	150	14 × 8 × 8	410
3 × 1½ × 1½	25	8 × 8 × 5	145	16 × 16 × 14	670
3½ × 3½ × 3	46	8 × 8 × 4	145	16 × 16 × 12	650
3½ × 3½ × 2½	44	8 × 6 × 8	155	16 × 16 × 10	620
3½ × 3½ × 2	42	8 × 6 × 6	140	16 × 16 × 8	610
4 × 4 × 3½	60	8 × 6 × 5	135	16 × 14 × 16	680
4 × 4 × 3	57	8 × 6 × 4	130	16 × 14 × 14	650
4 × 4 × 2½	55	8 × 5 × 8	150	16 × 14 × 12	630
4 × 4 × 2	53	8 × 5 × 6	135	16 × 14 × 10	600
4 × 3 × 4	57	8 × 5 × 5	130	16 × 14 × 8	580
4 × 3 × 3	50	8 × 5 × 4	125	16 × 12 × 16	670
4 × 3 × 2½	49	8 × 4 × 8	150	16 × 12 × 14	640
4 × 3 × 2	46	8 × 4 × 6	135	16 × 12 × 12	620
4 × 2½ × 4	56	8 × 4 × 5	130	16 × 12 × 10	590
4 × 2½ × 3	49	8 × 4 × 4	125	16 × 12 × 8	570
4 × 2½ × 2½	47	10 × 10 × 8	250	16 × 10 × 16	650
4 × 2½ × 2	45	10 × 10 × 6	240	16 × 10 × 14	620
4 × 2 × 4	54	10 × 10 × 5	230	16 × 10 × 12	600
4 × 2 × 3	47	10 × 8 × 10	260	16 × 10 × 10	570
4 × 2 × 2½	45	10 × 8 × 8	240	16 × 10 × 8	550
4 × 2 × 2	43	10 × 8 × 6	220	16 × 8 × 16	640
5 × 5 × 4	78	10 × 6 × 10	250	16 × 8 × 14	610
5 × 5 × 3½	74	10 × 6 × 8	230	16 × 8 × 12	580
5 × 5 × 3	70	10 × 6 × 6	210	16 × 8 × 10	560
5 × 5 × 2½	68	12 × 12 × 10	360	16 × 8 × 8	540
5 × 4 × 5	78	12 × 12 × 8	340	18 × 18 × 16	860
5 × 4 × 4	75	12 × 12 × 6	320	18 × 18 × 14	820
5 × 4 × 3	68	12 × 10 × 12	370	18 × 18 × 12	*660
5 × 4 × 2½	66	12 × 10 × 10	340	18 × 18 × 10	*640
5 × 3 × 5	72	12 × 10 × 8	320	20 × 20 × 18	1,060
5 × 3 × 4	68	12 × 10 × 6	310	20 × 20 × 16	1,040
5 × 3 × 3	61	12 × 8 × 12	350	20 × 20 × 14	*840
5 × 3 × 2½	59	12 × 8 × 10	330	20 × 20 × 12	*820
5 × 2½ × 5	70	12 × 8 × 8	310	20 × 20 × 10	*790
5 × 2½ × 4	67	12 × 8 × 6	300	24 × 24 × 20	1,640
5 × 2½ × 3	60	12 × 6 × 12	340	24 × 24 × 18	1,600
5 × 2½ × 2½	58	12 × 6 × 10	320	24 × 24 × 16	*1,170
6 × 6 × 5	99	12 × 6 × 8	300	24 × 24 × 14	*1,140
6 × 6 × 4	96	12 × 6 × 6	280	24 × 24 × 12	*1,110
6 × 6 × 3½	92	14 × 14 × 12	500		

[1] All weights listed are for fittings faced and drilled, based on minimum thicknesses and dimensions given in preceding tables, without allowances for variation. Cast iron is assumed to weigh .26 lb per cu in.

* These sizes made in short-body pattern only.

TABLE XCVII.—AMERICAN STANDARD CLASS 125 CAST-IRON FLANGED FITTINGS

Theoretical weights in pounds of reducing crosses[1,2]
(Table 5A, ASA B16 a)

Size	Weight
3 × 3 × 2½ × 2½	44
3 × 3 × 2 × 2	40
3 × 3 × 1½ × 1½	36
3½ × 3½ × 3 × 3	57
3½ × 3½ × 2½ × 2½	53
3½ × 3½ × 2 × 2	47
4 × 4 × 3½ × 3½	74
4 × 4 × 3 × 3	68
4 × 4 × 2½ × 2½	64
4 × 4 × 2 × 2	59
5 × 5 × 4 × 4	96
5 × 5 × 3½ × 3½	89
5 × 5 × 3 × 3	82
5 × 5 × 2½ × 2½	78
6 × 6 × 5 × 5	120
6 × 6 × 4 × 4	115
6 × 6 × 3½ × 3½	105
6 × 6 × 3 × 3	100
8 × 8 × 6 × 6	190
8 × 8 × 5 × 5	175
8 × 8 × 4 × 4	165
10 × 10 × 8 × 8	300
10 × 10 × 6 × 6	270
10 × 10 × 5 × 5	250
12 × 12 × 10 × 10	420
12 × 12 × 8 × 8	380
12 × 12 × 6 × 6	350
14 × 14 × 12 × 12	600
14 × 14 × 10 × 10	550
14 × 14 × 8 × 8	500
16 × 16 × 14 × 14	790
16 × 16 × 12 × 12	740
16 × 16 × 10 × 10	690
16 × 16 × 8 × 8	650
18 × 18 × 16 × 16	1,000
18 × 18 × 14 × 14	930
18 × 18 × 12 × 12	*750
18 × 18 × 10 × 10	*700
20 × 20 × 18 × 18	1,250
20 × 20 × 16 × 16	1,200
20 × 20 × 14 × 14	*960
20 × 20 × 12 × 12	*910
20 × 20 × 10 × 10	*860
24 × 24 × 20 × 20	1,900
24 × 24 × 18 × 18	1,810
24 × 24 × 16 × 16	*1,310
24 × 24 × 14 × 14	*1,250
24 × 24 × 12 × 12	*1,210

, based on minimum thickness and dimen- sions given in p variation. Cast iron is assumed to weigh 0.26 lb per cu in.

* These sizes made in short-body pattern only.

American Standard for
CAST-IRON PIPE FLANGES AND FLANGED FITTINGS, CLASS 250

ASA B16b-1944

Abstracted[1]

Pressure Rating.—Flanges and fittings to this standard are rated as follows:

Sizes	Saturated steam, psi gage	Water at ordinary air temperature, psi gage
1 to 12, inclusive	250	400
14 to 24, inclusive	200	300[2]
30 to 48, inclusive[1]	100	300[2]

[1] Sizes 30 to 48 in. are included for convenience where special fittings larger than 24 in. are required.
[2] Water-service pressure ratings are applicable to flanges only and not to fittings.

Sizes.—Sizes are identified by the corresponding "nominal pipe size."

Marking.—All fittings shall have marks cast on them indicating the manufacturer and figures indicating the saturated service pressure ratings for the particular size as given above, in accordance with the principles established in MSS Standard Practice SP25 (see abstract, page 687).

Material.—Flanges and fittings 12 in. and smaller shall be made of material at least equal to Class A regular gray iron of ASTM Specification A126. Flanges and fittings 14 in. and larger shall be made of material at least equal to Class B higher strength gray iron of ASTM Specification A126 (see abstract, page 625).

Dimensions.—The dimensions of Class 250 flanges and fittings are given in Tables XCVIII to CII.

Wall Thickness Tolerance.—The wall thicknesses of the castings at no point shall be less than 87½ per cent of the dimensions given.

Tests.—Fittings shall be designed to withstand hydrostatic test pressures of twice the rated steam pressure. Hydrostatic tests are not required unless specified by the user.

Facing.—A raised face $\frac{1}{16}$ in. high of the diameters given in Table XCVIII is included in the minimum flange thickness and center-to-face dimensions. An inspection limit of $\pm\frac{1}{32}$ in. shall be allowed on all center-to-contact-surface dimensions for sizes up to and including 10 in. and $\pm\frac{1}{16}$ in. on sizes larger than 10 in. Contact-surface-to-contact-surface inspection limits are two times the center-to-contact-surface limits.

Bolting.—Requirements for bolt holes and recommendations for use of bolt-studs are given in footnotes of Table XCVIII. Bolts shall be of steel with American Standard regular unfinished square heads or heavy unfinished hexagonal heads. Nuts shall be of carbon steel with heavy hexagonal dimensions

[1] For complete standard, reference may be made to ASA B16b (see note, p. 369).

TABLE XCVIII.—AMERICAN STANDARD, CLASS 250 CAST-IRON FLANGES AND FITTINGS

Dimensions of cast-iron flanges, drilling for bolts, and their lengths
(Table 2, ASA B16 b)
(All dimensions in inches)

Nominal pipe size	Diameter of flange	Thickness[1] of flange, min	Diameter of raised face	Diameter of bolt circle	Diameter[2] of bolt holes	Number[2] of bolts	Size of bolts	Length[3] of bolts	Length[4] of bolt-studs with two nuts	Size of ring gasket
1	4⅞	11/16	2 11/16	3½	¾	4	⅝	2½	....	1 × 2⅞
1¼	5¼	¾	3 1/16	3⅞	¾	4	⅝	2½	....	1¼ × 3¼
1½	6⅛	13/16	3 9/16	4½	⅞	4	¾	2¾	....	1½ × 3¾
2	6½	⅞	4 3/16	5	¾	8	⅝	2¾	....	2 × 4⅜
2½	7½	1	4 15/16	5⅞	⅞	8	¾	3¼	....	2½ × 5⅛
3	8¼	1⅛	5 11/16	6⅝	⅞	8	¾	3½	....	3 × 5⅞
3½	9	1 3/16	6 5/16	7¼	⅞	8	¾	3½	....	3½ × 6½
4	10	1¼	6 15/16	7⅞	⅞	8	¾	3¾	....	4 × 7⅛
5	11	1⅜	8 5/16	9¼	⅞	8	¾	4	....	5 × 8½
6	12½	1 7/16	9 11/16	10⅝	⅞	12	¾	4	....	6 × 9⅞
8	15	1⅝	11 15/16	13	1	12	⅞	4½	....	8 × 12⅛
10	17½	1⅞	14 1/16	15¼	1⅛	16	1	5¼	....	10 × 14¼
12	20½	2	16 7/16	17¾	1¼	16	1⅛	5½	....	12 × 16⅝
14 O.D.	23	2⅛	18 15/16	20¼	1¼	20	1⅛	6	....	13¼ × 19⅛
16 O.D.	25½	2¼	21 1/16	22½	1⅜	20	1¼	6¼	....	15¼ × 21¼
18 O.D.	28	2⅜	23 5/16	24¾	1⅜	24	1¼	6½	....	17 × 23½
20 O.D.	30½	2½	25 9/16	27	1⅜	24	1¼	6¾	....	19 × 25¾
24 O.D.	36	2¾	30¼	32	1 11/16	24	1½	7¾	9½	23 × 30½
*30 O.D.	43	3	37 3/16	39¼	2	28	1¾	8½	10½	29 × 37½
*36 O.D.	50	3⅜	43 11/16	46	2¼	32	2	9½	11¾	34½ × 44
*42 O.D.	57	3 11/16	50 7/16	52¾	2¼	36	2	10¼	12½	40¼ × 50¾
*48 O.D.	65	4	58 7/16	60¾	2¼	40	2	10¾	13	46 × 58¾

* These sizes are included for convenience where special fittings larger than 24 in. are required.

1 All Class 250 cast-iron flanges have a 1/16-in. raised face. This raised face is included in the face-to-face, center-to-face, and the minimum thickness of flange dimensions.

2 Drilling templates are in multiples of four, so that fittings may be made to face in any quarter, and bolt holes straddle the center line. For bolts smaller than 1½ in. the bolt holes shall be drilled ⅛ in. larger in diameter than the nominal diameter of the bolt. Holes for bolts 1½ in. shall be drilled 3/16 in. larger in diameter than the nominal diameter of the bolt. Holes for bolts 1¾ in. and larger shall be drilled ¼ in. larger than nominal diameter of bolts.

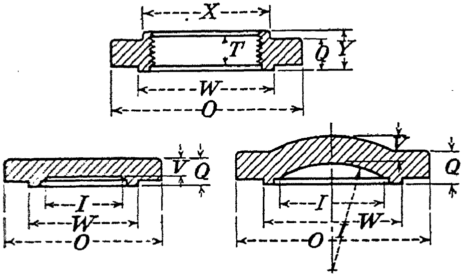
3 Spot facing is not required for ordinary services except as follows:

Flange sizes	If flanges after facing exceed the min thickness by the following	Spot face to min flange thickness with plus tolerance of
12 in. and smaller	⅛ in.	
14 to 24 in........	3/16 in.	
30 in. and larger..	¼ in.	

4 For bolts 1¾ in. in diameter and larger, bolt-studs with a nut on each end are recommended.

Table XCIX.—American Standard, Class 250 Cast-iron Flanges and Fittings

Dimensions of screwed companion and blind flanges[1,2,3]
(Table 3, ASA B16 b)
(All dimensions in inches)



8 in. and smaller   10 in. and larger

Nominal pipe size	Diameter[2] of port *I*	Diameter of flange *O*	Thickness[1] of flange, min *Q*	Wall thickness *V*	Diameter hub, min *X*	Length through hub, min *Y*	Length of threads, min *T*	Diameter[1] of raised face *W*
1	1	4 7/8	11/16	....	2 1/16	7/8	0.68	2 11/16
1 1/4	1 1/4	5 1/4	3/4	....	2 1/2	1	0.76	3 1/16
1 1/2	1 1/2	6 1/8	13/16	....	2 3/4	1 1/8	0.87	3 9/16
2	2	6 1/2	7/8	....	3 5/16	1 1/4	1.00	4 3/16
2 1/2	2 1/2	7 1/2	1	....	3 15/16	1 7/16	1.14	4 15/16
3	3	8 1/4	1 1/8	....	4 5/8	1 9/16	1.20	5 11/16
3 1/2	3 1/2	9	1 3/16	....	5 1/4	1 5/8	1.25	6 5/16
4	4	10	1 1/4	....	5 3/4	1 3/4	1.30	6 15/16
5	5	11	1 3/8	....	7	1 7/8	1.41	8 5/16
6	6	12 1/2	1 7/16	....	8 1/8	1 15/16	1.51	9 11/16
8	8	15	1 5/8	....	10 1/4	2 3/16	1.71	11 15/16
10	10	17 1/2	1 7/8	15/16	12 5/8	2 3/8	1.92	14 1/16
12	12	20 1/2	2	1	14 3/4	2 9/16	2.12	16 7/16
14 O.D.	13 1/4	23	2 1/8	1 1/8	16 1/4	2 11/16	2.25	18 15/16
16 O.D.	15 1/4	25 1/2	2 1/4	1 1/4	18 3/8	2 7/8	2.45	21 1/16
18 O.D.	17	28	2 3/8	1 3/8	......	....	....	23 5/16
20 O.D.	19	30 1/2	2 1/2	1 1/2	......	....	....	25 9/16
24 O.D.	23	36	2 3/4	1 5/8	......	....	....	30 1/4

[1] All Class 250 cast-iron flanges have a 1/16-in. raised face. This raised face is included in the minimum thickness of flange dimensions.

[2] All blind flanges for sizes 18 in. (17 1/2 in. O.D.) and larger must be dished, with inside radius equal to the port diameter. The wall thickness at no point shall be less than 87 1/2 per cent of the dimensions given in the table.

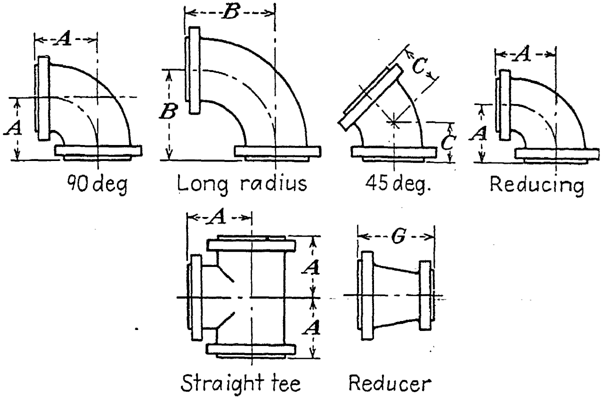
[3] For drilling templates, refer to Table XCVIII.

TABLE C.—AMERICAN STANDARD, CLASS 250 CAST-IRON FLANGES AND FITTINGS

Dimensions of elbows, tees, and reducers[1,6]

(Table 4, ASA B16 b)

(All dimensions in inches)



Nominal pipe size	Inside diameter of fitting, min	Wall[5] thickness of body	Diameter of flange	Thickness[1] of flange, min	Diameter[1] of raised face	Center[1,2,3] to face elbow and tee *A*	Center[1] to face long-radius elbow *B*	Center[1,3] to face 45-deg elbow *C*	Face[1,4] to face reducer *G*
2	2	7/16	6 1/2	7/8	4 3/16	5	6 1/2	3	5
2 1/2	2 1/2	1/2	7 1/2	1	4 15/16	5 1/2	7	3 1/2	5 1/2
3	3	9/16	8 1/4	1 1/8	5 11/16	6	7 3/4	3 1/2	6
3 1/2	3 1/2	9/16	9	1 3/16	6 5/16	6 1/2	8 1/2	4	6 1/2
4	4	5/8	10	1 1/4	6 15/16	7	9	4 1/2	7
5	5	11/16	11	1 3/8	8 5/16	8	10 1/4	5	8
6	6	3/4	12 1/2	1 7/16	9 11/16	8 1/2	11 1/2	5 1/2	9
8	8	13/16	15	1 5/8	11 15/16	10	14	6	11
10	10	15/16	17 1/2	1 7/8	14 1/16	11 1/2	16 1/2	7	12
12	12	1	20 1/2	2	16 7/16	13	19	8	14
14 O.D.	13 1/4	1 1/8	23	2 1/8	18 15/16	15	21 1/2	8 1/2	16
16 O.D.	15 1/4	1 1/4	25 1/2	2 1/4	21 1/16	16 1/2	24	9 1/2	18
18 O.D.	17	1 3/8	28	2 3/8	23 5/16	18	26 1/2	10	19
20 O.D.	19	1 1/2	30 1/2	2 1/2	25 9/16	19 1/2	29	10 1/2	20
24 O.D.	23	1 5/8	36	2 3/4	30 1/4	22 1/2	34	12	24

[1] All Class 250 cast-iron flanges have a 1/16-in. raised face. This raised face is included in the face-to-face, center-to-face, and minimum thickness of flange dimensions.

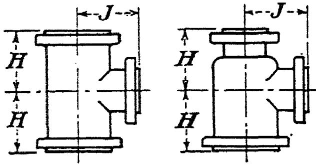
[2] Reducing elbows carry the same dimensions center to face as regular straight-size elbows corresponding to the size of the larger opening. Tees 16 in. and smaller reducing on the outlet have the same dimensions center to face and face to face as straight-size fittings corresponding to

(see ASA B18.2 abstracted on page 550). Bolts, or bolt-studs, and nuts shall be threaded in accordance with Class 2 fit of ASA B1.1 (see abstract, page 546).

Spot Facing.—Requirements for spot facing are given in footnotes of Table XCVIII.

TABLE CI.—AMERICAN STANDARD, CLASS 250 CAST-IRON FLANGES AND FITTINGS

Dimensions of reducing tees[2,4] (short-body patterns)
(Table 5, ASA B16 b)
(All dimensions in inches)



Nominal pipe size[2]	Size of outlet and smaller[3]	Center to face run[1] H	Face to face run[1] $H+H$	Center to face outlet[1] J
18 O.D.	12	14	28	17
20 O.D.	14	15½	31	18½
24 O.D.	16	17	34	21½

[1] All Class 250 cast-iron flanges have a 1/16-in. raised face. This raised face is included in the face-to-face, center-to-face and the minimum thickness of flange dimensions.

[2] Short-body patterns are used for sizes 18 in. and larger. All reducing tees sizes 16 in. and smaller have same face-to-face and center-to-face dimensions as straight sizes (see Table C).

[3] Long-body patterns are used when outlets are larger than given in the above table and, therefore, have the same dimensions as straight-size fittings.

[4] Tees reducing on the run carry same dimensions only center to face and face to face as straight-size fittings corresponding to size of the larger opening. Tees increasing on outlet, known as bull head tees, will have same center-to-face and face-to-face dimensions as a straight fitting of the size of the outlet. For example: A 12 × 12 × 18 in. tee will be governed by the dimensions of the 18-in. long-body tee, given in Table C; namely, 18 in. center to face of all openings and 36 in. face to face.

For flange dimensions, wall thicknesses, and port diameters, see Table C.
For drilling dimensions, see Table XCVIII.
For face-to-face dimensions of valves see pp. 555 to 560.

---

the size of the larger opening. Sizes 18 in. and larger reducing on the outlet are made in two lengths depending on the size of the outlet. For dimensions of the short-body pattern see Table CI.

[2] Special-degree elbows ranging from 1 to 45 deg, inclusive, have the same center-to-face dimensions given for 45-deg elbows, and those over 45 deg and up to 90 deg, inclusive, shall have the same center-to-face dimensions given for 90-deg elbows. The angle designation of an elbow is its deflection from straight line flow and is the angle between the flange faces.

[4] Reducers, for all reductions, use the same face-to-face dimensions given in the above table of dimensions for the larger opening.

[5] Wall thickness at no point shall be less than 87½ per cent of the dimensions given in the table.

[6] For drilling templates refer to Table XCVIII.

For external surface see p. 698.

For face-to-face dimensions of valves see pp. 555–560.

Reducing Fittings.—Reducing fittings shall be ordered by the designation of the outlets in their proper sequence as indicated in the sketches on page 599. The practice followed in dimensioning reducing fittings is given in the footnotes of Table CI.

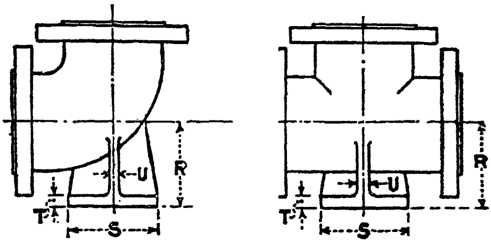
Elbows.—The angle designation of an elbow is its deflection from straight-line flow and is the angle between the flange faces. See footnote of Table C for center-to-face dimensions of special degree elbows.

TABLE CII.—AMERICAN STANDARD, CLASS 250 CAST-IRON FLANGES AND FITTINGS

Dimensions of base elbows and base tees[3,4]

(Table 6, ASA B16 b)

(All dimensions in inches)



Nominal pipe size	Center to base R	Diameter of round base[1,2] S	Thickness of base T	Thickness of ribs U	Size of supporting pipe for base[1]
2	4½	5¼	¾	½	1¼
2½	4¾	5¼	¾	½	1¼
3	5¼	6⅛	13/16	⅝	1½
3½	5⅝	6⅛	13/16	⅝	1½
4	6	6½	⅞	⅝	2
5	6¾	7½	1	¾	2½
6	7½	7½	1	¾	2½
8	9	10	1¼	⅞	4
10	10½	10	1¼	⅞	4
12	12	12½	1 7/16	1	6
14 O.D.	13½	12½	1 7/16	1	6
16 O.D.	14¾	12½	1 7/16	1⅛	6
18 O.D.	16¼	15	1⅝	1⅛	8
20 O.D.	17⅞	15	1⅝	1¼	8
24 O.D.	20¾	17½	1⅞	1¼	10

[1] Bases when drilled should be to the template of the flange of the supporting pipe size using only four holes in all cases so placed as to miss the ribs. For drilling templates refer to Table XCVIII. These bases are intended for supports in compression and are not to be used for anchors or supports in tension or shear.

[2] Size and center-to-face dimension of base are determined by the size of the largest opening of fitting.

[3] Dimensions for base fittings apply to straight and reducing sizes, and long- and short-body patterns.

[4] Bases not finished unless so ordered.

Drain Tappings.—The maximum size of hole that can be tapped in the wall of the fittings without adding a boss is shown in the following table:

MAXIMUM SIZE OF TAPPED HOLE IN FITTING WITHOUT ADDING BOSSES

Size of fitting, in.	2 to 3	4 to 5	6	8	10	12	14 to 24
Size of tapped hole, in.	3/8	1/2	3/4	1	1 1/4	1 1/2	2

When bosses are required, the method of designating the locations of the tapped holes for drains is shown in Fig. 39, page 600. Each possible location is designated by a letter so that desired locations for the various types of fittings may be definitely specified without the use of further sketches or description.

American Standard for 800-LB HYDRAULIC CAST-IRON FLANGES AND FLANGED FITTINGS

ASA B16 b1-1931

Abstracted[1]

This standard is recommended for water pressures up to 800 lb at or near ordinary air temperatures where no shock is involved.

All fittings shall be marked to identify the manufacturer and shall give the pressure rating of 800 lb in accordance with MSS Standard Practice SP25.

Materials.—The dimensions prescribed in this standard are based on gray iron castings of high quality produced under regular control of chemical and physical properties by a recognized process. The manufacturer shall be prepared to certify that his product has been so produced and that the chemical and physical properties thereof, as proved by test specimens, are equal at least to the minimum requirements specified in ASTM Specifications A-126 for Grade B, higher strength gray iron, *viz.*:

Sulphur	0.12 per cent, max
Phosphorus	0.75 per cent, max
Tensile strength	31,000 psi, min

Facing.—All 800-lb cast-iron flanges and flanged fittings shall have a 1/4-in. raised face not included in the flange thickness and of a diameter shown in Table CIII. The illustrations above Table CIII show the application of raised-face and male-female facings.

Bolting.—Drilling templates are in multiples of four, so that fittings may be made to face any quarter.

Bolt holes shall straddle the center line. All bolt holes shall be drilled 1/8 in. larger in diameter than the nominal size of the bolt.

[1] For complete standard, reference may be made to ASA B16 b1 (see note, p. 430). References to marking, materials, specifications, and bolting have been changed to agree with latest practice and specifications.

Bolts may be of steel or wrought iron with square heads and hexagonal nuts conforming to American Standard Unfinished Regular (or Heavy) Bolt Heads and Unfinished Heavy Nuts (see page 550).

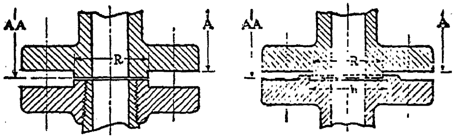
Spot Facing.—The bolt holes of 800-lb cast-iron flanges and flanged fittings need not be spot-faced, unless so ordered.

Reducing Fittings.—All reducing fittings, whether ells, tees, crosses, or reducers, and all double-branch and side-outlet fittings shall have the same center-to-face and face-to-face dimensions throughout as for the largest opening.

Elbows.—Special-degree elbows ranging from 1 to 45 deg, inclusive, shall have the same center-to-face dimension given for 45-deg elbows, and those over 45 deg and up to 90 deg, inclusive, shall have the same center-to-face dimensions given for 90-deg elbows. The angle designation of an elbow is its deflection from straight-line flow and is the angle between the flange faces.

Laterals.—The use of this type of fitting when made of Cast Iron is discouraged. Where necessary, use should be made of 600-lb Steel Flanged Laterals (B16 e, abstracted on page 660) of which the flange dimensions and bolting are interchangeable with this standard.

TABLE CIII.—800-LB HYDRAULIC CAST-IRON FLANGED FITTINGS
Facing dimensions
(Table 1, ASA B16 b1)
(All dimensions in inches)



RAISED FACE LARGE MALE-FEMALE

Nominal pipe size	Inside diameter of fitting	Outside diameter of male[1] R	Outside diameter of large female W	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange, minimum[1,2]	Diameter of bolt circle	Number of bolts	Size of bolts	Diameter of bolt holes
2	2	3⅝	3¹¹⁄₁₆	9⁄16	6½	1¼	5	8	⅝	¾
2½	2½	4⅛	4³⁄₁₆	⅝	7½	1⅜	5⅞	8	¾	⅞
3	3	5	5¹⁄₁₆	⅝	8¼	1½	6⅝	8	¾	⅞
3½	3½	5½	5⁹⁄₁₆	¾	9	1⅝	7¼	8	⅞	1
4	4	6³⁄₁₆	6¼	⅞	10¾	1⅞	8½	8	⅞	1
5	5	7⁵⁄₁₆	7⅜	1⅛	13	2⅛	10½	8	1	1⅛
6	6	8½	8⁹⁄₁₆	1⅛	14	2¼	11½	12	1	1⅛
8	7⅞	10⅝	10¹¹⁄₁₆	1⅜	16½	2½	13¾	12	1⅛	1¼
10	9¾	12¾	12¹³⁄₁₆	1⅝	20	2⅞	17	16	1¼	1⅜
12	11¾	15	15¹⁄₁₆	1¾	22	3	19¼	20	1¼	1⅜

[1] A tolerance of ±0.016 in. (1⁄64 in.) is allowed on the inside and outside diameters of all facings.
[2] The raised face of ¼ in. is not included in the thickness of flange, minimum. The depth of the female facing is to be 3⁄16 in.

TABLE CIV. 800-LB HYDRAULIC CAST-IRON FLANGED FITTINGS
Center-to-contact-surface dimensions
(Table 2, ASA B16 b1)
(All dimensions in inches)

Nominal pipe size	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange, minimum[1]	Diameter of bolt circle	Number of bolts	Size of bolts	Center to contact surface of raised face, elbow, tee, and cross[1,2] AA	Center to contact surface of raised face, 45-deg ell[1] CC	Long center to contact surface of raised face, lateral[1] EE	Short center to contact surface of raised face, lateral[1] FF	Contact surface to contact surface reducer[1] GG
2	2	9/16	6½	1¼	5	8	5/8	5¾	4¼	10¼	3½	6
2½	2½	5/8	7½	1⅜	5⅞	8	¾	6½	4½	11½	3½	6¾
3	3	5/8	8¼	1½	6⅝	8	¾	7	5	12¾	4	7¼
3½	3½	¾	9	1⅝	7¼	8	⅞	7½	5½	14	4½	7¾
4	4	⅞	10¾	1⅞	8½	8	⅞	8½	6	16½	4½	8¾
5	5	1⅛	13	2⅛	10½	8	1	10	7	19½	6	10¼
6	6	1⅛	14	2¼	11½	12	1	11	7½	21	6½	11¼
8	7⅞	1⅜	16½	2½	13¾	12	1⅛	13	8½	24½	7	13¼
10	9¾	1⅝	20	2⅞	17	16	1¼	15½	9½	29½	8	15¾
12	11¾	1¾	22	3	19¼	20	1¼	16½	10	31½	8½	16¾

[1] The raised face of ¼ in. is not included in thickness of flange, minimum, but is included in center-to-contact surface of raised face dimensions and contact-surface-to-contact-surface dimensions.

[2] Face-to-face dimensions on tees and crosses are double the center-to-face dimensions given in the above table.

NOTE.—Center-to-flange-edge dimensions are obtained by subtracting ¼ in. from center-to-contact-surface dimensions.

Standard Specifications for GRAY-IRON CASTINGS FOR VALVES, FLANGES, AND PIPE FITTINGS

ASTM A126-42

Abstracted[1]

These specifications cover gray iron for castings such as valve bodies, fittings, and flanges, including parts to be assembled into valves. Three grades of material are included: a regular gray iron and two grades of higher strength gray iron.

Chemical Composition.—Drillings taken from test ingots, broken test specimens, or castings shall conform to the following requirements as to chemical composition: sulphur, not over 0.12 per cent; phosphorus, not over 0.75 per cent.

Physical Properties.—The physical properties of cast-to-size bars shall conform to the following minimum requirements:

Class	Tensile strength, psi	Transverse test[1]	
		Load at center, lb	Deflection at center, in.
Class A, regular gray iron.........	21,000	2,200	0.10
Class B, higher-strength gray iron..	31,000	3,300	0.12
Class C, high-test cast iron........	41,000	4,000	0.12

[1] Transverse test is optional.

Detailed provisions are given in ASTM A126 to ensure that the tension test specimens and the transverse test bar, if required, are cast under conditions representative of those obtaining during production of the actual castings. One mold containing one or more tension test specimens and, when desired, one transverse specimen is required to be poured at least twice a day from each melt from which castings are made under these specifications.

Transverse specimens are cast to nominal dimensions of 1.20 in. in diameter and 13 in. long. Correction factors are included to take into account variations in diameter of the cast specimen from the specified diameter.

Tension specimens are cast to give a diameter 1⅛ in. at the breaking section. The over-all length of the specimen is approximately 7½ in. Specimens are tested without machining at the breaking section where cross-sectional area as cast is required to be within ±5 per cent of 1 sq. in. The ultimate stress is calculated on the minimum cross section.

Workmanship and Finish.—The castings shall be sound, clean, free from sand, of workmanlike finish, and soft enough to machine well.

Certification.—Upon request of purchaser, the manufacturer shall be prepared to certify that his product conforms to the requirements of these specifications.

[1] For complete specifications, reference may be made to ASTM A126 (see note, p. 369).

American Standard for STEEL PIPE FLANGES AND FLANGED FITTINGS ASA B16e-1939 (includes Addendum B16e.4-1940) Pressure-temperature Ratings Revised in Accordance with American War Standard ASA B16e5-1943

Abstracted[1]

These standards are known as the American 150-, 300-, 400-, 600-, 900-, 1,500-, and 2,500-lb Steel Flange Standards. The pressure-temperature ratings shown in Tables CV to CVIII have been revised in accordance with the American War Standard, ASA B16e.5. These ratings apply to steel pipe flanges and flanged fittings, as given in ASA B16e, and in addition to flanged and welding-end valves with the connecting end dimensions of the valve body conforming to ASA B16e.

Two groups of flange facings are recognized in establishing ratings: "ring-joint" and "other than ring-joint" facings. The ratings of the latter group are approximately 80 per cent of those for ring-joint facings. In applying these ratings to raised-face, lapped, and large male-female facings, when used with flat solid metal gaskets, it is necessary to restrict the gasket contact area to that of the large-tongue and groove-gasket contact area.

The revised ratings make possible a greatly extended use of carbon steel at temperatures below the primary rating temperatures. (Although designed for war emergency use, these ratings in the temperature ranges, where major changes were made, are so well supported by experience that they have been reproduced here in preference to the original ASA B16e ratings.)

Marking.—Each product conforming to this standard shall be marked in accordance with requirements of MSS Standard Practice, SP 25, abstracted on page 687.

Material.—The ratings apply to the steels listed below Tables CV to CVIII inclusive and to such others as may be authorized by the ASTM and the ASME Boiler Code Committee. With the exception of ASTM A27, all these specifications are abstracted in this chapter (Chap. IV). When flanges, fittings, and valves will be subject to fusion welding, the carbon content shall not exceed 0.35 per cent. The steels listed are to be in accordance with the latest edition of the ASTM Specifications for such steels.

The dimensional standards for bolting material including bolts, bolt-studs, washers, and nuts are based on material equal to that given in ASTM Specification A96, abstracted on page 537. Carbon-steel bolts and bolt-studs, conforming to ASTM Specification A107, may be used for steam pressures up to 250 psi and steam and water temperatures up to 450 F. For pressures and temperatures above these limits, the bolting material shall conform to ASTM Specification A96 abstracted on page 537 or Specification A193, abstracted on page 539. For temperatures in excess of 750 to 850 F, it is recommended that bolting conform

[1] For complete standards, reference may be made to ASA B16e and B16e.5, copies of which may be obtained from the American Standards Association, 29 West 39th St., New York 18, N.Y.

to ASTM Specification A193 (see page 530) which is intended to cover metal temperatures from 750 to 1100 F. Nuts shall be of carbon or alloy steel. Chemical requirements and Brinell hardness numbers for carbon-steel nuts suitable for both low-temperature and high-temperature service are given in ASTM Specification A194, abstracted on page 544. Washers for use under nuts shall be of forged or rolled steel.

Metal Thickness.—The minimum metal thicknesses given in the dimensional tables of this standard are based on an allowable stress of 7,000 psi using the ASME Boiler Construction Code's modification of Barlow's formula[1] for cylindrical sections and adding 50 per cent to the thickness thus determined to compensate for the shape of the fittings.

Facings.—The raised face shall be the standard facing for fittings, valves, and companion flanges. It shall be machined to dimension *R* given in Table CX and with the following height of raised face:

A 1/16-in. raised face, included in the minimum flange thickness, is provided for the 150- and 300-lb standards. A 1/4-in. raised face, added to the minimum flange thickness, is provided for standards 400 to 2,500 lb, inclusive. The outside diameter of the raised face for all pressure standards for a given size is the same.

Dimensions for tongue-groove and male-female types of joint facing also are given in Table CX. The bottom or contact surfaces of the groove and female facing shall be in the same plane as the edge of the flange. The projecting contact surfaces, *i.e.*, the tongues and males, shall project 1/4 in. beyond the edge of the flange. (For dimensions of Sarlun and modified Sargol facings, see Table CXXXVIII, page 680.)

The outside diameters of raised face, lapped (Van Stoned), large tongue, and large male types of joint facing are the same, being the minimum diameter considered practical for the lapped type of joint. The outside diameters of groove and female shall be 1/16 in. larger than the corresponding outside diameters of tongue and male, and the inside diameters of the groove shall be 1/16 in. smaller than the corresponding inside diameters of the tongue.

A tolerance of ± 0.016 in. (1/64 in.) is allowed on the inside and outside diameters of all facings. (See Table CX for dimensions.)

The 1/16-in. raised face, if applied to 400-, 600-, 900-, 1,500-, and 2,500-lb standard flanges shall not be cut from the minimum flange thicknesses specified in the respective standards, but shall be provided by facing down the 1/4-in. raised face.

Raised facings higher than 1/16 in. and tongue-groove and male-female facings may be specially required for 150- and 300-lb flanged fittings and flanges, and when so required shall be furnished as follows: (*a*) no metal shall be cut from the minimum flange thickness specified in the standard; (*b*) the full flange thickness shall be first provided and then the raised face, tongue, or male shall be added thereto; (*c*) in the case of groove and female faces, the flange shall be built to full minimum thickness and sufficient metal added thereto so that the bottom or contact face of the groove or female is in the same plane with the face of flange edge of a full thickness flange.

Ring-joint Facings.—The dimensions of ring- and groove-joint facings incorporated in this standard are identical with those given in API Standard 5G3

[1] For discussion of Barlow's formula, see page 41. For ASME Boiler Construction Code's modified Barlow's formula, see page 42.

TABLE CV.—PRESSURE-TEMPERATURE RATINGS FOR STEEL PIPE FLANGES, FLANGED FITTINGS, AND VALVES WITH DIMENSIONS ACCORDING TO ASA B16e-1939
(Table I, ASA, B16e5-1943)
Material: Carbon steels[1]
Facing: Other than ring joint

Fluid	Primary service pressure ratings	150	300	400	600	900	1,500	2,500
	Hydrostatic shell test pressures	350	900	1,200	1,800	2,400	4,200	7,200
	Service temperatures, deg F	Maximum, nonshock, service pressure ratings at temperatures from 100 to 1000 F						
Water, Steam, Oil	100	230	600	800	1,200	1,800	3,000	5,000
	150	220	590	785	1,180	1,770	2,950	4,915
	200	210	580	770	1,160	1,740	2,900	4,830
	250	200	570	760	1,140	1,710	2,850	4,750
	300	190	560	740	1,120	1,680	2,800	4,660
	350	180	550	725	1,095	1,645	2,740	4,565
	400	170	540	710	1,075	1,615	2,690	4,475
	450	160	525	700	1,050	1,580	2,630	4,380
	500	**150**	500	665	1,000	1,500	2,500	4,165
	550	140	475	630	950	1,420	2,370	3,950
	600	130	445	590	890	1,330	2,220	3,700
	650	120	415	550	830	1,240	2,070	3,450
	700	110	380	500	760	1,140	1,900	3,160
	750	100	340	450	680	1,020	1,700	2,830
	800	92	**300**	**400**	**600**	**900**	**1,500**	**2,500**
	850	82	245	300	490	740	1,230	2,050
	900	70[2]	210	280	420	630	1,050	1,750
	950	55[2]	165	220	330	495	825	1,375
	1000	40[2]	120	160	240	360	600	1,000

[1] Carbon-steel castings: ASTM Spec A27 Grades B and B2 (screwed and lapped flanges only).
ASTM Spec A95 (see p. 680).
ASTM Spec A216 Grades WCA and WCB (see p. 685).
Carbon-steel forgings: ASTM Spec A105 Grades I and II (see p. 530).
ASTM Spec A181 Classes I and II (for 150 and 300 series only, see p. 529).

[2] Pressure ratings for temperatures 900 to 1000 F, inclusive, of 150 series are for oil service only.

For temperatures above [illegible] F, consideration should be given to scaling due to oxidation.

Above table is revision of Tables 6 and 7, American Standard B16e-1939.

Primary service pressure ratings are in boldface type.

All pressures are in pounds per square inch gage. Temperatures and pressures listed are maximum internal fluid temperatures and pressures at flange.

All tests shall be made with water at a temperature not to exceed 125 F.

Raised-face, lapped, and large male-and-female facings, when used with flat solid metal gaskets, are permitted to be given the pressure-temperature ratings of this American War Standard only when the gasket contact area is not greater than the large tongue-and-groove gasket contact area.

[illegible] flanges, fittings, and valves will be subject to fusion welding, the carbon content shall [illegible] 0.35 per cent.

TABLE CVI.—PRESSURE-TEMPERATURE RATINGS FOR STEEL PIPE FLANGES, FLANGED FITTINGS, AND VALVES WITH DIMENSIONS ACCORDING TO ASA B16e-1939
(Table II, ASA B16e5-1943)
Material: Carbon steels[1]
Facing: Ring joint
Ratings apply also to valves and fittings with welding ends

Fluid	Primary service pressure ratings	150	300	400	600	900	1,500	2,500
	Hydrostatic shell test pressures	350	900	1,200	1,800	2,400	4,200	7,200
	Service temperatures, deg F	Maximum, nonshock, service pressure ratings at temperatures from 100 to 1000 F						
Water, Steam, Oil	100	275	720	960	1,440	2,160	3,600	6,000
	150	255	710	945	1,420	2,130	3,550	5,915
	200	240	700	930	1,400	2,100	3,500	5,830
	250	225	690	920	1,380	2,070	3,450	5,750
	300	210	680	910	1,365	2,050	3,415	5,690
	350	195	675	900	1,350	2,025	3,375	5,625
	400	180	665	890	1,330	2,000	3,330	5,550
	450	165	660	875	1,320	1,975	3,295	5,490
	500	**150**	625	835	1,250	1,875	3,125	5,210
	550	140	590	790	1,180	1,775	2,955	4,925
	600	130	555	740	1,110	1,660	2,770	4,620
	650	120	515	690	1,030	1,550	2,580	4,300
	700	110	470	635	940	1,410	2,350	3,920
	750	100	425	575	850	1,275	2,125	3,550
	800	92	365	490	730	1,100	1,830	3,050
	850	82	**300**	**400**	**600**	**900**	**1,500**	**2,500**
	900	70[2]	210	280	420	630	1,050	1,750
	950	55[2]	165	220	330	495	825	1,375
	1000	40[2]	120	160	240	360	600	1,000

[1] Carbon-steel castings: ASTM Spec A27 Grades B and B2 (screwed and lapped flanges only).
ASTM Spec A95.
ASTM Spec A216 Grades WCA and WCB.
Carbon-steel forgings: ASTM Spec A105 Grades I and II.
ASTM Spec A181 Classes I and II (for 150 and 300 series only).

[2] Pressure ratings for temperatures 900 to 1000 F, inclusive, of 150 series are for oil service only.

For temperatures above 950 F, consideration should be given to scaling due to oxidation.

Above table is revision of Tables 8 and 9, American Standard B16e-1939.

Primary service pressure ratings are in boldface type.

All pressures are in pounds per square inch gage. Temperatures and pressures listed are maximum internal fluid temperatures and pressures at flange.

All tests shall be made with water at a temperature not to exceed 125 F.

For series 300, [illegible] pressure-temperature ratings are the same as the ratings given in Table CV for the 300 series up to 900 F. For 900 F and above, the pressure-temperature ratings are 85 per cent of the ratings for the 300 series in above Table CVI.

When flanges, fittings, and valves will be subject to fusion welding, the carbon content shall not exceed 0.35 per cent.

TABLE CVII.—PRESSURE-TEMPERATURE RATINGS FOR STEEL PIPE FLANGES, FLANGED FITTINGS, AND VALVES WITH DIMENSIONS ACCORDING TO ASA B16e-1939
(Table III, ASA B16e5-1943)

Material: Carbon-molybdenum steels[1] and equivalent alloy steels
Facing: Other than ring-joint

Fluid	Primary service pressure ratings	300	400	600	900	1,500	2,500
	Hydrostatic shell test pressures	900	1,200	1,800	2,400	4,200	7,200
	Service temperatures, deg F	Maximum, nonshock, service pressure ratings at temperatures from 100 to 1000 F					
Water, Steam, Oil	100	600	800	1,200	1,800	3,000	5,000
	150	590	785	1,180	1,770	2,950	4,915
	200	580	770	1,160	1,740	2,900	4,830
	250	570	760	1,140	1,710	2,850	4,750
	300	560	740	1,120	1,680	2,800	4,660
	350	550	725	1,095	1,645	2,740	4,565
	400	540	710	1,075	1,615	2,690	4,475
	450	525	700	1,050	1,580	2,630	4,380
	500	500	665	1,000	1,500	2,500	4,165
	550	475	630	950	1,420	2,370	3,950
	600	445	590	890	1,330	2,220	3,700
	650	415	550	830	1,240	2,070	3,450
	700	380	500	760	1,140	1,900	3,160
	750	360	475	720	1,080	1,800	2,995
	800	340	450	680	1,020	1,700	2,830
	850	320	425	640	960	1,600	2,665
	900	**300**	**400**	**600**	**900**	**1,500**	**2,500**
	950	265	350	530	795	1,325	2,205
	1000	190	250	380	570	950	1,580

[1] Carbon-molybdenum steel castings: ASTM Spec A157 Grade C1.
ASTM Spec A217 Grades WC1 and WC2.
Carbon-molybdenum steel forgings: ASTM Spec A182 Grade F1.

For temperatures above 950 F, consideration should be given to scaling due to oxidation.
Above table is revision of Table 10, American Standard B16e-1939.
Primary service pressure ratings are in boldface type.
All pressures are in pounds per square inch gage. Temperatures and pressures listed are maximum [illegible] temperatures and pressures at flange.
All tests shall be made with water at a temperature not to exceed 125 F.
Raised-face, lapped, and large male-and-female facings, when used with flat solid-metal gaskets, are permitted to be given the pressure-temperature ratings of this American War Standard only when the gasket contact area is not greater than the large tongue-and-groove gasket contact area.

TABLE CVIII.—PRESSURE-TEMPERATURE RATINGS FOR STEEL PIPE FLANGES, FLANGED FITTINGS, AND VALVES WITH DIMENSIONS ACCORDING TO ASA B16e-1939
(Table IV, ASA B16e5-1943)

Material: Carbon-molybdenum steels[1] and equivalent alloy steels
Facing: Ring joint
Ratings apply also to valves and fittings with welding ends

Fluid	Primary service pressure ratings.	300	400	600	900	1,500	2,500
	Hydrostatic shell test pressures. .	900	1,200	1,800	2,400	4,200	7,200
	Service temperatures, deg F	Maximum, nonshock, service pressure ratings at temperatures from 100 to 1100 F					
Water Steam Oil	100	720	960	1,400	2,160	3,600	6,000
	150	710	945	1,420	2,130	3,550	5,915
	200	700	930	1,400	2,100	3,500	5,830
	250	690	920	1,380	2,070	3,450	5,750
	300	680	910	1,365	2,050	3,415	5,690
	350	675	900	1,350	2,025	3,375	5,625
	400	665	890	1,330	2,000	3,330	5,550
	450	660	875	1,320	1,975	3,295	5,490
	500	625	835	1,250	1,875	3,125	5,210
	550	590	790	1,180	1,775	2,955	4,925
	600	555	740	1,110	1,660	2,770	4,620
	650	515	690	1,030	1,550	2,580	4,300
	700	470	635	940	1,410	2,350	3,920
	750	425	575	850	1,275	2,125	3,550
	800	375	500	750	1,125	1,875	3,125
	850	350	475	700	1,050	1,750	2,925
	900	325	425	650	975	1,625	2,700
	950	**300**	**400**	**600**	**900**	**1,500**	**2,500**
	1000	230	310	470	700	1,170	1,950
All fluids except steam	1050	135	180	270	405	675	1,125
	1100	90	120	180	270	450	750

[1] Carbon-molybdenum steel castings: ASTM Spec A157 Grade C1.
ASTM Spec A217 Grades WC1 and WC2.
Carbon-molybdenum steel forgings: ASTM Spec A182 Grade F1.

For temperatures above 950 F, consideration should be given to scaling due to oxidation.
Above table is revision of Table 11, American Standard B16e-1939.
Primary service pressure ratings are in boldface type.
All pressures are in pounds per square inch gage. Temperatures and pressures listed are maximum internal fluid temperatures and pressures at flange.
All tests shall be made with water at a temperature not to exceed 125 F.
For series 300, where ring groove is cut into the basic flange thickness, the pressure-temperature ratings are the same as the ratings given in Table CVII for the 300 series up to and including 1000 F. Above 1000 F the pressure-temperature ratings are 85 per cent of the ratings for the 300 series in above Table CVIII.

(see tables herewith for dimensions). It is recommended that gaskets for ring joints be of material softer than the flange faces.

Bolting.—Drilling templates are in multiples of four so that the fittings may be made to face in any quarter. Bolt holes straddle the centerlines. Bolt holes are drilled $\frac{1}{8}$ in. larger in diameter than the nominal size of bolt.

Alloy-steel bolt-studs, threaded at both ends or full length, or bolts with hexagon heads, semifinished and in accordance with American Standard Heavy dimensions (ASA B18.2) may be used and shall be equipped with semifinished nuts in accordance with American Standard Heavy dimensions (ASA B18.2) (see page 550). Alloy-steel bolt-studs with a nut at each end are recommended for high-temperature service. Carbon-steel bolts may have American Standard Regular Unfinished Square Heads or American Standard Heavy Unfinished Hexagonal Heads and shall be equipped with American Standard Heavy Unfinished Hexagon Nuts (ASA B18.2).

All carbon-steel bolts, bolt-studs, and accompanying nuts shall be threaded in accordance with the American Standard for Screw Threads, ASA B1.1 Coarse Thread Series (see page 546). All alloy-steel bolts, bolt-studs, and accompanying nuts shall be threaded in accordance with the American Standard for Screw Threads, ASA B1.1, sizes 1 in. in diameter and smaller with the Coarse-Thread Series, and $1\frac{1}{8}$ in. in diameter and larger with the 8-Pitch-Thread Series. See also ASA B1.4 abstracted on page 548.

Allowable working fiber stress, considering internal allowable working pressure only, in bolting material for valve bonnet flanges, clean-out flanges, etc., shall not exceed 9,000 psi, assuming the pressure to act upon an area circumscribed by the periphery of the outside of the surface.

The [illegible]lar flange bolting is based on a stress not to exceed 7,000 psi assuming the pressure to act upon an area circumscribed by the outside diameter of raised face, etc., Column *R*, Table CX.

Spot Facing.—Regular flanged fittings and flanged valves shall be spot-faced or back-faced parallel to the flanged face. Metal removed in spot-facing or back-facing shall not reduce the thickness of the flange below the minimum given in the table. Flanges in sizes 18 in. and smaller shall be spot-faced or back-faced to the specified thickness of the flange (minimum) with a plus tolerance of [illegible] in., and flanges in sizes over 18 in. shall have a plus tolerance of $\frac{3}{16}$ in.

When back-facing the flange, the fillet between the body and the flange shall approximate closely the arc of a circle of generous radius.

Forged steel flanges do not require spot facing if the back of the flange is parallel to the face, provided that after being faced on the contact side they are not oversize more than the following: for sizes 18 in. and smaller, $\frac{1}{8}$ in.; and for sizes over 18 in., $\frac{3}{16}$ in.

For special fittings at least an additional 25 per cent allowance on oversize flange thickness is required.

Fitting Dimensions.—One of the principles of design in this standard is the maintenance of a fixed position for the flange itself with reference to the body of the fitting and the addition of any facing is beyond the outside edge of the flange except for the $\frac{1}{16}$-in. raised face in the 150- and 300-lb standards (see footnote 1 in Table CX). For dimensions, see Tables CIX to CXXXVI.

An inspection limit of $\pm\frac{1}{32}$ in. shall be allowed on all center-to-contact-surface dimensions for sizes up to and including 10 in., and $\pm\frac{1}{16}$ in. on sizes larger than 10 in. An inspection limit of $\pm\frac{1}{16}$ in. shall be allowed on all contact-surface-

to-contact-surface dimensions for sizes up to and including 10 in., and $\pm \frac{1}{8}$ in. on sizes larger than 10 in.

Laterals.—The 45-deg laterals of the larger sizes may require additional reinforcement to compensate for the inherent weakness in this shape of casting.

Reducing Fittings.—Reducing fittings shall have some center-to-flange-edge dimensions as those of straight-size fittings of the largest opening.

Side Outlet Fittings.—All side outlet fittings shall have all openings on the intersecting center lines.

Valve Dimensions.—The center-to-face dimensions of flanged-end valves and the center-to-end dimensions of welding-end valves for the various pressures shall be in accordance with the proposed American Standard Face-to-Face Dimensions of Ferrous Flanged- and Welding-end Valves, ASA B16.10 (see pp. 556–560).

Threading of Screwed Flanges.—The flanges shall be threaded for American Standard taper pipe thread, ASA B2.1. The 300-lb and 400-lb flanges shall have threads of sufficient length to take API line pipe thread (see page 1138). The 600-, 900-, 1,500, and 2,500-lb flanges shall have thread lengths longer than those used for the lower pressure in order to satisfy the general requirement for longer threads for the heavier pressures. The thread in flanges shall be gaged with an American Standard plug gage, and the notch shall come flush with the bottom of the counterbore with a tolerance of plus and minus one turn.

Bore of Flanges.—The bore of lapped and slip-on flanges, also counterbore of screwed flanges as given in the tables, is minimum; accordingly, the following plus tolerances have been set up: $\frac{1}{64}$ in., sizes 12 in. and smaller; $\frac{1}{32}$ in., sizes 14 in. and larger.[1]

Blind Flanges.—All blind flanges shall have diameter and thickness as given in the standard. They may be made flat back and shall conform to the materials specifications established for companion flanges.

Welding-neck Flanges.—The materials, facings, spot facings, etc., conform to the requirements given for other flanges, with the additional provision that the carbon content of the steel shall not exceed 0.35 per cent.

The welding end of the hub shall be cylindrical or its outside surface shall have a draft of not more than 6 deg for forging purposes. The length of this cylindrical portion shall be sufficient to ensure a sound weld. In no case, however, shall this cylindrical portion be less than $\frac{1}{4}$ in. long.

Concentric Reducing Welding-neck Flanges.—The distance from the back face of the flange to the end of the hub as well as all other hub dimensions shall agree with the same dimensions for standard flanges of the smaller size. Flange dimensions and drilling templates shall agree with the standard dimensions for the larger size.

Tolerance for Welding-neck Flanges.—

(1) OUTSIDE DIAMETER OF HUB.—Outside diameter of hub at beginning of chamfer

For sizes up to and including 5 in. $+\frac{3}{32}$ in. $-\frac{1}{32}$ in.

For sizes 6 in. and above, $+\frac{5}{32}$ in. $-\frac{1}{32}$ in.

[1] Lapped flanges must swivel on the pipe. Slip-on flanges should be as close a fit on commercial pipe as practical on account of welding. For screwed flanges having counterbore, the clearance must take care of variations in O.D. of pipe in relation to the pipe threads.

(2) THICKNESS OF HUB.—The thickness of hub shall not be less than 87½ per cent of the nominal thickness of the pipe to which the flange is to be attached.

(3) INSIDE DIAMETER OF BORE.—
For sizes up to and including 10 in., ± 1⁄32 in.
For sizes 12 in. to 18 in., inclusive, ± 1⁄16 in.
For sizes over 18 in., + 1⁄8 in. − 1⁄16 in.

(4) OVER-ALL LENGTH OF HUB.—
For sizes up to and including 10 in., ± 1⁄16 in.
For sizes 12 in. and above, ± 1⁄8 in.

Welding Bevel.—The recommended practice for the detail of welding bevel shall be as follows: (see Fig. 15, page 489):

For wall thicknesses 3⁄16 to 3⁄4 in., inclusive,—37½ deg ± 2½ deg straight bevel; land 1⁄16 in. ± 1⁄32 in.

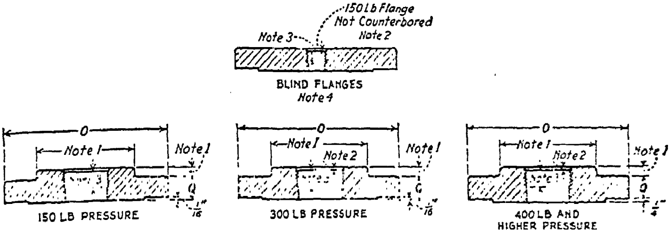
For wall thicknesses greater than 3⁄4 in.—20 deg ± 2½ deg U-bevel, 3⁄16 in. radius; land 1⁄16 in. ± 1⁄32 in.

Welding ends having thicknesses less than 3⁄16 in. shall be prepared with a slight chamfer or square in accordance with manufacturer's practice.

Table CIX.—Reducing Screwed Flanges for 150- to 2,500-lb Pressures

(Table 12, ASA B16 e)

(All dimensions in inches)



Nominal pipe size	Smallest size[1,4] tapping for hub flange	Nominal pipe size	Smallest size[1,4] tapping for hub flange	Nominal pipe size	Smallest size[1,4] tapping for hub flange
1	½	3½	1½	12	3½
1¼	½	4	1½	14	3½
1½	½	5	1½	16	4
2	1	6	2½	18	4
2½	1¼	8	3	20	4
3	1¼	10	3½	24	4

Dimension Q is minimum flange thickness.

Reducing flanges are designated by the size of tapping and the outside diameter of flange.

For dimensions not given, see flange tables.

American Standard for Taper Pipe Threads (ASA B2.1) shall be used in threading the above flanges. The notch of the American Standard taper working plug gage shall come flush with the bottom of the counterbore (at large end of thread) with a manufacturing tolerance of plus or minus one turn.

[1] Minimum diameter of hub and minimum height of hub above the flange are the same as for standard flange one size smaller and this hub is maintained for all other reductions except where blind flanges are used.

[2] Flanges for 150 lb do not have a counterbore. Flanges for 300 lb and higher pressures will have depth of counterbore of ¼ in. for 2-in. and smaller tappings, and ⅜ in. for 2½-in. and larger tappings. The diameter of counterbore is the same as that given in the tables of the corresponding screwed flanges.

[3] Minimum length of effective threads shall be at least equal to dimension T in the respective tables for the corresponding tapping but does not necessarily extend to the face of the flange as in regular flanges.

Example 1.—2½ in. × 12½ in.—300-lb reducing flange (see Table CXV for dimensions).

Tapping	2½ in.	Pipe tap
O	12½ in.	Diameter flange regularly having 6-in. tapping
	1$\frac{7}{16}$ in.	Thickness of standard 6 × 12½-in. flange
$\check{X}$	7 in.	Diameter hub for 5-in. standard flange
$Y - Q$	⅝ in.	Height of hub for 5-in. standard flange.

[4] For tappings smaller than those given in this table, blind flanges may be used.

Example 2.—2 × 12½ in—300-lb reducing flange. Use regular blind flange tapped for 2-in. pipe.

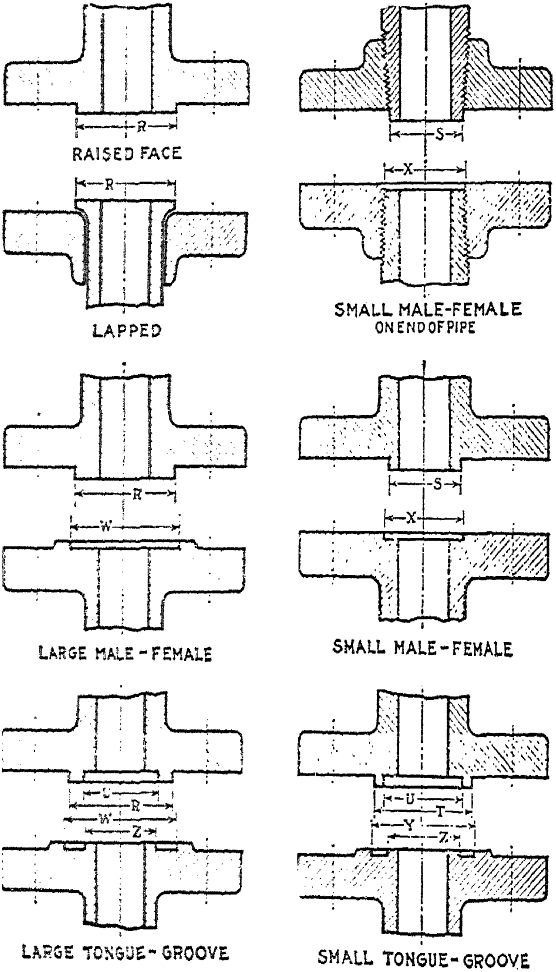


Fig. 40.—Typical flange facings (see Table CX, p. 637, for dimensions).

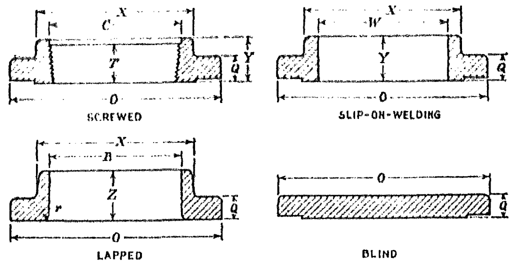
TABLE CX.—FACING DIMENSIONS FOR THE 150-, 300-, 400-, 600-, 900-, 1,500-, AND 2,500-LB FLANGES*
(All dimensions in inches)

Nominal pipe size	Outside diameter[3]: Raised face, lapped, large male, and large tongue[5] R	Outside diameter[3]: Small male[4,5] S	Outside diameter[3]: Small tongue[5] T	I.D. of large and small tongue[3,5] U	Outside diameter[3]: Large female and large groove[5] W	Outside diameter[3]: Small female[4,5] X	Outside diameter[3]: Small groove[5] Y	I.D. of large and small groove[3,5] Z	Height of raised face: 150 and 300 lb. standards[1]	Height of raised face: Large and small male, and tongue,[2] 400, 600, 900, 1,500, 2,500 lb	Depth of groove or female
1/2	1 3/8	23/32	1 3/8	1	1 7/16	25/32	1 7/16	15/16	1/16	1/4	3/16
3/4	1 11/16	15/16	1 11/16	1 5/16	1 3/4	1	1 3/4	1 1/4	1/16	1/4	3/16
1	2	1 3/16	1 7/8	1 1/2	2 1/16	1 1/4	1 15/16	1 7/16	1/16	1/4	3/16
1 1/4	2 1/2	1 1/2	2 1/4	1 7/8	2 9/16	1 9/16	2 5/16	1 13/16	1/16	1/4	3/16
1 1/2	2 7/8	1 3/4	2 1/2	2 1/8	2 15/16	1 13/16	2 9/16	2 1/16	1/16	1/4	3/16
2	3 5/8	2 1/4	3 1/4	2 7/8	3 11/16	2 5/16	3 5/16	2 13/16	1/16	1/4	3/16
2 1/2	4 1/8	2 11/16	3 3/4	3 3/8	4 3/16	2 3/4	3 13/16	3 5/16	1/16	1/4	3/16
3	5	3 5/16	4 5/8	4 1/4	5 1/16	3 3/8	4 11/16	4 3/16	1/16	1/4	3/16
3 1/2	5 1/2	3 13/16	5 1/8	4 3/4	5 9/16	3 7/8	5 3/16	4 11/16	1/16	1/4	3/16
4	6 3/16	4 5/16	5 11/16	5 3/16	6 1/4	4 3/8	5 3/4	5 1/8	1/16	1/4	3/16
5	7 5/16	5 3/8	6 13/16	6 5/16	7 3/8	5 7/16	6 7/8	6 1/4	1/16	1/4	3/16
6	8 1/2	6 3/8	8	7 1/2	8 9/16	6 7/16	8 1/16	7 7/16	1/16	1/4	3/16
8	10 5/8	8 3/8	10	9 3/8	10 11/16	8 7/16	10 1/16	9 5/16	1/16	1/4	3/16
10	12 3/4	10 1/2	12	11 1/4	12 13/16	10 9/16	12 1/16	11 3/16	1/16	1/4	3/16
12	15	12 1/2	14 1/4	13 1/2	15 1/16	12 9/16	14 5/16	13 7/16	1/16	1/4	3/16
14 O.D.	16 1/4	13 3/4	15 1/2	14 3/4	16 5/16	13 13/16	15 9/16	14 11/16	1/16	1/4	3/16
16 O.D.	18 1/2	15 3/4	17 5/8	16 3/4	18 9/16	15 13/16	17 11/16	16 11/16	1/16	1/4	3/16
18 O.D.	21	17 3/4	20 1/8	19 1/4	21 1/16	17 13/16	20 3/16	19 3/16	1/16	1/4	3/16
20 O.D.	23	19 3/4	22	21	23 1/16	19 13/16	22 1/16	20 15/16	1/16	1/4	3/16
24 O.D.	27 1/4	23 3/4	26 1/4	25 1/4	27 5/16	23 13/16	26 5/16	25 3/16	1/16	1/4	3/16

[1] Regular facing for 150- and 300-lb steel flanged fittings and companion flange standards is a 1/16-in. raised face included in the minimum flange-thickness dimensions given in Tables CXI to CXIV and CXV to CXVIII respectively. A 1/16-in. raised face may be supplied also on the 400-, 600-, 900-, 1,500-, and 2,500-lb flange standards, but it must be added to the minimum flange thickness. [2] Regular facing for 400-, 600-, 900-, 1,500-, and 2,500-lb flange standards is a 1/4-in. raised face not included in minimum flange-thickness dimensions. [3] Tolerance of 0.016 in. (1/64 in.) is allowed on the inside and outside diameters of all facings. [4] For small male-and-female joints care should be taken in the use of these dimensions to ensure that pipe used is thick enough to permit sufficient bearing surface to prevent the crushing of the gasket. The dimensions apply particularly on lines where the joint is made on the end of the pipe. Screwed companion flanges for small male-and-female joints are furnished with plain face and are threaded with American standard locknut thread. [5] Gaskets for male-female and tongue-groove joints shall cover the bottom of the recess with minimum clearances taking into account the tolerances prescribed in Note 3.

* Table 13, ASA B16e, see Fig. 40.

TABLE CXI. DIMENSIONS OF STEEL FLANGES FOR PRIMARY-SERVICE PRESSURE RATING OF 150 PSI GAGE
(For complete pressure-temperature ratings, see Tables CV to CVIII)
(Table 14, ASA B16 e)
(All dimensions in inches)



Nominal pipe size	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of hub[2]	Diameter of bolt circle	Number of bolts	Size of bolts	Bore: Slip-on welding flange,[3] minimum	Bore: Lapped flange,[3] minimum	Corner radius of bore of lapped flange and pipe	Length through hub: Screwed[1,4,5]	Length through hub: Lapped
	O	Q	X				W	B	r	Y	Z
½	3½	7/16	1 3/16	2⅜	4	½	0.88	0.93	⅛	⅝	⅝
¾	3⅞	½	1½	2¾	4	½	1.09	1.14	⅛	⅝	⅝
1	4¼	9/16	1 15/16	3⅛	4	½	1.38	1.41	⅛	11/16	11/16
1¼	4⅝	⅝	2 5/16	3½	4	½	1.72	1.75	3/16	13/16	13/16
1½	5	11/16	2 9/16	3⅞	4	½	1.97	1.99	¼	⅞	⅞

TABLE CXI.—(*Concluded*)

Nominal pipe size	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of hub[2]	Diameter of bolt circle	Number of bolts	Size of bolts	Bore		Corner radius of bore of lapped flange and pipe	Length through hub	
							Slip-on welding flange,[3] minimum	Lapped flange,[3] minimum		Screwed[1,4,5]	Lapped
	O	Q	X				W	B	r	Y	Z
2	6	3/4	3 1/16	4 3/4	4	5/8	2.44	2.50	5/16	1	1
2 1/2	7	7/8	3 9/16	5 1/2	4	5/8	2.94	3.00	5/16	1 1/8	1 1/8
3	7 1/2	15/16	4 1/4	6	4	5/8	3.56	3.63	3/8	1 3/16	1 3/16
3 1/2	8 1/2	15/16	4 13/16	7	8	5/8	4.06	4.13	3/8	1 1/4	1 1/4
4	9	15/16	5 5/16	7 1/2	8	5/8	4.56	4.63	7/16	1 5/16	1 5/16
5	10	15/16	6 7/16	8 1/2	8	3/4	5.66	5.69	7/16	1 7/16	1 7/16
6	11	1	7 9/16	9 1/2	8	3/4	6.72	6.75	1/2	1 9/16	1 9/16
8	13 1/2	1 1/8	9 11/16	11 3/4	8	3/4	8.72	8.81	1/2	1 3/4	1 3/4
10	16	1 3/16	12	14 1/4	12	7/8	10.88	10.94	1/2	1 15/16	1 15/16
12	19	1 1/4	14 3/8	17	12	7/8	12.88	12.94	1/2	2 3/16	2 3/16
14 O.D.	21	1 3/8	15 3/4	18 3/4	12	1	14.19	14.25	1/2	2 1/4	3 1/8
16 O.D.	23 1/2	1 7/16	18	21 1/4	16	1	16.19	16.25	1/2	2 1/2	3 7/16
18 O.D.	25	1 9/16	19 7/8	22 3/4	16	1 1/8	18.19	18.25	1/2	2 11/16	3 13/16
20 O.D.	27 1/2	1 11/16	22	25	20	1 1/8	20.19	20.25	1/2	2 7/8	4 1/16
24 O.D.	32	1 7/8	26 1/8	29 1/2	20	1 1/4	24.19	24.25	1/2	3 1/4	4 3/8

For the dimensions of reducing screwed flanges see Table CIX.

Blind flanges should be faced as follows: (1) When the center portion of the inside face is raised it need not be faced, but its diameter shall be at least 1 in. smaller than the inside diameter of the fitting given in Table CXIV. (2) When the center portion of the inside face is depressed it need not be faced, and its diameter shall not be greater than the inside diameter of the fitting given in Table CXIV.

[1] The raised face of 1/16 in. is included in thickness of flange, minimum, and in length through hub.

[2] This dimension is for large end of hub, which may be tapered 5 deg for draft.

[3] The dimensions given for the bore of the slip-on welding flanges and the bore of the lapped flange are based on the variation of the outside diameter (oversize) of steel pipe given in ASTM Specifications A-53, A-120, and API Specifications 5-L. The tolerances on the bore of these flanges are 1/32 in. for sizes 10 in. and smaller, and 1/16 in. for sizes 12 in. and larger.

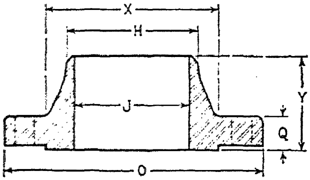
[4] Thread length takes American Standard for Pipe Threads, ASA B2.1. [5] Screwed flanges are threaded without counterbore.

TABLE CXII.—DIMENSIONS OF STEEL WELDING-NECK FLANGES FOR PRIMARY-SERVICE PRESSURE RATING OF 150 PSI GAGE

(For complete pressure-temperature ratings see Tables CV to CVIII)

(Table 15, ASA B16 e)

(All dimensions in inches)



Nominal pipe size	Diameter of flange O	Thickness of flange,[1] minimum Q	Diameter of hub X	Hub diameter, beginning of chamfer[2,3,4] H	Length through hub[1] Y	Inside diameter of pipe schedule 40[3,4] J	Diameter of bolt circle	Number of bolts	Size of bolts
½	3½	7/16	1 3/16	0.84	1⅞	0.62[5]	2⅜	4	½
¾	3⅞	½	1½	1.05	2 1/16	0.82[5]	2¾	4	½
1	4¼	9/16	1 15/16	1.32	2 3/16	1.05[5]	3⅛	4	½
1¼	4⅝	⅝	2 5/16	1.66	2¼	1.38[5]	3½	4	½
1½	5	11/16	2 9/16	1.90	2 7/16	1.61[5]	3⅞	4	½
2	6	¾	3 1/16	2.38	2½	2.07[5]	4¾	4	⅝
2½	7	⅞	3 9/16	2.88	2¾	2.47[5]	5½	4	⅝
3	7½	15/16	4¼	3.50	2¾	3.07[5]	6	4	⅝
3½	8½	15/16	4 13/16	4.00	2 13/16	3.55[5]	7	8	⅝
4	9	15/16	5 5/16	4.50	3	4.03[5]	7½	8	⅝
5	10	15/16	6 7/16	5.56	3½	5.05[5]	8½	8	¾
6	11	1	7 9/16	6.63	3½	6.07[5]	9½	8	¾
8	13½	1⅛	9 11/16	8.63	4	7.98[5]	11¾	8	¾
10	16	1 3/16	12	10.75	4	10.02[5]	14¼	12	⅞
12	19	1¼	14⅜	12.75	4½		17	12	⅞
14 O.D.	21	1⅜	15¾	14.00	5	To be specified by purchaser[6]	18¾	12	1
16 O.D.	23½	1 7/16	18	16.00	5		21¼	16	1
18 O.D.	25	[illegible]	19⅞	18.00	5½		22¾	16	1⅛
20 O.D.	27½	[illegible]	22	20.00	5 11/16		25	20	1⅛
24 O.D.	32	[illegible]	[illegible]	24.00	6		29½	20	1¼

See Welding Bevel and Reducing Welding-neck Flanges, in the introductory notes, p. 634.

[1] A raised face of 1/16 in. is included in thickness of flange minimum and in length through hub.

[2] The outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg.

[3] Dimensions H and J correspond to the outside and inside diameters of pipe as given in ASA B36.10, Schedule 40.

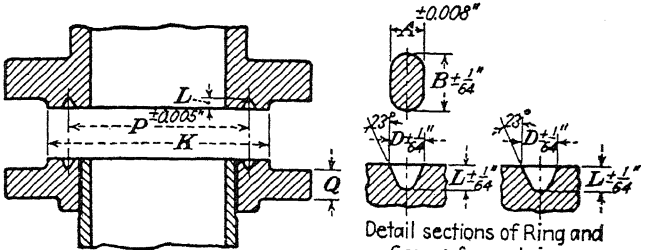
[4] See Tolerance for Welding-neck Flanges, in the introductory notes, p. 633.

[5] These diameters are identical with the diameters of what was formerly designated as "standard-weight pipe" of the corresponding sizes.

[6] If specified by the purchaser, flanges may be furnished and used with inside diameters corresponding to those of the pipe to which they are to be welded.

Table CXIII.—Facing Dimensions for 150-lb Ring-joint Flanges

(Table 18, ASA B16 e)
(All dimensions in inches)



Detail sections of Ring and Groove for oval rings

Nominal pipe size	Pitch diameter of ring and groove *P*	Width of ring *A*	Height of oval ring *B*	Width of groove *D*	Depth of groove[1] *L*	Diameter of raised face for ring joint *K*	Ring numbers[2]	Approximate distance between flanges of ring joints when ring is compressed[3]
1	1⅞	5/16	9/16	11/32	¼	2½	R 15	5/32
1¼	2¼	5/16	9/16	11/32	¼	2⅞	R 17	5/32
1½	2 9/16	5/16	9/16	11/32	¼	3¼	R 19	5/32
2	3¼	5/16	9/16	11/32	¼	4	R 22	5/32
2½	4	5/16	9/16	11/32	¼	4¾	R 25	5/32
3	4½	5/16	9/16	11/32	¼	5¼	R 29	5/32
3½	5 3/16	5/16	9/16	11/32	¼	6 1/16	R 33	5/32
4	5⅞	5/16	9/16	11/32	¼	6¾	R 36	5/32
5	6¾	5/16	9/16	11/32	¼	7⅝	R 40	5/32
6	7⅝	5/16	9/16	11/32	¼	8⅝	R 43	5/32
8	9¾	5/16	9/16	11/32	¼	10¾	R 48	5/32
10	12	5/16	9/16	11/32	¼	13	R 52	5/32
12	15	5/16	9/16	11/32	¼	16	R 56	5/32
14 O.D.	15⅝	5/16	9/16	11/32	¼	16¾	R 59	⅛
16 O.D.	17⅞	5/16	9/16	11/32	¼	19	R 64	⅛
18 O.D.	20⅜	5/16	9/16	11/32	¼	21½	R 68	⅛
20 O.D.	22	5/16	9/16	11/32	¼	23½	R 72	⅛
24 O.D.	26½	5/16	9/16	11/32	¼	28	R 76	⅛

Dimension Q is minimum flange thickness.

The use of grooves with flat bottoms is permitted and is optional with the manufacturer.

The corner radius for flat bottom grooves r is 1/32 in. for grooves used with rings having width of 5/16 in. This r is the maximum.

[1] The depth of groove is added to the minimum flange thickness, except in sizes 1 to 1½ in., inclusive, where the groove cuts 1/8 in. into the minimum flange thickness.

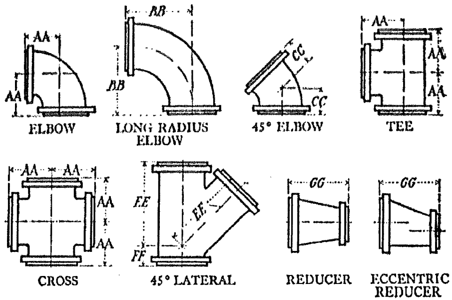
[2] For dimensions of rings, see Table CXXXVII.

[3] For calculating the "laying length" of fittings with ring joints the space dimensions given in the table must be added.

TABLE CXIV. DIMENSIONS OF STEEL-FLANGED FITTINGS WITH PROJECTING FACES FOR PRIMARY-SERVICE PRESSURE RATING OF 150 PSI GAGE

(For complete pressure-temperature ratings see Tables CV to CVIII) (Table 16, ASA B16 e)

(All dimensions in inches)



Nominal pipe size	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of bolt circle	Number of bolts	Size of bolts	1/16-in. raised face					
								Center to contact surface of raised face elbow, tee, and cross[1,2,3] AA	Center to contact surface of raised face, long-radius ell[1,2,3] BB	Center to contact surface of raised face, 45-deg ell[1,2,3] CC	Long center to contact surface of raised face, lateral[1,2,3] EE	Short center to contact surface of raised face, lateral[1,2,3] FF	Contact surface to contact surface reducer[1,2] GG
1	1	1/4	4 1/4	7/16	3 1/8	4	1/2	3 1/2	5	1 3/4	5 3/4	1 3/4	
1 1/4	1 1/4	1/4	4 5/8	1/2	3 1/2	4	1/2	3 3/4	5 1/2	2	6 1/4	1 3/4	
1 1/2	1 1/2	1/4	5	9/16	3 7/8	4	1/2	4	6	2 1/4	7	2	

TABLE CXIV.—(*Concluded*)

Nominal pipe size	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of bolt circle	Number of bolts	Size of bolts	$\frac{1}{16}$-in. raised face: Center to contact surface of raised face elbow, tee, and cross[1,2,3] *AA*	Center to contact surface of raised face, long-radius ell[1,2,3] *BB*	Center to contact surface of raised face, 45-deg ell[1,2,3] *CC*	Long center to contact surface of raised face, lateral[1,2,3] *EE*	Short center to contact surface of raised face, lateral[1,2,3] *FF*	Contact surface to contact surface reducer[1,2] *GG*
2	2	¼	6	⅝	4¾	4	⅝	4½	6½	2½	8	2½	5
2½	2½	¼	7	$\frac{11}{16}$	5½	4	⅝	5	7	3	9½	2½	5½
3	3	¼	7½	¾	6	4	⅝	5½	7¾	3	10	3	6
3½	3½	¼	8½	$\frac{13}{16}$	7	8	⅝	6	8½	3½	11½	3	6½
4	4	¼	9	$\frac{15}{16}$	7½	8	⅝	6½	9	4	12	3	7
5	5	$\frac{9}{32}$	10	$\frac{15}{16}$	8½	8	¾	7½	10¼	4½	13½	3½	8
6	6	$\frac{9}{32}$	11	1	9½	8	¾	8	11½	5	14½	3½	9
8	8	$\frac{5}{16}$	13½	1⅛	11¾	8	¾	9	14	5½	17½	4½	11
10	10	$\frac{11}{32}$	16	1$\frac{3}{16}$	14¼	12	⅞	11	16½	6½	20½	5	12
12	12	⅜	19	1¼	17	12	⅞	12	19	7½	24½	5½	14
14 O.D.	13¼	$\frac{13}{32}$	21	1⅜	18¾	12	1	14	21½	7½	27	6	16
16 O.D.	15¼	$\frac{7}{16}$	23½	1$\frac{7}{16}$	21¼	16	1	15	24	8	30	6½	18
18 O.D.	17¼	$\frac{15}{32}$	25	1$\frac{9}{16}$	22¾	16	1⅛	16½	26½	8½	32	7	19
20 O.D.	19¼	½	27½	1$\frac{11}{16}$	25	20	1⅛	18	29	9½	35	8	20
24 O.D.	23¼	$\frac{9}{16}$	32	1⅞	29½	20	1¼	22	34	11	40½	9	24

True Y (see Fig. 38). The dimensions of this fitting should be the same as those of the 90-deg elbow with a back outlet in which the dimension of the back outlet contact face to the intersecting center lines is the same as dimension *FF* of the 45-deg lateral. The contact face to intersecting center lines of each Y outlet is the same as dimension *AA* of the elbow.

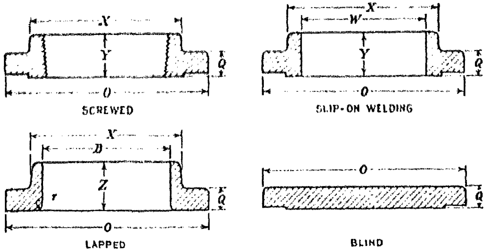
[1] A raised face of $\frac{1}{16}$ in. is provided on the flange of each opening of these fittings and is included in (1) thickness of flange, minimum, and (2) center-to-contact surface and (3) contact-surface-to-contact-surface dimensions.

[2] Where facings other than the $\frac{1}{16}$-in. raised face are used, the center-to-contact-surface dimensions shall remain unchanged, and new center-to-contact-surface dimensions shall be established to suit the facings used.

[3] Reducing fittings shall have the same center-to-contact-surface dimensions as those of straight-size fittings of the largest opening.

The center-to-end dimensions of ring-joint fittings are obtained by adding the depth of groove of ¼ in. to above center-to-contact-surface dimensions.

TABLE CXV. DIMENSIONS OF STEEL FLANGES FOR PRIMARY-SERVICE PRESSURE RATING OF 300 PSI GAGE
(For complete pressure-temperature ratings see Tables CV to CVIII)
(Table 20, ASA B16 e)
(All dimensions in inches)



Nominal pipe size	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of hub[2]	Diameter of bolt circle	Number of bolts	Size of bolts	Counter-bore screwed flange,[3] minimum	Bore: Slip-on welding flange,[3] minimum	Bore: Lapped flange,[3] minimum	Corner radius of bore of lapped flange and pipe	Length through hub: Screwed[1]	Length through hub: Lapped	Thread length[1,4]
	O	Q	X				C	W	B	r	Y	Z	T
½	3¾	9/16	1½	2⅝	4	½	0.93	0.88	0.93	⅛	⅞	⅞	⅝
¾	4⅝	⅝	1⅞	3¼	4	⅝	1.14	1.09	1.14	⅛	1	1	⅝
1	4⅞	11/16	2⅛	3½	4	⅝	1.41	1.38	1.41	⅛	1 1/16	1 1/16	11/16
1¼	5¼	¾	2½	3⅞	4	⅝	1.75	1.72	1.75	3/16	1 1/16	1 1/16	13/16
1½	6⅛	13/16	2¾	4½	4	¾	1.99	1.97	1.99	¼	1 3/16	1 3/16	⅞

TABLE CXV.—(*Concluded*)

Nominal pipe size	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of hub[2]	Diameter of bolt circle	Number of bolts	Size of bolts	Counter-bore screwed flange,[3] minimum	Bore		Corner radius of bore of lapped flange and pipe	Length through hub		Thread length[1,4]
								Slip-on welding flange,[3] minimum	Lapped flange,[3] minimum		Screwed[1]	Lapped	
	O	*Q*	*X*				*C*	*W*	*B*	*r*	*Y*	*Z*	*T*
2	6 1/2	7/8	3 5/16	5	8	5/8	2.50	2.44	2.50	5/16	1 5/16	1 5/16	1 1/8
2 1/2	7 1/2	1	3 15/16	5 7/8	8	3/4	3.00	2.94	3.00	5/16	1 1/2	1 1/2	1 1/4
3	8 1/4	1 1/8	4 5/8	6 5/8	8	3/4	3.63	3.56	3.63	3/8	1 11/16	1 11/16	1 1/4
3 1/2	9	1 3/16	5 1/4	7 1/4	8	3/4	4.13	4.06	4.13	3/8	1 3/4	1 3/4	1 7/16
4	10	1 1/4	5 3/4	7 7/8	8	3/4	4.63	4.56	4.63	7/16	1 7/8	1 7/8	1 7/16
5	11	1 3/8	7	9 1/4	8	3/4	5.69	5.66	5.69	7/16	2	2	1 11/16
6	12 1/2	1 7/16	8 1/8	10 5/8	12	3/4	6.75	6.72	6.75	1/2	2 1/16	2 1/16	1 13/16
8	15	1 5/8	10 1/4	13	12	7/8	8.75	8.72	8.81	1/2	2 7/16	2 7/16	2
10	17 1/2	1 7/8	12 5/8	15 1/4	16	1	10.88	10.88	10.94	1/2	2 5/8	3 3/4	2 3/16
12	20 1/2	2	14 3/4	17 3/4	16	1 1/8	12.94	12.88	12.94	1/2	2 7/8	4	2 3/8
14 O.D.	23	2 1/8	16 3/4	20 1/4	20	1 1/8	14.19	14.19	14.25	1/2	3	4 3/8	2 1/2
16 O.D.	25 1/2	2 1/4	19	22 1/2	20	1 1/4	16.19	16.19	16.25	1/2	3 1/4	4 3/4	2 11/16
18 O.D.	28	2 3/8	21	24 3/4	24	1 1/4	18.19	18.19	18.25	1/2	3 1/2	5 1/8	2 3/4
20 O.D.	30 1/2	2 1/2	23 1/8	27	24	1 1/4	20.19	20.19	20.25	1/2	3 3/4	5 1/2	2 7/8
24 O.D.	36	2 3/4	27 5/8	32	24	1 1/2	24.19	24.19	24.25	1/2	4 3/16	6	3 1/4

For the dimensions of reducing screwed flanges, see Table CIX.

Blind flanges should be faced as follows: (1) When the center portion of the inside face is raised it need not be faced, but its diameter shall be at least 1 in. smaller than the inside diameter of the fitting given in Table CXVII; (2) when the center portion of the inside face is depressed, it need not be faced, and its diameter shall not be greater than the inside diameter of the fitting given in Table CXVII.

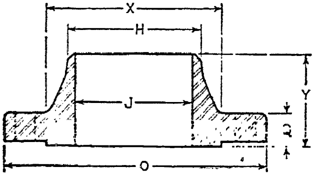
[1] The raised face of 1/16 in. is included in thickness of flange, minimum, length through hub, and thread length.

[2] This dimension is for large end of hub, which may be tapered 5 deg for draft.

[3] The dimensions given for the counterbore of the screwed flange, the bore of the slip-on welding flanges, and the bore of the lapped flange are based on the variation of the outside diameter (oversize) of steel pipe given in ASTM Specifications A-53, A-120, and API Specification 5-L. The tolerances on the bore of these flanges are +1/32 in. for sizes 10 in. and smaller, and +1/16 in. for sizes 12 in. and larger.

[4] Thread length given takes API line pipe thread. This standard thread is based on the American Standard for Pipe Threads (ASA B2.1), but some sizes are longer, averaging about two threads, with the exception of the 2-in. size which is four threads.

TABLE CXVI.—DIMENSIONS OF STEEL WELDING-NECK FLANGES FOR PRIMARY-SERVICE PRESSURE RATING OF 300 PSI GAGE
(For complete pressure-temperature ratings see Tables CV to CVIII)
(Table 21, ASA B16 e)
(All dimensions in inches)



Nominal pipe size	Diameter of flange *O*	Thickness of flange,[1] minimum *Q*	Diameter of hub *X*	Hub diameter beginning of chamfer[2,3,4] *H*	Length through hub[1] *Y*	Inside diameter of pipe, schedule 40[3,4] *J*	Inside diameter of pipe, schedule 80[3,4] *J*	Diameter of bolt circle	Number of bolts	Size of bolts
½	3¾	9/16	1½	0.84	2[illegible]	0.62[5]	0.55[7]	2[illegible]	4	[illegible]
¾	4⅝	⅝	1⅞	1.05	2[illegible]	0.82[5]	0.74[7]	3[illegible]	4	[illegible]
1	4⅞	11/16	2⅛	1.32	2[illegible]	1.05[5]	0.96[7]	3[illegible]	4	[illegible]
1¼	5¼	¾	2½	1.66	2[illegible]	1.38[5]	1.28[7]	3[illegible]	4	[illegible]
1½	6⅛	13/16	2¾	1.90	2[illegible]	1.61[5]	1.50[7]	4[illegible]	4	[illegible]
2	6½	⅞	3 5/16	2.38	2¾	2.07[5]	1.94[7]	5	8	⅝
2½	7½	1	3 15/16	2.88	3	2.47[5]	2.32[7]	5⅞	8	¾
3	8¼	1⅛	4⅝	3.50	3⅛	3.07[5]	2.90[7]	6⅝	8	¾
3½	9	1 3/16	5¼	4.00	3 3/16	3.55[5]	3.36[7]	7¼	8	¾
4	10	1¼	5¾	4.50	3⅜	4.03[5]	3.83[7]	7⅞	8	¾
5	11	1⅜	7	5.56	3⅞	5.05[5]	4.81[7]	9¼	8	¾
6	12½	1 7/16	8⅛	6.63	3⅞	6.07[5]	5.76[7]	10⅝	12	¾
8	15	1⅝	10¼	8.63	4⅜	7.98[5]	7.63[7]	13	12	⅞
10	17½	1⅞	12⅝	10.75	4⅝	10.02[5]	......	15¼	16	1
12	20½	2	14¾	12.75	5⅛	....	......	17¾	16	1⅛
14 O.D.	23	2⅛	16¾	14.00	5⅝	To be specified by purchaser[6]	To be specified by purchaser[6]	20¼	20	1⅛
16 O.D.	25½	2¼	19	16.00	5¾			22½	20	1¼
18 O.D.	28	2⅜	21	18.00	6¼			24¾	24	1¼
[illegible]	30[illegible]	[illegible]	23[illegible]	[illegible].00	6[illegible]			27	24	1[illegible]
[illegible]	36	[illegible]	27[illegible]	24.00	[illegible]			32	24	1[illegible]

See Welding Bevel and Reducing Welding-neck Flanges in the introductory notes, p. 634.

[1] A [illegible] is included in thickness of flange, minimum, and in length through hub.

[2] The outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg.

[3] Dim[illegible] H and J correspond to the outside and inside diameters of pipe as given in ASA B3[illegible], [illegible] 40 and 80. Purchaser's order must specify which of these two inside diameters is desired.

[4] See Tolerance for Welding-neck Flanges, in the introductory notes, p. 633.

[5] These diameters (0.62 to 10.02) are identical with the diameters of what was formerly designated as "standard-weight pipe" of the corresponding sizes.

[6] These flanges are regularly bored to match inside diameter of Schedule 40 pipe but are bored to Schedule 80 pipe when so ordered.

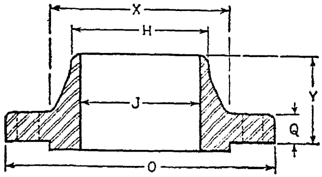
[7] These diameters (0.55 to 7.63) are identical with the diameters of what was formerly designated as "extra-strong pipe" of the corresponding sizes.

Table CXVII.—Dimensions of Steel Welding-neck Flanges for Primary-service Pressure Rating of 400 Psi Gage

(For complete pressure-temperature ratings see Tables CV to CVIII)

(Table 27, ASA B16 e)

(All dimensions in inches)



Nominal pipe size[1]	Diameter of flange *O*	Thickness of flange,[2] minimum *Q*	Diameter of hub *X*	Hub diameter beginning of chamfer[3,4,5] *H*	Length through hub[2] *Y*	Inside diameter of pipe, schedule 40[6] *J*	Inside diameter of pipe, schedule 80[8] *J*	Diameter of bolt circle	Number of bolts	Size of bolts
½	3¾	9/16	1½	0.84	2 1/16	0.62	0.55	2⅝	4	½
¾	4⅝	⅝	1⅞	1.05	2¼	0.82	0.74	3¼	4	⅝
1	4⅞	11/16	2⅛	1.32	2 7/16	1.05	0.96	3½	4	⅝
1¼	5¼	13/16	2½	1.66	2⅝	1.38	1.28	3⅞	4	⅝
1½	6⅛	⅞	2¾	1.90	2¾	1.61	1.50	4½	4	¾
2	6½	1	3 5/16	2.38	2⅞	2.07	1.94	5	8	⅝
2½	7½	1⅛	3 15/16	2.88	3⅛	2.47	2.32	5⅞	8	¾
3	8¼	1¼	4⅝	3.50	3¼	3.07	2.90	6⅝	8	¾
3½	9	1⅜	5¼	4.00	3⅜	3.55	3.36	7¼	8	⅞
4	10	1⅜	5¾	4.50	3½	4.03	3.83	7⅞	8	⅞
5	11	1½	7	5.56	4	5.05	4.81	9¼	8	⅞
6	12½	1⅝	8⅛	6.63	4 1/16	6.07	5.76	10⅝	12	⅞
8	15	1⅞	10¼	8.63	4⅝	7.98	7.63	13	12	1
10	17½	2⅛	12⅝	10.75	4⅞	10.02		15¼	16	1⅛
12	20½	2¼	14¾	12.75	5⅜			17¾	16	1¼
14 O.D.	23	2⅜	16¾	14.00	5⅞	To be specified by purchaser[7]	To be specified by purchaser[7]	20¼	20	1¼
16 O.D.	25½	2½	19	16.00	6			22½	20	1⅜
18 O.D.	28	2⅝	21	18.00	6½			24¾	24	1⅜
20 O.D.	30½	2¾	23⅛	20.00	6⅝			27	24	1½
24 O.D.	36	3	27⅝	24.00	6⅞			32	24	1¾

See Welding Bevel and Reducing Welding-neck Flanges, in the introductory notes, p. 634.

[1] The dimensions given and the marking for sizes ½ to 3½ in., inclusive, are identical with those of the 600 lb flanges.

[2] The raised face of ¼ in. is not included in thickness of flange, minimum, or in length through hub.

[3] The outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg.

[4] Dimensions *H* and *J* correspond to the outside and inside diameters of pipe as given in ASA B36.10, Schedules 40 and 80.

[5] See Tolerances for Welding-neck Flanges, in the introductory notes, p. 633.

[6] These diameters (0.62 to 10.02) are identical with the diameters of what was formerly designated as "standard-weight pipe" of the corresponding sizes.

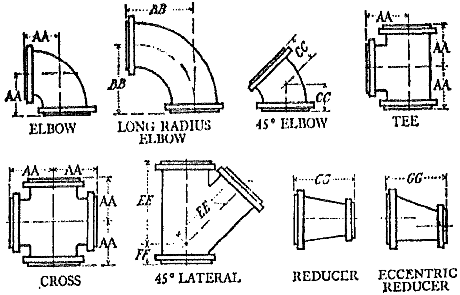
[7] These flanges are regularly bored to match inside diameter of Schedule 80 pipe but are bored to Schedule 40 pipe when so ordered.

[8] These diameters (0.55 to 7.63) are identical with the diameters of what was formerly designated as "extra-strong pipe" of the corresponding sizes.

TABLE CXVIII. DIMENSIONS OF STEEL FLANGED FITTINGS WITH PROJECTING FACES FOR PRIMARY-SERVICE PRESSURE RATING OF 300 PSI GAGE

(For complete pressure-temperature ratings see Tables CV to CVIII) (Table 22, ASA B16 e)

(All dimensions in inches)



Nominal pipe size	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of bolt circle	Number of bolts	Size of bolts	1/16-in. raised face					
								Center to contact surface of raised face elbow, tee, and cross[1,2,3] *AA*	Center to contact surface of raised face, long radius ell[1,2,3] *BB*	Center to contact surface of raised face, 45-deg. ell[1,2,3] *CC*	Long center to contact surface of raised face lateral[1,2,3] *EE*	Short center to contact surface of raised face lateral[1,2,3] *FF*	Contact surface to contact surface of raised face reducer[1,2,3] *GG*
1	1	1/4	4 7/8	11/16	3 1/2	4	5/8	4	5	2 1/4	6 1/2	2	4 1/2
1 1/4	1 1/4	1/4	5 1/4	3/4	3 7/8	4	5/8	4 1/4	5 1/2	2 1/2	7 1/4	2 1/4	4 1/2
1 1/2	1 1/2	1/4	6 1/8	13/16	4 1/2	4	3/4	4 1/2	6	2 3/4	8 1/2	2 1/2	4 1/2

TABLE CXVIII.—(*Concluded*)

Nominal pipe size	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of bolt circle	Number of bolts	Size of bolts	$\frac{1}{16}$-in. raised face: Center to contact surface of raised face elbow, tee, and cross[1,2,3] *AA*	Center to contact surface of raised face, long radius ell[1,2,3] *BB*	Center to contact surface of raised face, 45-deg. ell[1,2,3] *CC*	Long center to contact surface of raised face lateral[1,2,3] *EE*	Short center to contact surface of raised face lateral[1,2,3] *FF*	Contact surface to contact surface of raised face reducer[1,2,3] *GG*
2	2	1/4	6 1/2	7/8	5	8	5/8	5	6 1/2	3	9	2 1/2	5
2 1/2	2 1/2	1/4	7 1/2	1	5 7/8	8	3/4	5 1/2	7	3 1/2	10 1/2	2 1/2	5 1/2
3	3	9/32	8 1/4	1 1/8	6 5/8	8	3/4	6	7 3/4	3 1/2	11	3	6
3 1/2	3 1/2	9/32	9	1 3/16	7 1/4	8	3/4	6 1/2	8 1/2	4	12 1/2	3	6 1/2
4	4	5/16	10	1 1/4	7 7/8	8	3/4	7	9	4 1/2	13 1/2	3	7
5	5	3/8	11	1 3/8	9 1/4	8	3/4	8	10 1/4	5	15	3 1/2	8
6	6	3/8	12 1/2	1 7/16	10 5/8	12	3/4	8 1/2	11 1/2	5 1/2	17 1/2	4	9
8	8	7/16	15	1 5/8	13	12	7/8	10	14	6	20 1/2	5	11
10	10	1/2	17 1/2	1 7/8	15 1/4	16	1	11 1/2	16 1/2	7	24	5 1/2	12
12	12	9/16	20 1/2	2	17 3/4	16	1 1/8	13	19	8	27 1/2	6	14
14 O.D.	13 1/4	5/8	23	2 1/8	20 1/4	20	1 1/8	15	21 1/2	8 1/2	31	6 1/2	16
16 O.D.	15 1/4	11/16	25 1/2	2 1/4	22 1/2	20	1 1/4	16 1/2	24	9 1/2	34 1/2	7 1/2	18
18 O.D.	17	3/4	28	2 3/8	24 3/4	24	1 1/4	18	26 1/2	10	37 1/2	8	19
20 O.D.	19	13/16	30 1/2	2 1/2	27	24	1 1/4	19 1/2	29	10 1/2	40 1/2	8 1/2	20
24 O.D.	23	15/16	36	2 3/4	32	24	1 1/2	22 1/2	34	12	47 1/2	10	24

True Y (see Fig. 38). The dimensions of this fitting should be the same as those of the 90-deg elbow with a back outlet in which the dimension of the back-outlet contact face to the intersecting center lines is the same as dimension *FF* of the 45-deg lateral. The contact face to intersecting center lines of each Y outlet is the same as dimension *AA* of the elbow.

[1] A raised face of $\frac{1}{16}$ in. is provided on the flange of each opening of these fittings and is included in (1) thickness of flange, minimum; (2) center-to-contact surface, and (3) contact-surface-to-contact-surface dimensions.

[2] Where facings other than the $\frac{1}{16}$-in. raised face are used, the center-to-contact-surface dimensions shall remain unchanged, and new center-to-contact-surface dimensions shall be established to suit the facings used.

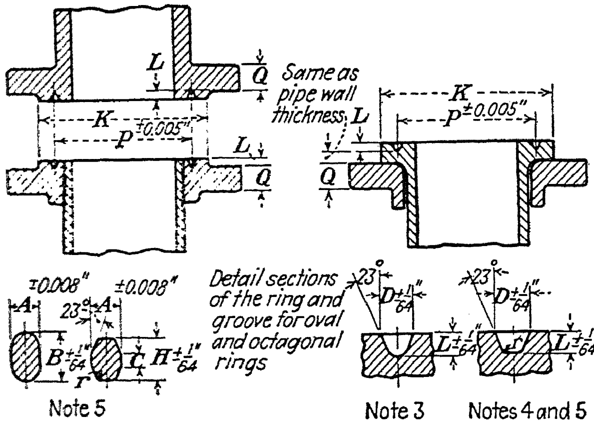
[3] Reducing fittings shall have the same center-to-contact-surface dimensions as those of straight-size fittings of the largest opening.

The center-to-end dimensions of ring-joint fittings are obtained by adding the depth of groove as given in Table CXIX to the center-to-contact-surface dimension.

TABLE CXIX.—FACING DIMENSIONS FOR 300-, 400-, AND 600-LB RING-JOINT FLANGES*

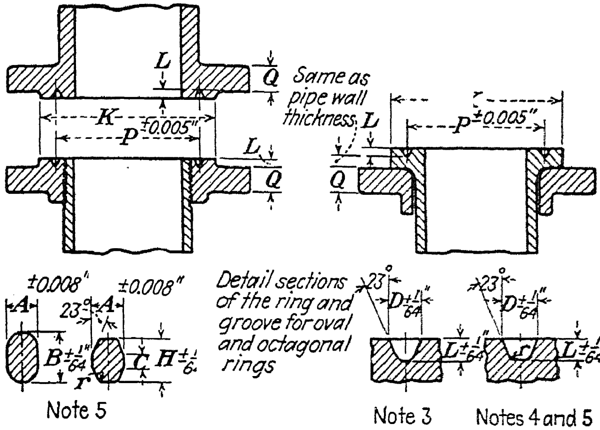
(Condensed from Tables 24, 30, and 36, ASA B16 e)

(All dimensions in inches)



Nominal pipe size	Pitch diameter of ring and groove	Width of ring	Height of ring		Height of flat on octagonal rings	Width of groove	Depth of groove[1]	Diameter of raised face for ring joint or lapped	Ring numbers*	Approximate distance between flanges of ring joints when ring is compressed[2]		
			Oval	Octagonal						300 lb	400 lb	600 lb
	P	A	B	H	C	D	L	K				
1/2	1 11/32	1/4	7/16	....	...	9/32	7/32	2	R 11	1/8	1/8	1/8
3/4	1 11/16	5/16	9/16	....	...	11/32	1/4	2 1/2	R 13	5/32	5/32	5/32
1	2	5/16	9/16	....	...	11/32	1/4	2 3/4	R 16	5/32	5/32	5/32
1 1/4	2 3/8	5/16	9/16	....	...	11/32	1/4	3 1/8	R 18	5/32	5/32	5/32
1 1/2	2 11/16	5/16	9/16	....	...	11/32	1/4	3 9/16	R 20	5/32	5/32	5/32
2	3 1/4	7/16	11/16	....	...	15/32	5/16	4 1/4	R 23	7/32	3/16	3/16
2 1/2	4	7/16	11/16	....	...	15/32	5/16	5	R 26	7/32	3/16	3/16
3	4 7/8†	7/16	11/16	....	...	15/32	5/16	5 3/4	R 31†	7/32	3/16	3/16
3 1/2	5 3/16	7/16	11/16	....	...	15/32	5/16	6 1/4	R 34	7/32	3/16	3/16
4	5 7/8	7/16	11/16	....	...	15/32	5/16	6 7/8	R 37	7/32	7/32	3/16
5	7 1/8	7/16	11/16	....	...	15/32	5/16	8 1/4	R 41	7/32	7/32	3/16
6	8 5/16	7/16	11/16	....	...	15/32	5/16	9 1/2	R 45	7/32	7/32	3/16
8	10 5/8	7/16	11/16	....	...	15/32	5/16	11 7/8	R 49	7/32	7/32	3/16
10	12 3/4	7/16	11/16	....	...	15/32	5/16	14	R 53	7/32	7/32	3/16
12	15	7/16	11/16	....	...	15/32	5/16	16 1/4	R 57	7/32	7/32	3/16

TABLE CXIX.—(*Concluded*)



Nominal pipe size	Pitch diameter of ring and groove *P*	Width of ring *A*	Height of ring: Oval *B*	Height of ring: Octagonal *H*	Height of flat on octagonal rings *C*	Width of groove *D*	Depth of groove[1] *L*	Diameter of raised face for ring joint or lapped *K*	Ring numbers*	Approximate distance between flanges of ring joints when ring is compressed[2]: 300 lb	400 lb	600 lb
14 O.D.	16½	7/16	11/16	....	...	15/32	5/16	18	R 61	7/32	7/32	3/16
16 O.D.	18½	7/16	11/16	....	...	15/32	5/16	20	R 65	7/32	7/32	3/16
18 O.D.	21	7/16	11/16	....	...	15/32	5/16	22⅝	R 69	7/32	7/32	3/16
24 O.D.	23	½	¾	11/16	5/16	17/32	⅜	25	R 73	7/32	7/32	3/16
24 O.D.	27¼	⅝	⅞	13/16	5/16	21/32	7/16	29½	R 77	¼	¼	7/32

Dimension Q is minimum flange thickness.

* The dimensions of the rings and grooves of ring joint flanges rated at 300, 400, and 600 lb are identical. For dimensions of rings, see Table CXXXVII. When ordering rings for nominal pipe sizes which may have either oval- or octagonal-shaped rings, purchasers must specify oval- or octagonal-shaped rings as desired.

† For ring joints with lapped flanges, the pitch diameter of the ring should be 4⅝ in. instead of 4[illegible] in. and the ring number should be R50 instead of R31.

[1] The depth of groove is added to the minimum flange thickness, except in sizes ½ to 1½ in. inclusive where the groove cuts 1/32 in. into the minimum flange thickness.

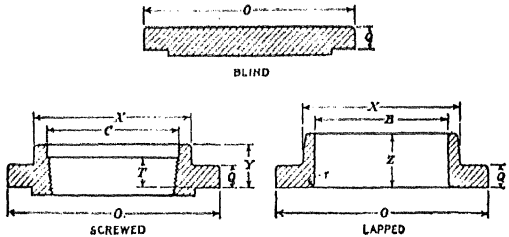
[2] For calculating the "laying length" of fittings with ring joints the space dimensions given in the table must be added.

[3] This type of groove which should be used for oval rings having widths of ¼, 5/16, and 7/16 in., may be made with flat bottom at the option of the manufacturer.

[4] This type of groove should be used for rings having widths of ½ in. and greater.

[5] The corner radius for octagonal rings and flat bottom grooves r is [illegible] in. for rings having widths of ¼, 5/16, and 7/16 in., and [illegible] in. for rings having widths [illegible] in. The tolerance on this dimension is ± 1/64 in. for the ring. This r is maximum for the groove.

TABLE CXX. DIMENSIONS OF STEEL FLANGES FOR PRIMARY-SERVICE PRESSURE RATING OF 400 PSI GAGE
(For complete pressure-temperature ratings see Tables CV to CVIII)
(Table 26, ASA B16 e)
(All dimensions in inches)



Nominal pipe size[1]	Outside diameter of flange	Thickness of flange,[2] minimum	Diameter of hub[3]	Diameter of bolt circle	Number of bolts	Size of bolts	Counter-bore screwed flange,[4] minimum	Bore of lapped flange,[4] minimum	Corner radius of bore of lapped flange and pipe	Length through hub		Thread length[1,5]
										Screwed[1]	Lapped	
	O	Q	X				C	B	r	Y	Z	T
½	3¾	9/16	1½	2⅝	4	½	0.93	0.93	⅛	⅞	⅞	⅝
¾	4⅝	⅝	1⅞	3¼	4	⅝	1.14	1.14	⅛	1	1	⅝
1	4⅞	11/16	2⅛	3½	4	⅝	1.41	1.41	⅛	1 1/16	1 1/16	11/16
1¼	5¼	13/16	2½	3⅞	4	⅝	1.75	1.75	3/16	1⅛	1⅛	13/16
1½	6⅛	⅞	2¾	4½	4	¾	1.99	1.99	¼	1¼	1¼	⅞

TABLE CXX.—(*Concluded*)

Nominal pipe size[1]	Outside diameter of flange	Thickness of flange,[2] minimum	Diameter of hub[3]	Diameter of bolt circle	Number of bolts	Size of bolts	Counter-bore screwed flange,[4] minimum	Bore of lapped flange,[4] minimum	Corner radius of bore of lapped flange and pipe	Length through hub		Thread length[1,5]
										Screwed[1]	Lapped	
	O	Q	X				C	B	r	Y	Z	T
2	6½	1	3⁵⁄₁₆	5	8	⅝	2.50	2.50	⁵⁄₁₆	1⁷⁄₁₆	1⁷⁄₁₆	1⅛
2½	7½	1⅛	3¹⁵⁄₁₆	5⅞	8	¾	3.00	3.00	⁵⁄₁₆	1⅝	1⅝	1¼
3	8¼	1¼	4⅝	6⅝	8	¾	3.63	3.63	⅜	1¹³⁄₁₆	1¹³⁄₁₆	1⅜
3½	9	1⅜	5¼	7¼	8	⅞	4.13	4.13	⅜	1¹⁵⁄₁₆	1¹⁵⁄₁₆	1⁹⁄₁₆
4	10	1⅜	5¾	7⅞	8	⅞	4.63	4.63	⁷⁄₁₆	2	2	1⁷⁄₁₆
5	11	1½	7	9¼	8	⅞	5.69	5.69	⁷⁄₁₆	2⅛	2⅛	1¹¹⁄₁₆
6	12½	1⅝	8⅛	10⅝	12	⅞	6.75	6.75	½	2¼	2¼	1¹³⁄₁₆
8	15	1⅞	10¼	13	12	1	8.75	8.81	½	2¹¹⁄₁₆	2¹¹⁄₁₆	2
10	17½	2⅛	12⅝	15¼	16	1⅛	10.88	10.94	½	2⅞	4	2³⁄₁₆
12	20½	2¼	14¾	17¾	16	1¼	12.94	12.94	½	3⅛	4¼	2⅜
14 O.D.	23	2⅜	16¾	20¼	20	1¼	14.19	14.25	½	3⁵⁄₁₆	4⅝	2½
16 O.D.	25½	2½	19	22½	20	1⅜	16.19	16.25	½	3¹¹⁄₁₆	5	2¹¹⁄₁₆
18 O.D.	28	2⅝	21	24¾	24	1⅜	18.19	18.25	½	3⅞	5⅜	2¾
20 O.D.	30½	2¾	23⅛	27	24	1½	20.19	20.25	½	4	5¾	2⅞
24 O.D.	36	3	27⅝	32	24	1¾	24.19	24.25	½	4½	6¼	3¼

[1] The dimensions given and the marking for sizes ½ to 3½ in., inclusive, are identical with those of the 600-lb fittings.

For the dimensions of reducing screwed flanges, see Table CIX.

Blind flanges should be faced as follows: (1) When the center portion of the inside face is raised, it need not be faced, but its diameter shall be at least 1 in. smaller than the inside diameter of the fitting given in Table CXXI, (2) when the center portion of the inside face is depressed, it need not be faced, and its diameter shall not be greater than the inside diameter of the fitting given in Table CXXI.

[2] The raised face of ¼ in. is not included in thickness of flange, minimum, length through hub, or thread length.

[3] This dimention is for large end of hub, which may be tapered 5 deg for draft.

[4] The dimensions given for the counterbore of the screwed flange and the bore of the lapped flange are based on the variation of the outside diameter (oversize) of steel pipe given in ASTM Specifications A-53, A-120, and API Specification 5-L. The tolerances on the bore of these flanges are +¹⁄₃₂ in. for sizes 10 in. and smaller, and +¹⁄₁₆ in. for sizes 12 in. and larger.

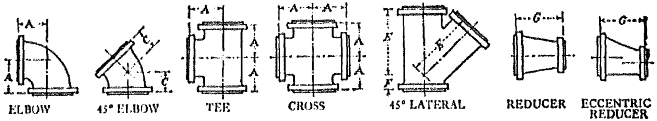
[5] Thread length given takes API line pipe thread. This standard thread is based on the American Standard for Pipe Threads (ASA B2.1), but some sizes are longer, averaging about two threads, with the exception of the 2-in. size which is four threads.

TABLE CXXI.—CENTER-TO-FLANGE-EDGE DIMENSIONS OF STEEL-FLANGED FITTINGS FOR PRIMARY-SERVICE PRESSURE RATING OF 400 PSI GAGE

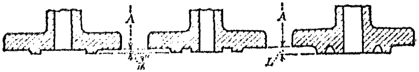
(For complete pressure-temperature ratings see Tables CV to CVIII)

(Table 28, ASA B16 e)

(All dimensions in inches)



Female, Groove, and Ring-Joint Facings for Fittings



Nominal pipe size[1]	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange,[2] minimum	Diameter of bolt circle	Number of bolts	Size of bolts	Flange edge: Center to flange edge elbow, tee, and cross[2,3,4] A	Flange edge: Center to flange edge 45-deg ell[2,3,4] C	Flange edge: Long center to flange edge lateral[2,3,4] E	Flange edge: Short center to flange edge lateral[2,3,4] F	Flange edge: Flange edge to flange edge reducer[2,3] G
1/2	1/2	1/4	3 3/4	9/16	2 5/8	4	1/2	3	1 3/4	5 1/2	1 1/2	4 1/2
3/4	3/4	1/4	4 5/8	5/8	3 1/4	4	5/8	3 1/2	2 1/4	6 1/2	1 3/4	4 1/2
1	1	1/4	4 7/8	11/16	3 1/2	4	5/8	4	2 1/4	7	2	4 1/2
1 1/4	1 1/4	1/4	5 1/4	13/16	3 7/8	4	5/8	4 1/4	2 1/2	7 3/4	2 1/4	4 1/2
1 1/2	1 1/2	1/4	6 1/8	7/8	4 1/2	4	3/4	4 1/2	2 3/4	8 3/4	2 1/2	4 1/2

TABLE CXXI.—(*Concluded*)

Nominal pipe size[1]	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange,[2] minimum	Diameter of bolt circle	Number of bolts	Size of bolts	Flange edge: Center to flange edge elbow, tee, and cross[2,3,4] *A*	Flange edge: Center to flange edge 45-deg ell[2,3,4] *C*	Flange edge: Long center to flange edge lateral[2,3,4] *E*	Flange edge: Short center to flange edge lateral[2,3,4] *F*	Flange edge: Flange edge to flange edge reducer[2,3] *G*
2	2	5/16	6½	1	5	8	5/8	5½	4	10	3¼	5½
2½	2½	3/8	7½	1⅛	5⅞	8	¾	6¼	4¼	11¼	3¼	6¼
3	3	3/8	8¼	1¼	6⅝	8	¾	6¾	4¾	12½	3¾	6¾
3½	3½	7/16	9	1⅜	7¼	8	⅞	7¼	5¼	13¾	4¼	7¼
4	4	3/8	10	1⅜	7⅞	8	⅞	7¾	5¼	15¾	4¼	7¾
5	5	7/16	11	1½	9¼	8	⅞	8¾	5¾	16½	4¾	8¾
6	6	7/16	12½	1⅝	10⅝	12	⅞	9½	6	18½	5	9½
8	8	9/16	15	1⅞	13	12	1	11½	6½	22	5½	11½
10	10	11/16	17½	2⅛	15¼	16	1⅛	13	7½	25½	6	13
12	12	¾	20½	2¼	17¾	16	1¼	14¾	8½	29½	6¼	14¾
14 O.D.	13⅛	13/16	23	2⅜	20¼	20	1¼	16	9	32½	6¾	16
16 O.D.	15	⅞	25½	2½	22½	20	1⅜	17½	10	36	7¾	18
18 O.D.	17	15/16	28	2⅝	24¾	24	1⅜	19	10½	39	8¼	19
20 O.D.	18⅞	1 1/16	30½	2¾	27	24	1½	20½	11	42½	8¾	20½
24 O.D.	22⅝	1 3/16	36	3	32	24	1¾	24	12½	50	10¼	24

True *Y* (see Fig. 38). The dimensions of this fitting should be the same as those of the 90-deg elbow with a back outlet in which the dimension of the back-outlet contact face to the intersecting center lines is the same as dimension *FF* of the 45-deg lateral. The contact face to intersecting center lines of each Y outlet is the same as dimension *AA* of the elbow.

[1] The dimensions given and the marking for sizes ½ to 3½ in., inclusive, are identical with those of the 600-lb fittings.

[2] The raised face of ¼ in. is not included in (1) thickness of flange, minimum, (2) center to flange edge or (3) flange-edge-to-flange-edge dimensions.

[3] Where facings other than the ¼-in. raised face are used the center-to-flange-edge dimensions shall remain unchanged, and the new center-to-contact-surface or contact-surface-to-contact-surface dimensions shall be established to suit the facing used.

[4] Reducing fittings shall have the same center-to-flange-edge dimensions as those of straight-size fittings of the largest opening.

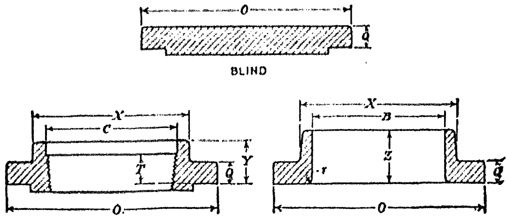
The center-to-end dimensions of ring-joint fittings are obtained by adding the depth of groove as given in Table CXIX to the center-to-flange-edge dimensions.

TABLE CXXII. DIMENSIONS OF STEEL FLANGES FOR PRIMARY-SERVICE PRESSURE RATING OF 600 PSI GAGE

(For complete pressure-temperature ratings see Tables CV to CVIII)

(Table 32, ASA B16 e)

(All dimensions in inches)



BLIND

SCREWED

LAPPED

Nominal pipe size	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of hub[2]	Diameter of bolt circle	Number of bolts	Size of bolts	Counter-bore screwed flange,[3] minimum	Bore of lapped flange,[3] minimum	Corner radius of bore of lapped flange and pipe	Length through hub		Thread length[1,4]
										Screwed[1]	Lapped	
	O	Q	X				C	B	r	Y	Z	T
½	3¾	9/16	1½	2⅝	4	½	0.93	0.93	⅛	⅞	⅞	⅝
¾	4⅝	⅝	1⅞	3¼	4	⅝	1.14	1.14	⅛	1	1	⅝
1	4⅞	11/16	2⅛	3½	4	⅝	1.41	1.41	⅛	1 1/16	1 1/16	11/16
1¼	5¼	13/16	2½	3⅞	4	⅝	1.75	1.75	3/16	1⅛	1⅛	13/16
1½	6⅛	⅞	2¾	4½	4	¾	1.99	1.99	¼	1¼	1¼	⅞

TABLE CXXII.—(*Concluded*)

Nominal pipe size	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of hub[2]	Diameter of bolt circle	Number of bolts	Size of bolts	Counter-bore screwed flange,[3] minimum	Bore of lapped flange,[3] minimum	Corner radius of bore of lapped flange and pipe	Length through hub		Thread length,[1,4]
										Screwed[1]	Lapped	
	O	Q	X				C	B	r	Y	Z	T
2	6½	1	3 5/16	5	8	⅝	2.50	2.50	5/16	1 7/16	1 7/16	1⅛
2½	7½	1⅛	3 15/16	5⅞	8	¾	3.00	3.00	5/16	1⅝	1⅝	1¼
3	8¼	1¼	4⅝	6⅝	8	¾	3.63	3.63	⅜	1 13/16	1 13/16	1⅜
3½	9	1⅜	5¼	7¼	8	⅞	4.13	4.13	⅜	1 15/16	1 15/16	1 9/16
4	10¾	1½	6	8½	8	⅞	4.63	4.63	7/16	2⅛	2⅛	1⅝
5	13	1¾	7 7/16	10½	8	1	5.69	5.69	7/16	2⅜	2⅜	1⅞
6	14	1⅞	8¾	11½	12	1	6.75	6.75	½	2⅝	2⅝	2
8	16½	2 3/16	10¾	13¾	12	1⅛	8.75	8.81	½	3	3	2¼
10	20	2½	13½	17	16	1¼	10.88	10.94	½	3⅜	4⅜	2 9/16
12	22	2⅝	15¾	19¼	20	1¼	12.94	12.94	½	3⅝	4⅝	2¾
14 O.D.	23¾	2¾	17	20¾	20	1⅜	14.19	14.25	½	3 11/16	5	2⅞
16 O.D.	27	3	19½	23¾	20	1½	16.19	16.25	½	4 3/16	5½	3 1/16
18 O.D.	29¼	3¼	21½	25¾	20	1⅝	18.19	18.25	½	4⅝	6	3⅛
20 O.D.	32	3½	24	28½	24	1⅝	20.19	20.25	½	5	6½	3¼
24 O.D.	37	4	28¼	33	24	1⅞	24.19	24.25	½	5½	7¼	3⅝

For the dimensions of reducing screwed flanges see Table CIX.

Blind flanges should be faced as follows: (1) When the center portion of the inside face is raised, it need not be faced, but its diameter shall be at least 1 in. smaller than the inside diameter of the fitting given in Table CXXV, (2) when the center portion of the inside face is depressed, it need not be faced, and its diameter shall not be greater than the inside diameter of the fitting given in Table CXXV.

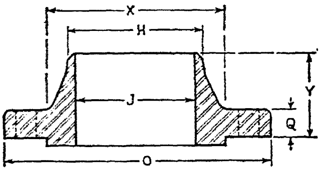
[1] The raised face of ¼ in. is not included in thickness of flange, minimum, length through hub, or thread length.

[2] This dimension is for large end of hub, which may be tapered 5 deg for draft.

[3] The dimensions given for the counterbore of the screwed flange and the bore of the lapped flange are based on the variation of the outside diameter (oversize) of steel pipe given in ASTM Specifications A-53, A-120, and API Specification 5-L. The tolerances on the bore of these flanges are +1/32 in. for sizes 10 in. and smaller, and +1/16 in. for sizes 12 in. and larger.

[4] Thread length given is proportionately longer than required for API or American Standard for Pipe Threads (ASA B2.1) and satisfies the general requirement for longer threads for the heavier pressures.

TABLE CXXIII.—DIMENSIONS OF STEEL WELDING-NECK FLANGES FOR PRIMARY-SERVICE PRESSURE RATING OF 600 PSI GAGE
(For complete pressure-temperature ratings see Tables CV to CVIII)
(Table 33, ASA B16 e)
(All dimensions in inches)



Nominal pipe size	Diameter of flange O	Thickness of flange,[1] minimum Q	Diameter of hub X	Hub diameter beginning of chamfer[2,3,4] H	Length through hub[1] Y	Inside diameter of pipe[4] J	Diameter of bolt circle	Number of bolts	Size of bolts
½	3¾	9/16	1½	0.84	2 1/16		2⅝	4	½
¾	4⅝	⅝	1⅞	1.05	2¼		3¼	4	⅝
1	4⅞	11/16	2⅛	1.32	2 7/16		3½	4	⅝
1¼	5¼	13/16	2½	1.66	2⅝		3⅞	4	⅝
1½	6⅛	⅞	2¾	1.90	2¾		4½	4	¾
2	6½	1	3 5/16	2.38	2⅞	To be specified by purchaser	5	8	⅝
2½	7½	1⅛	3 15/16	2.88	3⅛		5⅞	8	¾
3	8¼	1¼	4⅝	3.50	3¼		6⅝	8	¾
3½	9	1⅜	5¼	4.00	3⅜		7¼	8	⅞
4	10¾	1½	6	4.50	4		8½	8	⅞
5	13	1¾	7 7/16	5.56	4½		10½	8	1
6	14	1⅞	8¾	6.63	4⅝		11½	12	1
8	16½	2 3/16	10¾	8.63	5¼		13¾	12	1⅛
10	20	2½	13½	10.75	6		17	16	1¼
12	22	2⅝	15¾	12.75	6⅛		19¼	20	1¼
14 O.D.	23¾	2¾	17	14.00	6½		20¾	20	1⅜
16 O.D.	27	3	19½	16.00	7		23¾	20	1½
18 O.D.	29¼	3¼	21½	18.00	7¼		25¾	20	1⅝
20 O.D.	32	3½	24	20.00	7½		28½	24	1⅝
24 O.D.	37	4	28¼	24.00	8		33	24	1⅞

See Welding Bevel and Reducing Welding-neck Flanges in the introductory notes, p. 634.

[1] The raised face of ¼ in. is not included in thickness of flange, minimum, or length through hub.

[2] The outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg.

[3] Dimension H corresponds to the outside diameters of pipe as given in ASA B36.10.

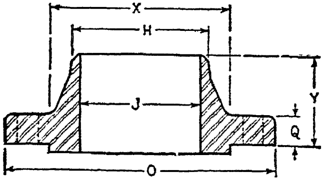
[4] See Tolerance for Welding-neck Flanges in the introductory notes p. 633.

TABLE CXXIV.—DIMENSIONS OF STEEL WELDING-NECK FLANGES FOR PRIMARY-SERVICE PRESSURE RATING OF 900 PSI GAGE

(For complete pressure-temperature ratings see Tables CV to CVIII)

(Table 39, ASA B16 e)

(All dimensions in inches)



Nominal pipe size[1]	Diameter of flange *O*	Thickness of flange,[2] minimum *Q*	Diameter of hub *X*	Hub diameter of beginning of chamfer[3,4,5] *H*	Length through hub[2] *Y*	Inside diameter of pipe[6] *J*	Diameter of bolt circle	Number of bolts	Size of bolts
½	4¾	⅞	1½	0.84	2⅜		3¼	4	¾
¾	5⅛	1	1¾	1.05	2¾		3½	4	¾
1	5⅞	1⅛	2 1/16	1.32	2⅞		4	4	⅞
1¼	6¼	1⅛	2½	1.66	2⅞		4⅜	4	⅞
1½	7	1¼	2¾	1.90	3¼		4⅞	4	1
2	8½	1½	4⅛	2.38	4	To be specified by purchaser	6½	8	⅞
2½	9⅝	1⅝	4⅞	2.88	4⅛		7½	8	1
3	9½	1½	5	3.50	4		7½	8	⅞
4	11½	1¾	6¼	4.50	4½		9¼	8	1⅛
5	13¾	2	7½	5.56	5		11	8	1¼
6	15	2 3/16	9¼	6.63	5½		12½	12	1⅛
8	18½	2½	11¾	8.63	6⅜		15½	12	1⅜
10	21½	2¾	14½	10.75	7¼		18½	16	1⅜
12	24	3⅛	16½	12.75	7⅞		21	20	1⅜
14 O.D.	25¼	3⅜	17¾	14.00	8⅜		22	20	1½
16 O.D.	27¾	3½	20	16.00	8½		24¼	20	1⅝
18 O.D.	31	4	22¼	18.00	9		27	20	1⅞
20 O.D.	33¾	4¼	24½	20.00	9¾		29½	20	2
24 O.D.	41	5½	29½	24.00	11½		35½	20	2½

See Welding Bevel and Reducing Welding-neck Flanges in the introductory notes, p. 634.

[1] The dimensions given and the marking for sizes ½ to 2½ in., inclusive, are identical with those of the 1,500-lb flanges.

[2] The raised face of ¼ in. is not included in thickness of flange, minimum, or length through hub.

[3] The outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg.

[4] Dimension *H* corresponds to the outside diameters of pipe as given in ASA B36.10.

[5] See Tolerance for Welding-neck Flanges in the introductory notes, p. 633.

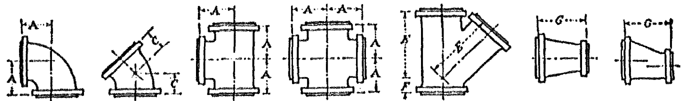
[6] If specified by the purchaser, flanges may be furnished and used with inside diameters corresponding to those of the pipe to which they are to be welded.

TABLE CXXV. CENTER-TO-FLANGE-EDGE DIMENSIONS OF STEEL-FLANGED FITTINGS FOR PRIMARY-SERVICE PRESSURE RATING OF 600 PSI GAGE

(For complete pressure-temperature ratings see Tables CV to CVIII)

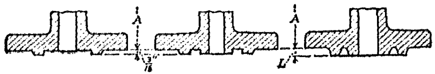
(Table 34, ASA B16 e)

(All dimensions in inches)



ELBOW 45° ELBOW TEE CROSS 45° LATERAL REDUCER ECCENTRIC REDUCER

Female, Groove, and Ring-Joint Facings for Fittings



Nominal pipe size	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange,[1] minimum[1]	Diameter of bolt circle	Number of bolts	Size of bolts	Flange edge: Center to flange edge elbow, tee, and cross[1,2,3] A	Center to flange edge 45-deg ell[1,2,3] C	Long center to flange edge lateral[1,2,3] E	Short center to flange edge lateral[1,2,3] F	Flange edge to flange edge reducer[1,2] G
1/2	1/2	1/4	3 3/4	9/16	2 5/8	4	1/2	3	1 3/4	5 1/2	1 1/2	4 1/2
3/4	3/4	1/4	4 5/8	5/8	3 1/4	4	5/8	3 1/2	2 1/4	6 1/2	1 3/4	4 1/2
1	1	1/4	4 7/8	11/16	3 1/2	4	5/8	4	2 1/4	7	2	4 1/2
1 1/4	1 1/4	1/4	5 1/4	13/16	3 7/8	4	5/8	4 1/4	2 1/2	7 3/4	2 1/4	4 1/2
1 1/2	1 1/2	1/4	6 1/8	7/8	4 1/2	4	3/4	4 1/2	2 3/4	8 3/4	2 1/2	4 1/2

TABLE CXXV.—(*Concluded*)

Nominal pipe size	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange,[1] minimum[1]	Diameter of bolt circle	Number of bolts	Size of bolts	Flange edge: Center to flange edge elbow, tee, and cross[1,2,3] *A*	Flange edge: Center to flange edge 45-deg ell[1,2,3] *C*	Flange edge: Long center to flange edge lateral[1,2,3] *E*	Flange edge: Short center to flange edge lateral[1,2,3] *F*	Flange edge: Flange edge to flange edge reducer[1,2] *G*
2	2	5/16	6½	1	5	8	⅝	5½	4	10	3¼	5½
2½	2½	⅜	7½	1⅛	5⅞	8	¾	6¼	4¼	11¼	3¼	6¼
3	3	⅜	8¼	1¼	6⅝	8	¾	6¾	4¾	12½	3¾	6¾
3½	3½	7/16	9	1⅜	7¼	8	⅞	7¼	5¼	13¾	4¼	7¼
4	4	½	10¾	1½	8½	8	⅞	8¼	5¾	16¼	4¼	8¼
5	5	9/16	13	1¾	10½	8	1	9¾	6¾	19¼	5¾	9¾
6	6	⅝	14	1⅞	11½	12	1	10¾	7¼	20¾	6¼	10¾
8	7⅞	¾	16½	2 3/16	13¾	12	1⅛	12¾	8¼	24¼	6¾	12¾
10	9¾	⅞	20	2½	17	16	1¼	15¼	9¼	29¼	7¾	15¼
12	11¾	1	22	2⅝	19¼	20	1¼	16¼	9¾	31¼	8¼	16¼
14 O.D.	12⅞	1⅛	23¾	2¾	20¾	20	1⅜	17¼	10½	34	8¾	17¼
16 O.D.	14¾	1¼	27	3	23¾	20	1½	19¼	11½	38¼	9¾	19¼
18 O.D.	16½	1⅜	29¼	3¼	25¾	20	1⅝	21¼	12	41¾	10¼	21¼
20 O.D.	18¼	1½	32	3½	28½	24	1⅝	23¼	12¾	45¼	10¾	23¼
24 O.D.	22	1¾	37	4	33	24	1⅞	27¼	14½	52¾	12¾	27¼

True *Y* (see Fig. 38). The dimensions of this fitting should be the same as those of the 90-deg elbow with a back outlet in which the dimension of the back-outlet contact face to the intersecting center lines is the same as dimension *FF* of the 45-deg lateral. The contact face to intersecting center lines of each Y outlet is the same as dimension *AA* of the elbow.

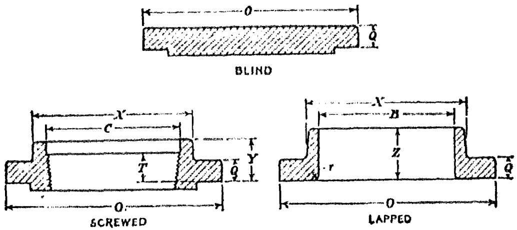
[1] The raised face of ¼ in. is not included in (1) thickness of flange, minimum, (2) center to flange edge, or (3) flange-edge-to-flange-edge dimensions.

[2] Where facings other than the ¼-in. raised face are used, the center-to-flange-edge dimensions shall remain unchanged, and new center-to-contact-surface or contact-surface-to-contact-surface dimensions shall be established to suit the facing used.

[3] Reducing fittings shall have the same center-to-flange-edge dimensions as those of straight-size fittings of the large opening.

The center-to-end dimensions of ring-joint fittings are obtained by adding the depth of groove as given in Table CXIX to the center-to-flange-edge dimensions.

TABLE CXXVI. DIMENSIONS OF STEEL FLANGES FOR PRIMARY-SERVICE PRESSURE OF 900 PSI GAGE
(For complete pressure-temperature ratings see Tables CV to CVIII)
(Table 38, ASA B16 e)
(All dimensions in inches)



Nominal pipe size[1]	Outside diameter of flange	Thickness of flange,[2] minimum	Diameter of hub[3]	Diameter of bolt circle	Number of bolts	Size of bolts	Counter-bore screwed flange,[4] minimum	Bore of lapped flange,[4] minimum	Corner radius of bore of lapped flange and pipe	Length through hub		Thread length[2,5]
										Screwed[2]	Lapped[2]	
	O	Q	X				C	B	r	Y	Z	T
½	4¾	⅞	1½	3¼	4	¾	0.93	0.93	⅛	1¼	1¼	⅞
¾	5⅛	1	1¾	3½	4	¾	1.14	1.14	⅛	1⅜	1⅜	1
1	5⅞	1⅛	2$\frac{1}{16}$	4	4	⅞	1.41	1.41	⅛	1⅝	1⅝	1⅛
1¼	6¼	1⅛	2½	4⅜	4	⅞	1.75	1.75	$\frac{3}{16}$	1⅝	1⅝	1$\frac{3}{16}$

TABLE CXXVI.—(*Concluded*)

Nominal pipe size[1]	Outside diameter of flange *O*	Thickness of flange,[2] minimum *Q*	Diameter of hub[3] *X*	Diameter of bolt circle	Number of bolts	Size of bolts	Counter-bore screwed flange,[4] minimum *C*	Bore of lapped flange,[4] minimum *B*	Corner radius of bore of lapped flange and pipe *r*	Length through hub: Screwed[2] *Y*	Length through hub: Lapped[2] *Z*	Thread length[2,5] *T*
1½	7	1¼	2¾	4⅞	4	1	1.99	1.99	¼	1¾	1¾	1¼
2	8½	1½	4⅛	6½	8	⅞	2.50	2.50	5/16	2¼	2¼	1½
2½	9⅝	1⅝	4⅞	7½	8	1	3.00	3.00	5/16	2½	2½	1⅞
3	9½	1½	5	7½	8	⅞	3.63	3.63	⅜	2⅛	2⅛	1⅝
4	11½	1¾	6¼	9¼	8	1⅛	4.63	4.63	7/16	2¾	2¾	1⅞
5	13¾	2	7½	11	8	1¼	5.69	5.69	7/16	3⅛	3⅛	2⅛
6	15	2 3/16	9¼	12½	12	1⅛	6.75	6.75	½	3⅜	3⅜	2¼
8	18½	2½	11¾	15½	12	1⅜	8.75	8.81	½	4	4½	2½
10	21½	2¾	14½	18½	16	1⅜	10.88	10.94	½	4¼	5	2 13/16
12	24	3⅛	16½	21	20	1⅜	12.94	12.94	½	4⅝	5⅝	3
14 O.D.	25¼	3⅜	17¾	22	20	1½	14.19	14.25	½	5⅛	6⅛	3¼
16 O.D.	27¾	3½	20	24¼	20	1⅝	16.19	16.25	½	5¼	6½	3⅜
18 O.D.	31	4	22¼	27	20	1⅞	18.19	18.25	½	6	7½	3½
20 O.D.	33¾	4¼	24½	29½	20	2	20.19	20.25	½	6¼	8¼	3⅝
24 O.D.	41	5½	29½	35½	20	2½	24.19	24.25	½	8	10½	4

[1] The dimensions given and the marking for sizes ½ to 2½ in., inclusive, are identical with those of the 1,500-lb flanges.

For the dimensions of reducing screwed flanges see Table CIX.

Blind flanges should be faced as follows: (1) When the center portion of the inside face is raised it need not be faced, but its diameter shall be at least 1 in. smaller than the inside diameter of the fitting given in Table CXXVII, (2) when the center portion of the inside face is depressed, it need not be faced, and its diameter shall not be greater than the inside diameter of the fitting given in Table CXXVII.

[2] The raised face of ¼ in. is not included in thickness of flange, minimum, length through hub, or thread length.

[3] This dimension is for large end of hub, which may be tapered 5 deg for draft.

[4] The dimensions given for the counterbore of the screwed flange and the bore of the lapped flange are based on the variation of the outside diameter (oversize) of steel pipe given in ASTM Specifications A-53, A-120, and API Specification 5-L. The tolerances on the bore of these flanges are +1/32 in. for sizes 10 in. and smaller, and +1/16 in. for sizes 12 in. and larger.

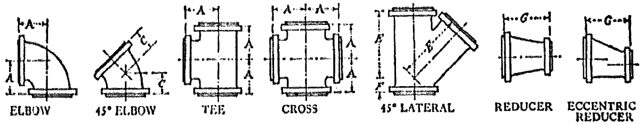
[5] Thread length given is proportionately longer than required for API or American Standard for Pipe Threads and satisfies the general requirement for longer threads for the heavier pressures.

TABLE CXXVII. CENTER-TO-FLANGE-EDGE DIMENSIONS OF STEEL-FLANGED FITTINGS FOR PRIMARY-SERVICE PRESSURE RATING OF 900 PSI GAGE

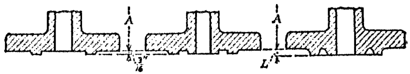
(For complete pressure-temperature ratings see Tables CV to CVIII)

(Table 40, ASA B16 e)

(All dimensions in inches)



Female, Groove, and Ring-Joint Facings for Fittings



Nominal pipe size[1]	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange,[2] minimum	Diameter of bolt circle	Number of bolts	Size of bolts	Flange edge: Center to flange edge elbow, tee, and cross[1,2,3] A	Center to flange edge 45-deg ell[1,2,3] C	Long center to flange edge lateral[1,2,3] E	Short center to flange edge lateral[1,2,3] F	Flange edge to flange edge reducer[1,2] G
½	½	5/16	4¾	⅞	3¼	4	¾	4¼	3			
¾	11/16	5/16	5⅛	1	3½	4	¾	4½	3¼			
1	⅞	⅜	5⅞	1⅛	4	4	⅞	4¾	3¼	8¾	2¼	4½
1¼	1⅛	⅜	6¼	1⅛	4⅜	4	⅞	5¼	3¾	9¾	2¾	5¼

TABLE CXXVII.—(*Concluded*)

Nominal pipe size[1]	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange,[2] minimum	Diameter of bolt circle	Number of bolts	Size of bolts	Flange edge: Center to flange edge elbow, tee, and cross[2,3,4] *A*	Flange edge: Center to flange edge 45-deg ell[2,3,4] *C*	Flange edge: Long center to flange edge lateral[2,3,4] *E*	Flange edge: Short center to flange edge lateral[2,3,4] *F*	Flange edge: Flange edge to flange edge reducer[2,3] *G*
1½	1⅜	7/16	7	1¼	4⅞	4	1	5¾	4	10¾	3¼	5¾
2	1⅞	9/16	8½	1½	6½	8	⅞	7	4½	13	3¾	6¾
2½	2¼	11/16	9⅝	1⅝	7½	8	1	8	5	15	4¼	7¾
3	2⅞	½	9½	1½	7½	8	⅞	7¼	5¼	14¼	4¼	7¼
4	3⅞	⅝	11½	1¾	9¼	8	1⅛	8¾	6¼	17¼	5¼	8¾
5	4¾	¾	13¾	2	11	8	1¼	10¾	7¼	20¾	6¼	10¾
6	5¾	13/16	15	2 3/16	12½	12	1⅛	11¾	7¾	22¼	6¼	11¾
8	7½	1 1/16	18½	2½	15½	12	1⅜	14¼	8¾	27¼	7¼	14¼
10	9⅜	1¼	21½	2¾	18½	16	1⅜	16¼	9¾	31¼	8¼	16¼
12	11⅛	1 7/16	24	3⅛	21	20	1⅜	18¾	10¾	34¼	8¾	17¾
14 O.D.	12¼	1 9/16	25¼	3⅜	22	20	1½	20	11¼	36¼	9¼	18½
16 O.D.	14	1 13/16	27¾	3½	24¼	20	1⅝	22	12¼	40½	10¼	20½
18 O.D.	15¾	2	31	4	27	20	1⅞	23¾	13	45¼	11¾	24
20 O.D.	17½	2¼	33¾	4¼	29½	20	2	25¾	14¼	50	12¾	26
24 O.D.	21	2⅝	41	5½	35½	20	2½	30¼	17¾	59¾	15¼	30

True Y (see *Fig.* 38). The dimensions of this fitting should be the same as those of the 90-deg elbow with a back outlet in which the dimension of the back-outlet contact face to the intersecting center lines is the same as dimension *FF* of the 45-deg lateral. The contact face to intersecting center lines of each Y outlet is the same as dimension *AA* of the elbow.

[1] The dimensions given and the marking for sizes ½ to 2½ in., inclusive, are identical with those of the 1,500-lb fittings.

[2] The raised face of ¼ in. is not included in (1) thickness of flange, minimum, (2) center to flange edge, or (3) flange-edge-to-flange-edge dimensions.

[3] Where facings other than the ¼-in. raised face are used the center-to-flange-edge dimensions shall remain unchanged, and new center-to-contact-surface and contact-surface-to-contact-surface dimensions shall be established to suit the facing used.

[4] Reducing fittings shall have the same center-to-flange-edge dimensions as those of straight-size fittings of the largest opening.

The center-to-end dimensions of ring-joint fittings are obtained by adding the depth of groove as given in Table CXXVIII to the center-to-flange-edge dimensions.

Table CXXVIII.—Facing Dimensions for 900-lb Ring-joint Flanges*

(Table 42, ASA B16 e)

(All dimensions in inches)

(For cuts see Table CXIX, page 650)

Nominal pipe size	Pitch diameter of ring and groove	Width of ring	Height of ring		Height of flat on octagonal rings	Width of groove	Depth of groove[1]	Diameter of raised face for ring joint or lapped	Ring numbers*	Approximate distance between flanges of ring joints when ring is compressed[2]
			Oval	Octagonal						
	P	*A*	*B*	*H*	*C*	*D*	*L*	*K*		
1/2†	1 9/16	5/16	9/16	.....	...	11/32	1/4	2 3/8	R 12	5/32
3/4†	1 3/4	5/16	9/16	.....	...	11/32	1/4	2 5/8	R 14	5/32
1†	2	5/16	9/16	.....	...	11/32	1/4	2 13/16	R 16	5/32
1 1/4†	2 3/8	5/16	9/16	.....	...	11/32	1/4	3 3/16	R 18	5/32
1 1/2†	2 11/16	5/16	9/16	.....	...	11/32	1/4	3 5/8	R 20	5/32
2†	3 3/4	7/16	11/16	.....	...	15/32	5/16	4 7/8	R 24	1/8
2 1/2†	4 1/4	7/16	11/16	.....	...	15/32	5/16	5 3/8	R 27	1/8
3	4 7/8	7/16	11/16	.....	...	15/32	5/16	6 1/8	R 31	5/32
4	5 7/8	7/16	11/16	.....	...	15/32	5/16	7 1/8	R 37	5/32
5	7 1/8	7/16	11/16	.....	...	15/32	5/16	8 1/2	R 41	5/32
6	8 5/16	7/16	11/16	.....	...	15/32	5/16	9 1/2	R 45	5/32
8	10 5/8	7/16	11/16	.....	...	15/32	5/16	12 1/8	R 49	5/32
10	12 3/4	7/16	11/16	.....	...	15/32	5/16	14 1/4	R 53	5/32
12	15	7/16	11/16	.....	...	15/32	5/16	16 1/2	R 57	5/32
14 O.D.	16 1/2	5/8	7/8	13/16	5/16	21/32	7/16	18 3/8	R 62	5/32
16 O.D.	18 1/2	5/8	7/8	13/16	5/16	21/32	7/16	20 5/8	R 66	5/32
18 O.D.	21	3/4	1	15/16	5/16	25/32	1/2	23 3/8	R 70	3/16
20 O.D.	23	3/4	1	15/16	5/16	25/32	1/2	25 1/2	R 74	3/16
24 O.D.	27 1/4	1	1 5/16	1 1/4	1/2	1 1/16	5/8	30 3/8	R 78	7/32

Dimension *Q* is minimum flange thickness.

* For dimensions of rings, see Table CXXXVII. When ordering rings for nominal pipe sizes which may have either oval or octagonal-shaped rings, purchasers must specify oval- or octagonal-shaped rings as desired.

† The dimensions given and the marking for sizes 1/2 to 2 1/2 in., inclusive, are identical with those of the 1,500-lb ring-joint flanges.

[1] The depth of groove is added to the minimum flange thickness, except in sizes 1/2 to 1 1/2 in. inclusive, where the groove cuts 1/32 in. into the minimum flange thickness.

[2] For calculating the "laying length" of fittings with ring joints, the space dimensions given in the table must be added.

[3] This type of groove, which should be used for oval rings having widths of 1/4, 5/16, and 7/16 in., may be made with flat bottom at the option of the manufacturer.

[4] This type of groove should be used for rings having widths of 1/2 in. and greater.

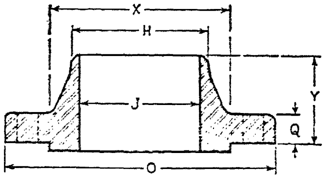
[5] The corner radius for octagonal rings and flat bottom grooves *r* is 1/32 in. for rings havi widths of 5/16 and 7/16 in., 1/16 in. for rings having widths 5/8 and 3/4 in., and 3/32 in. for rings having widths of 1 in. The tolerance on this dimension is ± 1/64 in. for the ring. This *r* is maximum for the groove.

Table CXXIX.—Dimensions of Steel Welding-neck Flanges for Primary-service Pressure Rating of 1,500 Psi Gage

(For complete pressure-temperature ratings see Tables CV to CVIII)

(Table 45, ASA B16 e)

(All dimensions in inches)



Nominal pipe size	Diameter of flange *O*	Thickness of flange,[1] minimum *Q*	Diameter of hub *X*	Hub diameter of beginning of chamfer[2,3,4] *H*	Length through hub[1] *Y*	Inside diameter of pipe[4] *J*	Diameter of circle	Number of bolts	Size of bolts
½	4¾	⅞	1½	0.84	2⅜		3¼	4	¾
¾	5⅛	1	1¾	1.05	2¾		3½	4	¾
1	5⅞	1⅛	2 1/16	1.32	2⅞		4	4	⅞
1¼	6¼	1⅛	2½	1.66	2⅞		4⅜	4	⅞
1½	7	1¼	2¾	1.90	3¼		4⅞	4	1
2	8½	1½	4⅛	2.38	4	To be specified by purchaser	6½	8	⅞
2½	9⅝	1⅝	4⅞	2.88	4⅛		7½	8	1
3	10½	1⅞	5¼	3.50	4⅝		8	8	1⅛
4	12¼	2⅛	6⅜	4.50	4⅞		9½	8	1¼
5	14¾	2⅞	7¾	5.56	6⅛		11½	8	1½
6	15½	3¼	9	6.63	6¾		12½	12	1⅜
8	19	3⅝	11½	8.63	8⅜		15½	12	1⅝
10	23	4¼	14½	10.75	10		19	12	1⅞
12	26½	4⅞	17¾	12.75	11⅛		22½	16	2
14 O.D.	29½	5¼	19½	14.00	11¾		25	16	2¼
16 O.D.	32½	5¾	21¾	16.00	12¼		27¾	16	2½
18 O.D.	36	6⅜	23½	18.00	12⅞		30½	16	2¾
20 O.D.	38¾	7	25¼	20.00	14		32¾	16	3
24 O.D.	46	8	30	24.00	16		39	16	3½

See Welding Bevel and Reducing Welding-neck Flanges in the introductory notes, p. 634.

[1] The raised face of ¼ in. is not included in thickness of flange, minimum, or in length through hub.

[2] The outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg.

[3] Dimension *H* corresponds to the outside diameters of pipe as given in ASA B36.10.

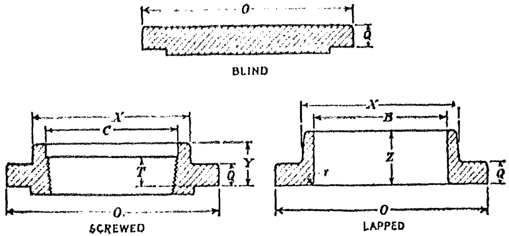
[4] See Tolerance for Welding-neck Flanges, in the introductory notes, p. 633.

TABLE CXXX. DIMENSIONS OF STEEL FLANGES FOR PRIMARY-SERVICE PRESSURE RATING OF 1,500 PSI GAGE

(For complete pressure-temperature ratings see Tables CV to CVIII)

(Table 44, ASA B16 e)

(All dimensions in inches)



Nominal pipe size	Outside diameter of flange O	Thickness of flange,[1] minimum Q	Diameter of hub[2] X	Diameter of bolt circle	Number of bolts	Size of bolts	Counter-bore screwed flange,[3] minimum C	Bore of lapped flange,[3] minimum B	Corner radius of bore of lapped flange and pipe r	Length through hub: Screwed[1] Y	Length through hub: Lapped Z	Thread length[1,4] T
½	4¾	⅞	1½	3¼	4	¾	0.93	0.93	⅛	1¼	1¼	⅞
¾	5⅛	1	1¾	3½	4	¾	1.14	1.14	⅛	1⅜	1⅜	1
1	5⅞	1⅛	2¹⁄₁₆	4	4	⅞	1.41	1.41	⅛	1⅝	1⅝	1⅛
1¼	6¼	1⅛	2½	4⅜	4	⅞	1.75	1.75	³⁄₁₆	1⅝	1⅝	1³⁄₁₆

TABLE CXXX.—(*Concluded*)

Nominal pipe size	Outside diameter of flange O	Thickness of flange,[1] minimum Q	Diameter of hub[2] X	Diameter of bolt circle	Number of bolts	Size of bolts	Counter-bore screwed flange,[3] minimum C	Bore of lapped flange,[3] minimum B	Corner radius of bore of lapped flange and pipe r	Length through hub, Screwed[1] Y	Length through hub, Lapped Z	Thread length[1,4] T
1½	7	1¼	2¾	4⅞	4	1	1.99	1.99	¼	1¾	1¾	1¼
2	8½	1½	4⅛	6½	8	⅞	2.50	2.50	5/16	2¼	2¼	1½
2½	9⅝	1⅝	4⅞	7½	8	1	3.00	3.00	5/16	2½	2½	1⅞
3	10½	1⅞	5¼	8	8	1⅛	3.63	3.63	⅜	2⅞	2⅞	2
4	12¼	2⅛	6⅜	9½	8	1¼	4.63	4.63	7/16	3 9/16	3 9/16	2¼
5	14¾	2⅞	7¾	11½	8	1½	5.69	5.69	7/16	4⅛	4⅛	2½
6	15½	3¼	9	12½	12	1⅜	6.75	6.75	½	4 11/16	4 11/16	2¾
8	19	3⅝	11½	15½	12	1⅝	8.75	8.81	½	5⅝	5⅝	3
10	23	4¼	14½	19	12	1⅞	10.88	10.94	½	6¼	7	3 5/16
12	26½	4⅞	17¾	22½	16	2	12.94	12.94	½	7⅛	8⅝	3⅝
14 O.D.	29½	5¼	19½	25	16	2¼	14.25	14.25	½	.....	9½	
16 O.D.	32½	5¾	21¾	27¾	16	2½	16.25	16.25	½	.....	10¼	
18 O.D.	36	6⅜	23½	30½	16	2¾	18.25	18.25	½	.....	10⅞	
20 O.D.	38¾	7	25¼	32¾	16	3	20.25	20.25	½	.....	11½	
24 O.D.	46	8	30	39	16	3½	24.25	24.25	½	.....	13	

For the dimensions of reducing screwed flanges see Table CIX.

Blind flanges should be faced as follows: (1) When the center portion of the inside face is raised, it need not be faced, but its diameter shall be at least 1 in. smaller than the inside diameter of the fitting given in Table CXXXI, (2) when the center portion of the inside face is depressed, it need not be faced, and its diameter shall not be greater than the inside diameter of the fitting given in Table CXXXI.

[1] The raised face of ¼ in. is not included in thickness of flange, minimum, length through hub, or thread length.

[2] This dimension is for large end of hub, which may be tapered 5 deg for draft.

[3] The dimensions given for the counterbore of the screwed flange and the bore of the lapped flange are based on the variation of the outside diameter (oversize) of steel pipe given in ASTM Specifications A-53, A-120, and API Specification 5-L. The tolerances on the bore of these flanges are +1/32 in. for sizes 10 in. and smaller, and +1/16 in. for sizes 12 in. and larger.

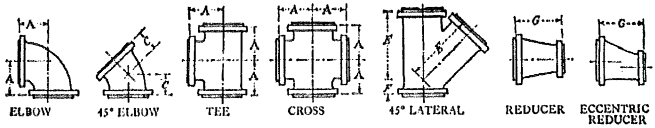
[4] Thread length given is proportionately longer than required for API or American standard for pipe threads and satisfies the general requirement for longer threads for the heavier pressures.

TABLE CXXXI.—CENTER-TO-FLANGE-EDGE DIMENSIONS OF STEEL-FLANGED FITTINGS FOR PRIMARY-SERVICE PRESSURE RATING OF 1,500 PSI GAGE

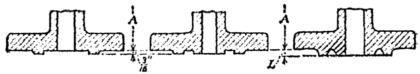
(For complete pressure-temperature ratings see Tables CV to CVIII)

(Table 46, ASA B16 e)

(All dimensions in inches)



Female, Groove, and Ring-Joint Facings for Fittings



Nominal pipe size	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of bolt circle	Number of bolts	Size of bolts	Flange edge: Center to flange edge elbow, tee, and cross[1,2,3] A	Flange edge: Center to flange edge 45-deg ell[1,2,3] C	Flange edge: Long center to flange edge lateral[1,2,3] E	Flange edge: Short center to flange edge lateral[1,2,3] F	Flange edge: Flange edge to flange edge reducer[1,2] G
½	½	5/16	4¾	⅞	3¼	4	¾	4	2¾			
¾	11/16	5/16	5⅛	1	3½	4	¾	4¼	3			
1	⅞	⅜	5⅞	1⅛	4	4	⅞	4¾	3¼	8¾	2¼	4½
1¼	1⅛	⅜	6¼	1⅛	4⅜	4	⅞	5¼	3¾	9¾	2¾	5¼

TABLE CXXXI.—(*Concluded*)

Nominal pipe size	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of bolt circle	Number of bolts	Size of bolts	Flange edge: Center to flange edge elbow, tee, and cross[1,2,3] A	Flange edge: Center to flange edge 45-deg ell[1,2,3] C	Flange edge: Long center to flange edge lateral[1,2,3] E	Flange edge: Short center to flange edge lateral[1,2,3] F	Flange edge: Flange edge to flange edge reducer[1,2] G
1½	1⅜	7/16	7	1¼	4⅞	4	1	5¾	4	10¾	3¼	5¾
2	1⅞	9/16	8½	1½	6½	8	⅞	7	4½	13	3¾	6¾
2½	2¼	11/16	9⅝	1⅝	7½	8	1	8	5	15	4¼	7¾
3	2¾	¾	10½	1⅞	8	8	1⅛	9	5½	17	4¾	8¾
4	3⅝	1	12¼	2⅛	9½	8	1¼	10½	7	19	5¾	10¼
5	4⅜	1⅛	14¾	2⅞	11½	8	1½	13	8½	23	7¼	13¼
6	5⅜	1 5/16	15½	3¼	12½	12	1⅜	13⅝	9⅜	24⅝	7⅞	14
8	7	1⅝	19	3⅝	15½	12	1⅝	16⅛	10⅝	29⅝	8⅞	16½
10	8¾	2	23	4¼	19	12	1⅞	19¼	11¾	35¾	10	19¾
12	10⅜	2 5/16	26½	4⅞	22½	16	2	22	13	40½	11¾	22½
14 O.D.	11⅜	2½	29½	5¼	25	16	2¼	24½	14	43¾	12¼	25¼
16 O.D.	13	2⅞	32½	5¾	27¾	16	2½	27	16	48	14½	27¾
18 O.D.	14⅝	3¼	36	6⅜	30½	16	2¾	30	17½	53	16¼	31
20 O.D.	16⅜	3⅝	38¾	7	32¾	16	3	32½	18½	57½	17½	33½
24 O.D.	19⅝	4¼	46	8	39	16	3½	38	20½	67	20¼	39¼

True Y (see Fig. 38). The dimensions of this fitting should be the same as those of the 90-deg elbow with a back outlet in which the dimension of the back-outlet contact face to the intersecting center lines is the same as dimension *FF* of the 45-deg lateral. The contact face to intersecting center lines of each Y outlet is the same as dimension *AA* of the elbow.

[1] The raised face of ¼ in. is not included in (1) thickness of flange, minimum, (2) center to flange edge, or (3) flange-edge-to-flange-edge dimensions.

[2] Where facings other than the ¼-in. raised face are used the center-to-flange-edge dimensions shall remain unchanged, and new center-to-contact-surface and contact-surface-to-contact-surface dimensions shall be established to suit the facing used.

[3] Reducing fittings shall have same center-to-flange-edge dimensions as those of straight-size fittings of the largest opening.

The center-to-end dimensions of ring-joint fittings are obtained by adding the depth of groove as given in Table CXXXII to the center-to-flange-edge dimensions.

Table CXXXII.—Facing Dimensions for 1,500-lb Ring-joint Flanges*

(For complete pressure-temperature ratings see Tables CV to CVIII)

(Table 48, ASA B16 e)

(All dimensions in inches)

(For cuts see Table CXIX, page 650)

Nominal pipe size	Pitch diameter of ring and groove	Width of ring	Height of ring		Height of flat on octagonal rings	Width of groove	Depth of groove[1]	Diameter of raised face for ring joint or lapped	Ring numbers*	Approximate distance between flanges of ring joints when ring is compressed[2]
			Oval	Octagonal						
	P	A	B	H	C	D	L	K		
1/2	1 9/16	5/16	9/16	.....	....	11/32	1/4	2 3/8	R 12	5/32
3/4	1 3/4	5/16	9/16	.....	....	11/32	1/4	2 5/8	R 14	5/32
1	2	5/16	9/16	.....	....	11/32	1/4	2 13/16	R 16	5/32
1 1/4	2 3/8	5/16	9/16	.....	....	11/32	1/4	3 3/16	R 18	5/32
1 1/2	2 11/16	5/16	9/16	.....	....	11/32	1/4	3 5/8	R 20	5/32
2	3 3/4	7/16	11/16	.....	....	15/32	5/16	4 7/8	R 24	1/8
2 1/2	4 1/4	7/16	11/16	.....	....	15/32	5/16	5 3/8	R 27	1/8
3	5 3/8	7/16	11/16	.....	....	15/32	5/16	6 5/8	R 35	1/8
4	6 3/8	7/16	11/16	.....	....	15/32	5/16	7 5/8	R 39	1/8
5	7 5/8	7/16	11/16	.....	....	15/32	5/16	9	R 44	1/8
6	8 5/16	1/2	3/4	11/16	5/16	17/32	3/8	9 3/4	R 46	1/8
8	10 5/8	5/8	7/8	13/16	5/16	21/32	7/16	12 1/2	R 50	5/32
10	12 3/4	5/8	7/8	13/16	5/16	21/32	7/16	14 5/8	R 54	5/32
12	15	7/8	1 1/8	1 1/16	3/8	29/32	9/16	17 1/4	R 58	3/16
14 O.D.	16 1/2	1	1 5/16	1 1/4	1/2	1 1/16	5/8	19 1/4	R 63	7/32
16 O.D.	18 1/2	1 1/8	1 7/16	1 3/8	9/16	1 3/16	11/16	21 1/2	R 67	5/16
18 O.D.	21	1 1/8	1 7/16	1 3/8	9/16	1 3/16	11/16	24 1/8	R 71	5/16
20 O.D.	23	1 1/4	1 9/16	1 1/2	5/8	1 5/16	11/16	26 1/2	R 75	3/8
24 O.D.	27 1/4	1 3/8	1 3/4	1 5/8	11/16	1 7/16	13/16	31 1/4	R 79	7/16

Dimension Q is minimum flange thickness.

* For dimensions of rings, see Table CXXXVII. When ordering rings for nominal pipe sizes which may have either oval- or octagonal-shaped rings, purchasers must specify oval- or octagonal-shaped rings as desired.

[1] The depth of groove is added to the minimum flange thickness, except in sizes 1/2 to 1 1/2 in. inclusive where the groove cuts [illegible] in. into the minimum flange thickness.

[2] [illegible] "laying length" of fittings with ring joints the space dimensions given in [illegible] must be added.

[3] This type of groove which should be used for oval rings having widths of 1/4, 5/16, and 7/16 in., may be made with flat bottom at the option of the manufacturer.

[4] This type of groove should be used for rings having widths of 1/2 in. and greater.

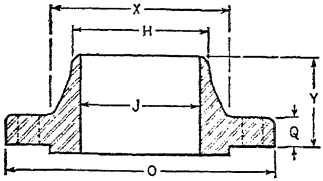
[5] The corner radius for octagonal rings and flat bottom grooves r is 1/32 in. for rings having widths of [illegible] in., [illegible] in. for rings having widths of 1/2, 5/8, and 7/8 in., and 3/32 in. for rings having widths of 1, [illegible], [illegible], and [illegible] in. The tolerance on this dimension is ± 1/64 in. for the ring. This r is maximum for the groove.

Table CXXXIII.—Dimensions of Steel Welding-neck Flanges for Primary-service Pressure Rating of 2,500 Psi Gage

(For complete pressure-temperature ratings see Tables CV to CVIII)

(Table 51, ASA B16 e)

(All dimensions in inches)



Nominal pipe size	Diameter of flange *O*	Thickness of flange,[1] minimum *Q*	Diameter of hub *X*	Hub diameter of beginning of chamfer[2,3,4] *H*	Length through hub[1] *Y*	Inside diameter of pipe[4] *J*	Diameter of bolt circle	Number of bolts	Size of bolts
½	5¼	1³⁄₁₆	1¹¹⁄₁₆	0.84	2⅞		3¼	4	¾
¾	5½	1¼	2	1.05	3⅛		3¾	4	¾
1	6¼	1⅜	2¼	1.32	3½		4¼	4	⅞
1¼	7¼	1½	2⅞	1.66	3¾		5⅛	4	1
1½	8	1¾	3⅛	1.90	4⅜	To be specified by purchaser	5¾	4	1⅛
2	9¼	2	3¾	2.38	5		6¾	8	1
2½	10½	2¼	4½	2.88	5⅝		7¾	8	1⅛
3	12	2⅝	5¼	3.50	6⅝		9	8	1¼
4	14	3	6½	4.50	7½		10¾	8	1½
5	16½	3⅝	8	5.56	9		12¾	8	1¾
6	19	4¼	9¼	6.63	10¾		14½	8	2
8	21¾	5	12	8.63	12½		17¼	12	2
10	26½	6½	14¾	10.75	16½		21¼	12	2½
12	30	7¼	17⅜	12.75	18¼		24⅜	12	2¾

See Welding Bevel and Reducing Welding-neck Flanges in the introductory notes, p. 634.

[1] The raised face of [illegible] in. is not included in thickness of flange [illegible] minimum or in length through hub.

[2] The outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg.

[3] Dimension *H* corresponds to the outside diameters of pipe as given in ASA B36.10.

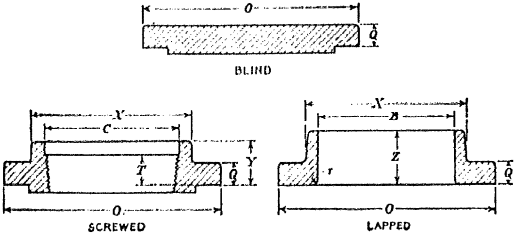
[4] See Tolerance for Welding-neck Flanges in the introductory notes, p. 633.

TABLE CXXXIV. DIMENSIONS OF STEEL FLANGES FOR PRIMARY-SERVICE PRESSURE RATING OF 2,500 PSI GAGE

(For complete pressure-temperature ratings see Tables CV to CVIII)

(Table 50, ASA B16 e)

(All dimensions in inches)



Nominal pipe size	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of hub[2]	Diameter of bolt circle	Number of bolts	Size of bolts	Counterbore screwed flange,[3] minimum	Bore of lapped flange,[3] minimum	Length through hub		Thread length[1,4]
									Screwed[1]	Lapped	
	O	*Q*	*X*				*C*	*B*	*Y*	*Z*	*T*
½	5¼	1³⁄₁₆	1¹¹⁄₁₆	3½	4	¾	0.93	0.93	1⁹⁄₁₆	1⁹⁄₁₆	1⅛
¾	5½	1¼	2	3¾	4	¾	1.14	1.14	1¹¹⁄₁₆	1¹¹⁄₁₆	1¾
1	6¼	1⅜	2¼	4¼	4	⅞	1.41	1.41	1⅞	1⅞	1⅜
1¼	7¼	1½	2⅞	5⅛	4	1	1.75	1.75	2¹⁄₁₆	2¹⁄₁₆	1½
1½	8	1¾	3⅛	5¾	4	1⅛	1.99	1.99	2⅜	2⅜	1¾

TABLE CXXXIV.—(*Concluded*)

Nominal pipe size	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of hub[2]	Diameter of bolt circle	Number circle	Size of bolts	Counterbore screwed flange,[3] minimum	Bore of lapped flange,[3] minimum	Length through hub		Thread length[1,4]
									Screwed[1]	Lapped	
	O	*Q*	*X*				*C*	*B*	*Y*	*Z*	*T*
2	9¼	2	3¾	6¾	8	1	2.50	2.50	2¾	2¾	2
2½	10½	2¼	4½	7¾	8	1⅛	3.00	3.00	3⅛	3⅛	2¼
3	12	2⅝	5¼	9	8	1¼	3.63	3.63	3⅝	3⅝	2½
4	14	3	6½	10¾	8	1½	4.63	4.63	4¼	4¼	2¾
5	16½	3⅝	8	12¾	8	1¾	5.69	5.69	5⅛	5⅛	3
6	19	4¼	9¼	14½	8	2	6.75	6.75	6	6	3¼
8	21¾	5	12	17¼	12	2	8.75	8.81	7	7	3¾
10	26½	6½	14¾	21¼	12	2½	10.88	10.94	9	9	4¼
12	30	7¼	17⅜	24⅜	12	2¾	12.94	12.94	10	10	4¾

For the dimensions of reducing screwed flanges, see Table CIX.

Blind flanges should be faced as follows: (1) When the center portion of the inside face is raised, it need not be faced, but its diameter shall be at least 1 in. smaller than the inside diameter of the fitting given in Table CXXXV, (2) when the center portion of the inside face is depressed, it need not be faced and its diameter shall not be greater than the inside diameter of the fitting given in Table CXXXV.

[1] The raised face of ¼ in. is not included in thickness of flange, minimum, length through hub, or thread length.

[2] This dimension is for large end of hub, which may be tapered 5 deg for draft.

[3] The dimensions given for the counterbore of the screwed flange and the bore of the lapped flange are based on the variation of the outside diameter (oversize) of steel pipe given in ASTM Specifications A-53, A-120, and API Specification 5-L. The tolerances on the bore of these flanges are 1/32 in. for sizes 10 in. and smaller, and 1/16 for sizes 12 in. and larger.

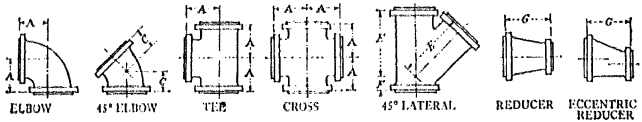
[4] Thread length given is proportionately longer than required for API or American Standard for Pipe Threads and satisfies the general requirement for longer threads for the heavier pressures.

TABLE CXXXV. DIMENSIONS OF STEEL-FLANGED FITTINGS FOR PRIMARY-SERVICE PRESSURE RATING OF 2,500 PSI GAGE

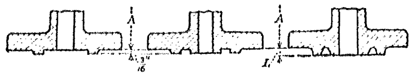
(For complete pressure-temperature ratings see Tables CV to CVIII)

(Table 52, ASA B16 e)

(All dimensions in inches)



Female, Groove, and Ring-Joint Facings for Fittings



Nominal pipe size	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of bolt circle	Number of bolts	Size of bolts	Flange edge: Center to flange edge elbow, tee, and cross[1,2,3] A	Center to flange edge, 45 deg ell[1,2,3] C	Long center to flange edge lateral[1,2,3] E	Short center to flange edge lateral[1,2,3] F	Flange edge to flange edge reducer[1,2] G
½	7/16	3/8	5¼	1 3/16	3½	4	¾	4 15/16				
¾	9/16	7/16	5½	1¼	3¾	4	¾	5⅛				
1	¾	½	6¼	1⅜	4¼	4	⅞	5 13/16	3¾			
1¼	1	⅝	7¼	1½	5⅛	4	1	6⅝	4			
1½	1⅛	11/16	8	1¾	5¾	4	1⅛	7 5/16	4½			

TABLE CXXXV.—(*Concluded*)

Nominal pipe size	Inside diameter of fitting	Metal thickness of fitting, minimum	Outside diameter of flange	Thickness of flange,[1] minimum	Diameter of bolt circle	Number of bolts	Size of bolts	Flange edge				
								Center to flange edge elbow, tee, and cross[1,2,3]	Center to flange edge, 45 deg ell[1,2,3]	Long center to flange edge lateral[1,2,3]	Short center to flange edge lateral[1,2,3]	Flange edge to flange edge reducer[1,2]
								A	*C*	*E*	*F*	*G*
2	1½	13/16	9¼	2	6¾	8	1	8⅝	5½	15	5	9
2½	1⅞	1	10½	2¼	7¾	8	1⅛	9¾	6	17	5½	10
3	2¼	1 3/16	12	2⅝	9	8	1¼	11⅛	7	19½	6½	11¼
4	2⅞	1 7/16	14	3	10¾	8	1½	13	8¼	22¾	7½	13
5	3⅝	1 13/16	16½	3⅝	12¾	8	1¾	15⅜	9¾	27	9	15¼
6	4⅜	2 1/16	19	4¼	14½	8	2	17¾	11¼	31	10¼	17½
8	5¾	2⅝	21¾	5	17¼	12	2	19⅞	12½	35	11½	20
10	7¼	3¼	26½	6½	21¼	12	2½	24¾	15¾	43	14½	25
12	8⅝	3 13/16	30	7¼	24⅜	12	2¾	27¾	17½	49	16	28½

[1] The raised face of ¼ in. is not included in (1) thickness of flange, minimum, (2) center to flange edge, or (3) flange-edge-to-flange-edge dimensions.

[2] Where facings other than the ¼-in. raised face are used, the center-to-flange-edge dimensions shall remain unchanged, and new center-to-contact-surface and contact-surface-to-contact-surface dimensions shall be established to suit the facing used.

[3] Reducing fittings shall have same center-to-flange-edge dimensions as those of straight-size fittings of the largest opening.

The center-to-end dimensions of ring-joint fittings are obtained by adding the depth of groove as given in Table CXXXVI to the center-to-flange-edge dimensions.

Table CXXXVI.—Facing Dimensions for 2,500-lb Ring-joint Flanges*

(Table 54, ASA B16 e)

(All dimensions in inches)

(For cuts see Table CXIX, page 650)

Nominal pipe size	Pitch diameter of ring and groove	Width of ring	Height of ring		Height of flat on octagonal rings	Width of groove	Depth of groove[1]	Diameter of raised face for ring joint or lapped	Ring numbers*	Approximate distance between flanges of ring joints when ring is compressed[2]
			Oval	Octagonal						
	P	A	B	H	C	D	L	K		
½	1 11/16	5/16	9/16	.....	...	11/32	¼	2 9/16	R 13	5/32
¾	2	5/16	9/16	.....	...	11/32	¼	2 7/8	R 16	5/32
1	2 3/8	5/16	9/16	.....	...	11/32	¼	3¼	R 18	5/32
1¼	2 27/32	7/16	11/16	.....	...	15/32	5/16	4	R 21	1/8
1½	3¼	7/16	11/16	.....	...	15/32	5/16	4½	R 23	1/8
2	4	7/16	11/16	.....	...	15/32	5/16	5¼	R 26	1/8
2½	4 3/8	½	¾	11/16	5/16	17/32	3/8	5 7/8	R 28	1/8
3	5	½	¾	11/16	5/16	17/32	3/8	6 5/8	R 32	1/8
4	6 3/16	5/8	7/8	13/16	5/16	21/32	7/16	8	R 38	5/32
5	7½	¾	1	15/16	5/16	25/32	½	9½	R 42	5/32
6	9	¾	1	15/16	5/16	25/32	½	11	R 47	5/32
8	11	7/8	1 1/8	1 1/16	3/8	29/32	9/16	13 3/8	R 51	3/16
10	13½	1 1/8	1 7/16	1 3/8	9/16	1 3/16	11/16	16¾	R 55	¼
12	16	1¼	1 9/16	1½	5/8	1 5/16	11/16	19½	R 60	5/16

Dimension Q is minimum flange thickness.

* For dimensions of rings, see Table CXXXVII. When ordering rings for nominal pipe sizes which may have either oval- or octagonal-shaped rings, purchasers must specify oval- or octagonal-shaped rings as desired.

[1] The depth of groove is added to the minimum flange thickness, except in sizes ½ to 1 in., inclusive, where the groove cuts [illegible] in. into the minimum flange thickness.

[2] For calculating the "laying length" of fittings with ring joints the space dimensions given in the table must be added.

[3] This type of groove which should be used for oval rings having widths of ¼, 5/16, and 7/16 in. may be made with flat bottom at the option of the manufacturer.

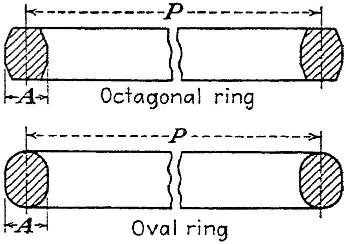
[4] This type of groove should be used for rings having widths of ½ in. and greater.

[5] The corner radius for octagonal rings and flat bottom grooves r is 1/32 in. for rings having widths of [illegible] and [illegible] in., [illegible] in. for rings having widths of ½, 5/8, ¾, and 7/8 in., and 3/32 in. for rings having widths of [illegible] and [illegible] in. The tolerance of this dimension is ± 1/64 in. for the ring. This r is maximum for the groove.

TABLE CXXXVII.—NUMBERS FOR RING-JOINT GASKETS AND GROOVES FOR STEEL FLANGES AND FLANGED FITTINGS

(Table 55, ASA B16 e)

(All dimensions in inches)

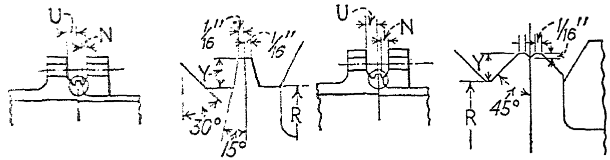


Number	Pitch diameter P	Width of ring A	Number	Pitch diameter P	Width of ring A
R 11	$1\frac{11}{32}$	$\times \frac{1}{4}$	R 45	$8\frac{5}{16}$	$\times \frac{7}{16}$
R 12	$1\frac{9}{16}$	$\times \frac{5}{16}$	R 46[1]	$8\frac{5}{16}$	$\times \frac{1}{2}$
R 13	$1\frac{11}{16}$	$\times \frac{5}{16}$	R 47[1]	9	$\times \frac{3}{4}$
R 14	$1\frac{3}{4}$	$\times \frac{5}{16}$	R 48	$9\frac{3}{4}$	$\times \frac{5}{16}$
R 15	$1\frac{7}{8}$	$\times \frac{5}{16}$	R 49	$10\frac{5}{8}$	$\times \frac{7}{16}$
R 16	2	$\times \frac{5}{16}$	R 50[1]	$10\frac{5}{8}$	$\times \frac{5}{8}$
R 17	$2\frac{1}{4}$	$\times \frac{5}{16}$	R 51[1]	11	$\times \frac{7}{8}$
R 18	$2\frac{3}{8}$	$\times \frac{5}{16}$	R 52	12	$\times \frac{5}{16}$
R 19	$2\frac{9}{16}$	$\times \frac{5}{16}$	R 53	$12\frac{3}{4}$	$\times \frac{7}{16}$
R 20	$2\frac{11}{16}$	$\times \frac{5}{16}$	R 54[1]	$12\frac{3}{4}$	$\times \frac{5}{8}$
R 21	$2\frac{27}{32}$	$\times \frac{7}{16}$	R 55[1]	$13\frac{1}{2}$	$\times 1\frac{1}{8}$
R 22	$3\frac{1}{4}$	$\times \frac{5}{16}$	R 56	15	$\times \frac{5}{16}$
R 23	$3\frac{1}{4}$	$\times \frac{7}{16}$	R 57	15	$\times \frac{7}{16}$
R 24	$3\frac{3}{4}$	$\times \frac{7}{16}$	R 58[1]	15	$\times \frac{7}{8}$
R 25	4	$\times \frac{5}{16}$	R 59	$15\frac{5}{8}$	$\times \frac{5}{16}$
R 26	4	$\times \frac{7}{16}$	R 60[1]	16	$\times 1\frac{1}{4}$
R 27	$4\frac{1}{4}$	$\times \frac{7}{16}$	R 61	$16\frac{1}{2}$	$\times \frac{7}{16}$
R 28[1]	$4\frac{3}{8}$	$\times \frac{1}{2}$	R 62[1]	$16\frac{1}{2}$	$\times \frac{5}{8}$
R 29	$4\frac{1}{2}$	$\times \frac{5}{16}$	R 63[1]	$16\frac{1}{2}$	$\times 1$
R 30	$4\frac{5}{8}$	$\times \frac{7}{16}$	R 64	$17\frac{7}{8}$	$\times \frac{5}{16}$
R 31	$4\frac{7}{8}$	$\times \frac{7}{16}$	R 65	$18\frac{1}{2}$	$\times \frac{7}{16}$
R 32[1]	5	$\times \frac{1}{2}$	R 66[1]	$18\frac{1}{2}$	$\times \frac{5}{8}$
R 33	$5\frac{3}{16}$	$\times \frac{5}{16}$	R 67[1]	$18\frac{1}{2}$	$\times 1\frac{1}{8}$
R 34	$5\frac{3}{16}$	$\times \frac{7}{16}$	R 68	$20\frac{3}{8}$	$\times \frac{5}{16}$
R 35	$5\frac{3}{8}$	$\times \frac{7}{16}$	R 69	21	$\times \frac{7}{16}$
R 36	$5\frac{7}{8}$	$\times \frac{5}{16}$	R 70[1]	21	$\times \frac{3}{4}$
R 37	$5\frac{7}{8}$	$\times \frac{7}{16}$	R 71[1]	21	$\times 1\frac{1}{8}$
R 38[1]	$6\frac{3}{16}$	$\times \frac{5}{8}$	R 72	22	$\times \frac{5}{16}$
R 39	$6\frac{3}{8}$	$\times \frac{7}{16}$	R 73[1]	23	$\times \frac{1}{2}$
R 40	$6\frac{3}{4}$	$\times \frac{5}{16}$	R 74[1]	23	$\times \frac{3}{4}$
R 41	$7\frac{1}{8}$	$\times \frac{7}{16}$	R 75[1]	23	$\times 1\frac{1}{4}$
R 42[1]	$7\frac{1}{2}$	$\times \frac{3}{4}$	R 76	$26\frac{1}{2}$	$\times \frac{5}{16}$
R 43	$7\frac{5}{8}$	$\times \frac{5}{16}$	R 77[1]	$27\frac{1}{4}$	$\times \frac{5}{8}$
R 44	$7\frac{5}{8}$	$\times \frac{7}{16}$	R 78[1]	$27\frac{1}{4}$	$\times 1$
			R 79[1]	$27\frac{1}{4}$	$\times 1\frac{3}{8}$

The edge of each flange and the outside circumference of each ring shall carry corresponding identification marks: *i.e.*, R11, R45, etc.

[1] When ordering rings for nominal pipe sizes which may have either oval- or octagonal-shaped rings, purchasers must specify oval- or octagonal-shaped rings as desired.

TABLE CXXXVIII.—TYPICAL FACING DIMENSIONS FOR SARGOL[1] AND SARLUN[2] JOINTS, 150- TO 2,500-LB FLANGES
(All dimensions in inches)



Patented Sarlun Modified Sargol

Nominal pipe size	Basic raised face, outside diameter *B*	Height of face		Height of front hub		Height of welding projections	
		Sargol *U*	Sarlun *U*	Sargol *N*	Sarlun *N*	Sargol *Y*	Sarlun *Y*
2½	4⅛	½	11/16	¼	5/16	⅛	5/16
3	5	⅝	11/16	5/16	5/16	3/16	5/16
4	6 3/16	¾	¾	7/16	13/32	5/16	5/16
5	7 5/16	⅞	⅞	½	½	⅜	⅜
6	8½	1	1	9/16	9/16	⅜	⅜
8	10⅝	1⅛	1⅛	⅝	⅝	⅜	⅜
10	12¾	1¼	1¼	⅝	¾	⅜	⅜
12	15	1 5/16	1 5/16	⅝	13/16	⅜	⅜
14	16¼	1⅜	1⅜	⅝	⅞	⅜	⅜
16	18½	1⅜	1⅜	⅝	⅞	⅜	⅜

[1] Dimensions of modified Sargol joint.
[2] Dimensions of Sarlun facings recommended by Sargent and Lundy, Inc.

ASTM Standard Specifications for
CARBON-STEEL CASTINGS FOR VALVES, FLANGES, AND FITTINGS FOR HIGH-TEMPERATURE SERVICE

Serial Designation A95-44 (ASA G17.1)

Abstracted[1]

These specifications cover carbon-steel castings for valves, flanges, and fittings for high-temperature service. For pressure ratings at various temperatures, reference should be made to American Steel Flanged Standards, ASA B16e, abstracted on page 626. Material suitable for welded fabrication is covered by Tentative Specifications for Carbon-Steel Castings Suitable for Fusion Welding for Service at Temperatures up to 850 F, ASTM Designation A216, abstracted on page 685.

The steel shall be made by the electric-furnace, open-hearth, or other process approved by the purchaser and shall conform to the following requirements as to chemical composition and physical properties:

[1] For complete specification, reference may be made to ASTM A95 (see note, p. 369).

Chemical Composition

Carbon, per cent	0.15 to 0.45
Manganese, min, per cent	0.50
Silicon, min, per cent	0.20
Phosphorus, max, per cent	0.05
Sulphur, max, per cent	0.06

Minimum Physical Properties

Tensile strength, psi	70,000
Yield point, psi	36,000
Elongation in 2 in., per cent	22
Reduction of area, per cent	30

Heat Treatment.—All castings shall receive a heat treatment proper to the design and chemical composition of the castings. Unless otherwise specified by the purchaser, heat treatment shall consist of full annealing or of normalizing followed by full annealing or drawing back to a temperature below the critical range (see ASTM E44 and page 324 for heat-treating procedures).

Tension Tests.—One tension test shall be made from each melt in each heat-treatment charge and, when specified, from each casting weighing 500 lb or over. For small castings and where the design of the casting is such that test bars should not be attached to the castings, the test bars shall be cast attached to special blocks.

Bend Tests.—One bend test shall be made, if specified on the purchase order, from each melt in each heat-treatment charge and, when specified, from each casting weighing 500 lb or over. The bend-test specimen shall stand being bent cold through an angle of 90 deg around a pin 1 in. in diameter, without cracking on the outside of the bent portion.

Hydrostatic Tests.—Each casting shall be tested after machining to the hydrostatic pressure specified in the table below and shall show no leaks. Castings ordered under these specifications for working pressures other than those listed below shall be tested to such pressures as may be agreed upon:

Primary Service Pressure Rating at 750 F, Psi	Standard Hydrostatic Test Pressure, Psi
100 (150 lb at 500 F)	350
300	750
400	1,000
600	1,500
900	2,000
1,500	3,500

Magnetic Particle Testing.—If specified in the inquiry, contract, or order, and when mutually agreed upon by the manufacturer and the purchaser, castings ordered to these specifications may be subject to magnetic particle inspection. When so specified, the magnetic particle inspection shall be in accordance with the Tentative Methods of Magnetic Particle Testing of Commercial Steel Castings, ASTM Designation A272.

Marking.—Pressure containing castings, made in accordance with these specifications, shall be marked for identification with the melt number or melt identification. Marking shall be in such position as not to injure the usefulness of the castings.

Finished products shall be marked in accordance with the Standard Marking System for Valves, Fittings, Flanges, and Unions (No. SP-25) of the Manufacturers Standardization Society of the Valve and Fittings Industry (see abstract, page 687).

ASTM Standard Specifications for ALLOY-STEEL CASTINGS FOR VALVES, FLANGES, AND FITTINGS FOR SERVICE AT TEMPERATURES FROM 750 TO 1100 F

Serial Designation A157-44

Abstracted[1]

These specifications cover alloy-steel castings for valves, pipe flanges, and pipe fittings intended for service at metal temperatures from 750 to 1100 F. For alloy castings suitable for welding, see abstract of ASTM Specification A217, abstracted on page 686. Carbon-steel castings for high-temperature service are covered by ASTM Specification A95, abstracted on page 680, and ASTM Specification A216, abstracted on page 685. The latter specification covers carbon steel suitable for fusion welding.

Eleven grades of alloy material are covered which have been rather extensively used for valves, flanges, and fittings. Not all are suitable for the entire temperature range.

Chemical Composition.—Each alloy shall conform to the chemical requirements of the accompanying table or to other chemical requirements as specified in the purchase order. Alloys have been numbered so that similar compositions in the corresponding specifications for pipe, forgings, and bolting for high-temperature service bear the same number for alloys numbered 10 or below.

Physical Properties.—The castings shall conform to the following minimum requirements as to tensile properties at room temperature:

PHYSICAL REQUIREMENTS

	Tensile strength, min, psi.	Yield point, min, psi.	Elongation in 2 in., min, per cent	Reduction of area, min, per cent
Grade C 1	70,000	45,000	22	35
Grade C 3A	90,000	60,000	18	30
Grade C 3B	90,000	60,000	18	30
Grade C 4	100,000	65,000	18	30
Grade C 5A	90,000	60,000	18	30
Grade C 5B	90,000	60,000	18	30
Grade C 6	85,000	55,000	20	40
Grade C 11	100,000	65,000	18	30
Grade C 12	90,000	60,000	18	30
Grade C 9	70,000	30,000	35	40
Grade C 10	65,000	30,000	30	35

Heat Treatment.—All castings shall receive heat treatment suitable to their design and chemical composition. Heat treatment shall be performed before

[1] For complete specification, reference may be made to ASTM A157 (see note, p. 369).

Chemical Requirements

Type	Ferritic steels					
Identification symbol	C 1	C 3A	C 3B	C 4	C 5A	C 5B
Grade	Carbon molybdenum	Chromium-molybde-		num	4 to 6 per cent chromium	4 to 6 per cent chromium-silicon-molybdenum
Carbon, per cent	0.35 max	0.30 max	0.30 max	0.45 max	0.30 max	0.30 max
Manganese, per cent	1.00 max	1.00 max	1.00 max	1.00 max	1.00 max	1.00 max
Phosphorus, max, per cent	0.05	0.05	0.05	0.05	0.05	0.05
Sulphur, max, per cent	0.06	0.06	0.06	0.06	0.06	0.06
Silicon, per cent	0.20 min	0.75 max	0.75 to 1.25	0.20 min	0.75 max	0.75 to 1.50
Nickel, per cent				0.75 to 1.50		
Chromium, per cent		1.50 to 2.25	1.50 to 2.25	0.50 to 1.00	4.00 to 6.50	4.00 to 6.50
Molybdenum, per cent	0.40 min	0.45 to 0.65	0.45 to 0.65	0.30 to 0.60	0.40 to 0.65[c]	0.40 to 0 65[c]
Tungsten, per cent		....	....	....	0.80 to 1.25[c]	0.80 to 1.25[c]

Type	Ferritic steels			Austenitic steels	
Identification symbol	C 6	C 11	C 12	C 9[a]	C 10
Grade	13 per cent chromium	Nickel-chromium	8 to 10 per cent chromium-molybdenum	18 per cent chromium, 8 per cent nickel	20 per cent nickel, 8 per cent chromium
Carbon, per cent	0.15 max	0.45 max	0.30 max	0.15 max	0.35 max
Manganese, per cent	0.75 max	1.00 max	1.00 max	1.00 max	1.50 max
Phosphorus, max, per cent	0.05	0.05	0.05	0.05	0.05
Sulphur, max, per cent	0.05	0.06	0.06	0.05	0.05
Silicon, per cent	1.00 max	0.02 min	1.00 max	2.00 max	2.00 max
Nickel, per cent	0.80 max	1.00 to 2.25	....	8.00 min	19.0 to 22.0
Chromium, per cent	11.5 to 13.5	0.50 to 1.00	8.00 to 10.00	18.00 min	8.00 to 10.00
Molybdenum, per cent	0.50 max	0.20 to 0.40[b]	1.10 to 1.50	As agreed upon	
Tungsten, per cent	....	0.40 to 0.80[b]			

[a] For the more severe general corrosive conditions, and when so specified, the carbon content shall not exceed 0.07 per cent.

The addition of such elements as molybdenum, tungsten, titanium, columbium, and vanadium for purposes of stabilization shall be a matter of agreement between the manufacturer and the purchaser.

When the purchaser requires free-machining or better nonseizing, nongalling properties, the composition may contain suitable combinations of selenium and phosphorus or molybdenum and sulphur:

Selenium and phosphorus:

Selenium, per cent	0.20 to 0.35
Phosphorus, max, per cent	0.17

Molybdenum and sulphur:

Molybdenum, per cent	0.40 to 0.80
Sulphur, per cent	0.20 to 0.40

[b] Either molybdenum or tungsten may be used, if desired.

[c] Either molybdenum or tungsten shall be used.

machining, except in instances where reheat treating is necessary. Ferritic steels may be fully annealed, fully annealed and drawn, or normalized and drawn, or any combination thereof, at the option of the manufacturer unless the method is made a matter of agreement by the purchaser. The austenitic steels shall be given a stabilizing treatment consisting of heating to the proper stabilizing temperature and holding at that temperature for sufficient time, followed by rapid air cooling or by quenching in a liquid medium (ASTM E44 and page 324).

Tension Tests.—One tension test shall be made from each melt in each heat-treatment charge and, when specified by the purchaser, from each casting weighing 500 lb or over. For castings weighing less than 500 lb, the test bars shall be cast attached to special blocks.

Bend Tests.—When specified in the purchase order, one bend test shall be made from each melt in each heat-treatment charge and, when specified from each casting weighing 500 lb or over. The bend-test specimens for all grades, excepting Grade C9, shall stand being bent cold through an angle of 90 deg around a pin 1 in. in diameter without cracking on the outside of the bent portion. Grade C9 shall stand being bent 120 deg around a 1 in. diameter pin.

Hydrostatic Tests.—Each casting shall be tested after machining to the hydrostatic pressure specified in the following table and shall show no leaks. Castings ordered under these specifications, for working pressures other than those listed below, shall be tested to such pressures as may be agreed upon by the manufacturer and purchaser.

Primary-service Pressure Rating at 750 F, Psi	Standard Hydrostatic-test Pressure, Psi
100	350
300	750
400	1,000
600	1,500
900	2,000
1,500	3,500

Magnetic Particle Testing.—See ASTM Specification A95, page 681.

Radiographic or Destruction Tests.—When so specified in the purchase order, the purchaser may, at his own expense, examine the castings for internal defects by means of X ray or gamma rays of radium, and castings showing injurious defects shall be rejected.

When so specified in the purchase order, the purchaser may, at his own expense, select representative castings from each melt or heat-treatment charge and crush or cut up and etch or otherwise prepare the sections for examination for internal defects. Should injurious defects be found which evidence unsound steel or faulty foundry technique, all the castings from that particular pattern, melt, and heat-treatment charge may be rejected. All the rejected castings, including those cut up, shall be replaced by the manufacturer, without charge. Coarse-grained fracture or other evidence of improper treatment shall be cause for reheat treating.

Marking.—Pressure-containing castings shall be marked for material identification with the symbols for the grade of steel, C1, C3A, etc., and the melt number or melt identification. Finished products shall be marked in accordance with the Standard Marking System for Valves, Fittings, Flanges, and Unions (see abstract MSS Standard Practice SP25, page 687).

ASTM Tentative Specifications for CARBON-STEEL CASTINGS SUITABLE FOR FUSION WELDING FOR SERVICE AT TEMPERATURES UP TO 850 F

Serial Designation A216-44T

Abstracted[1]

These specifications cover carbon-steel castings for valves, flanges, fittings, or other pressure-containing parts suitable for assembly with other castings or wrought-steel parts by fusion welding. For alloy castings suitable for fusion welding, see abstract of ASTM Specification A217, page 686. In addition to Grades WCA and WCB for general use, Grades EWCC and EWCD for pressure-containing turbine castings have been included in these specifications as alternates.

The steel shall be made by the open-hearth, electric-furnace, or other process approved by the purchaser and shall conform to the following requirements as to chemical composition and physical properties.

Chemical Composition

	Grade WCA	Grade WCB	Grade EWCC	Grade EWCD
Carbon, max, per cent	0.25[a]	0.35[a]	0.30[a]	0.35
Manganese, max, per cent	0.70[a]	0.70[a]	0.60[a]	0.70
Phosphorus, max, per cent	0.05	0.05	0.05	0.05
Sulphur, max, per cent	0.06	0.06	0.06	0.06
Silicon, max, per cent	0.60	0.60	0.60	0.60

[a] For each reduction of 0.01 per cent below the specified maximum carbon content, an increase of 0.04 per cent manganese above the specified maximum will be permitted up to a maximum of 1.10 per cent for WCA and WCB and to a maximum of 1.00 per cent for EWCC and EWCD.

Physical Properties

	Grade WCA	Grade WCB	Grade EWCC	Grade EWCD
Tensile strength, min, psi	60,000	70,000	60,000	65,000
Yield point, min, psi	30,000	36,000	30,000	35,000
Elongation in 2 in., min, per cent	24	22	24	20
Reduction of area, min, per cent	35	35	35	30

Heat Treatment.—Unless otherwise specified by the purchaser, heat treatment shall consist of full annealing, normalizing, or of normalizing followed by drawing back to at least 1100 F (see ASTM E44 and page 324).

Tension Tests.—One tension test shall be made from each melt in each heat-treatment charge and, when specified, from each casting weighing 500 lb net or

[1] For complete specification, reference may be made to ASTM A216 (see note, p. 369). Abstract includes Emergency Alternate Provisions, issued Apr. 6, 1942.

over. If agreed upon by the manufacturer and the purchaser, tension test specimens may be cut from heat-treated castings instead of from test bars.

Bend Tests.—Bend tests shall be made only when so specified in the order (see abstract of ASTM A95 for detailed requirements, page 681).

Hydrostatic Tests.—Castings of grades WCA, EWCC, and EWCD shall be tested to such pressures as may be agreed upon by the manufacturer and the purchaser. Grade WCB castings ordered for working pressures given in ASA B16e shall be tested to the hydrostatic pressures given therein (see page 629).

Magnetic Particle Testing.—See ASTM Specification A95, page 681.

Marking.—Pressure-containing castings shall be marked with the appropriate symbol for the grade of steel, the melt number, or melt identification. Finished products shall be marked in accordance with MSS Standard Practice SP-25 (see abstract, page 687).

ASTM Tentative Specifications for ALLOY-STEEL CASTINGS SUITABLE FOR FUSION WELDING FOR SERVICE AT TEMPERATURES FROM 750 F TO 1100 F

Serial Designation A217-44T

Abstracted[1]

These specifications cover alloy-steel castings for valves, flanges, fittings, and other pressure-containing parts suitable for assembly with other castings or wrought-steel parts by fusion welding. For carbon-steel castings suitable for fusion welding, see abstract of ASTM Specification A216, page 685. In addition to the four grades for general purposes, a new grade of weldable alloy-steel casting for pressure-containing turbine castings has been included as an alternate.

The steel shall be made by the open-hearth, electric-furnace, or other approved process and shall conform to the following chemical composition and physical properties:

CHEMICAL COMPOSITION

	Grade WC1	Grade WC2	Grade WC4	Grade EWC5
Carbon, per cent	0.30[a] max	0.25[a] max	0.20 to 0.30	0.30[a] max
Manganese, per cent	0.70[a] max	0.70[a] max	0.40 to 0.70	0.70[a] max
Phosphorus, max, per cent	0.05	0.05	0.05	0.05
Sulphur, max, per cent	0.06	0.06	0.06	0.06
Silicon, per cent	0.60 max	0.60 max	0.25 to 0.55	0.60
Nickel, per cent	....		0.75 to 1.05	
Chromium, per cent	....	....	0.40 to 0.70	
Molybdenum, per cent	0.40 to 0.60	0.40 to 0.60	0.30 to 0.45	0.40 to 0.60

[a] For each reduction of 0.01 per cent below the specified maximum carbon content, an increase of 0.04 per cent manganese above the specified maximum will be permitted up to a maximum of 1.10 per cent for Grades WC1, WC2, and WC4, and to a maximum of 1.00 for EWC5.

[1] For complete specification, reference may be made to ASTM A217 (see note, p. 369). Abstract includes Emergency Alternate Provisions, issued Apr. 6, 1942.

PHYSICAL PROPERTIES

	Grade[1] WC1A	Grade WC1	Grade WC2	Grade WC4	Grade EWC5
Tensile strength, min. psi	70,000	70,000	65,000	80,000	65,000
Yield point, min. psi	40,000	45,000	35,000	55,000	35,000
Elongation in 2 in., min. per cent	20	22	24	20	20
Reduction of area, min, per cent	35	35	35	35	30

[1] Grade WC1 when ordered in the full-annealed condition is designated as WC1A.

Heat Treatment.—Unless otherwise specified by the purchaser, heat treatment shall consist of full-annealing, normalizing, or of normalizing followed by drawing back to a temperature between 1100 F and 1250 F (see ASTM E44 and page 324 for heat-treating procedures).

Tension Tests.—One tension test shall be made from each heat-treatment charge and, when specified, each casting weighing 500 lb net or more. If agreed upon by the manufacturer and the purchaser, tension test specimens may be cut from heat-treated castings instead of from test bars.

Bend Tests.—Bend tests shall be made only when so specified in the order (see abstract of ASTM A95 for detailed requirements, page 681).

Hydrostatic Tests.—Castings ordered for working pressures given in ASA B16e shall be tested to the hydrostatic pressures given therein for carbon molybdenum and equivalent alloy steels (see page 630). Castings ordered for working pressures other than those noted above shall be tested to such pressures as may be agreed upon by the manufacturer and the purchaser.

Magnetic Particle Testing.—See ASTM Specification A95, page 681.

Radiographic or Destruction Tests.—When so specified in the order, the purchaser may examine castings for internal defects by means of X ray, or gamma rays of radium. Castings showing injurious defects as defined by standards agreed upon shall be rejected or repaired with the approval of the purchaser. Or, when so specified in the order, the purchaser may at his own expense select representative castings and section them for examination of internal defects. If injurious defects that evidence unsound steel or faulty foundry technique are found, all the castings from that particular pattern melt and heat-treatment charge may be rejected.

Marking.—Pressure-containing castings shall be marked with the applicable symbol for the grade of steel, and the melt number or melt identification. Finished products shall be marked in accordance with MSS Standard Practice SP-25 (see abstract, page 687).

MSS Standard Marking System for VALVES, FITTINGS, FLANGES, AND UNIONS

MSS Standard Practice SP-25

Abstracted[1]

This standard, which covers a comprehensive system for marking valves, fittings, flanges, and unions, was developed by the

[1] Abstract contains revisions of Addendum 2, adopted June 24, 1940.

Manufacturers Standardization Society of the Valve and Fittings Industry.[1] It has been approved by the National Association of Purchasing Agents, Inc., and appropriate committees of the American Petroleum Institute for submission to the American Standards Association as an American Standard. Marking requirements of the various ASA and API standards for piping already conform to this marking system. Special markings such as relieving capacity, code stamps, etc., required for safety or relief valves, regulating valves, and other special designs are not covered by this standard, although it is recommended that this marking system be used as far as possible for such special valves. Sections I and II of MSS SP-25 are given in their entirety here, but Section III, which covers specific requirements for marking individual lines of valves and fittings, is not reproduced. It should be noted that certain lines are exempt from the complete marking called for by the general rules, for example: lower pressure screwed fittings including 125-lb cast iron, 150-lb malleable iron, and 125- and 250-lb nonferrous lines. These items are marked only with the manufacturer's name or trademark in conformity with marking requirements in the applicable ASA and MSS dimensional standards.

Section I. General Requirements

1. Scope.—This marking system applies to the various classes of valves (including cocks), flanges, fittings, and unions, hereinafter termed "products," of whatever material made, which manufacturers recognize as regularly available patterns and which are specifically covered by the tables in Section III of this standard. The term "valve" applies to such types as gate, globe, angle, cross, needle, check of both the lift and swing type and stop check. The term "valve" does not apply to pop safety, relief, regulating, or such other designs as require special markings to indicate the particular class of service for which the valve has been produced. It is recommended, however, that this marking system be used in so far as possible for such special designs.

2. Application.—This marking system shall supersede the marking provisions of all existing product standards that do not conform hereto and shall be used where applicable to all future standards. All products falling within the scope of this marking system and produced after the effective date hereof shall be marked in accordance herewith.

3. Mandatory Markings.—Each product of a size and shape permitting legible marking shall be marked in accordance with the general provisions of Section I and also according to those specific requirements of Section II cited in the tables, Section III of this standard (not reproduced here).

[1] Copies of the complete standard and of other MSS standard practice specifications and standards may be obtained from the Manufacturers Standardization Society of the Valve and Fittings Industry, 420 Lexington Ave., New York 17, N.Y.

4. Marking to Be Omitted in Following Order.—(*a*) On product of small size or shape, which will not permit all required markings, they may be omitted to the degree which conditions require in the sequence of:

(1) Size.
(2) Thread identification.
(3) Valve trim identification.
(4) Temperature rating.
(5) Material.
Steel products:
(*a*) Steel.
(*b*) Melt number.
(*c*) Alloy symbol.
(6) Pressure.
(7) Manufacturer's name or trade-mark.

(*b*) No marking is mandatory on "finished" nonferrous products because the processes used to produce the "finish" preclude preservation of markings to a degree that will permit correct deciphering in all cases.

5. Method of Applying Marking. (*a*) *Fittings, Flanges, and Unions.*—All required markings shall be cast or stamped upon the exterior surface of the product.

(*b*) *Valves.*—(1) The following markings when required shall be cast or stamped on valve bodies:

Manufacturer's name or trade-mark.
Pressure.
Material.
Thread identification.

(2) Trim symbols shall not be applied to valve bodies but shall be confined to identification plates securely attached to the valves. All other markings, when required, may be applied either on the valve bodies or on identification plates or on both at the option of the manufacturer, except as mentioned below.

(3) Steel valves, except certain Bar-stock Valves and Lubricated-plug Valves mentioned below, are required to have identification plates on which must appear

Manufacturer's name or trade-mark.
Pressure.
Body material.
Temperature designation (note exception in Section II, Rule 4).
Valve trim identification.

(4) The melt identification must appear on all pressure-containing steel castings.

(5) Steel Lubricated Plug Valves in which the plugs are of the same material as the bodies and Steel Bar-stock Valves need not have identification plates and all mandatory marking may be cast or stamped upon the valve bodies.

6. Lined Valves and Fittings.—Valves and fittings made of one material and lined with another are in common use. Such products shall carry the regular markings specified by this standard and additional markings which will indicate whether partially or completely lined and the material used for lining. These additional markings shall be cast or stamped on the products or marked on plates or tags securely fastened to the products.

7. Bolting.—Alloy-steel bolts, screws, studs, and nuts used with products covered by this standard shall be marked with the symbols designated in ASTM Specifications for bolting materials to identify the grade of steel.

8. Ring Joints.—Flanges grooved for standard ring joints and the rings to be used with them shall be marked with the letter *R* and the corresponding ring number. This identification shall be placed on the edges of the flanges and on the outside surface of the rings. For ring numbers, see Table CXXXVII.

9. Special Identification.—Products made to special designs or to meet particular requirements not available in regular lines may carry special symbols to distinguish such lines from all others. Nothing in this standard shall be construed as prohibiting the use of additional markings such as catalogue reference numbers, pattern numbers, patent number dates, etc., provided the markings are applied in such manner as to avoid confusion with the standard marking provided herein.

Section II. Rules for Marking

(Tables for specific applications given in Section III are not reproduced here.)

Rule 1. Manufacturer's Name or Trade-mark.—All products falling within the scope of this standard shall be marked with the manufacturer's name or trademark, except as set forth in Section I–4*a* and 4*b*, under General Requirements.

Rule 2. Pressure Designation (Service Rating). (*a*) *Service Symbols.*—The following letters shall be used as symbols to signify class of service, with exceptions given in Rule 25. These symbols may be used in any order:

A to signify air.
G to signify gas.
L to signify liquid.
O to signify oil.
S to signify steam.
W to signify water.

(*b*) *Steam Service Ratings* (*Primary*).—If the primary-service pressure rating is for steam and steam pressure marking only is used, no symbol in addition to numerals is necessary, and the markings shall consist of the numerals comprising the primary steam rating in pounds per square inch. It is permissible but not mandatory to supplement primary steam pressure marking by secondary-service pressure marking. If the steam pressure marking on a fitting or a valve part is supplemented by secondary-service pressure marking comprised of numerals, each service marking shall be accompanied by symbols identifying the class of service (see Rule 2*a*). For example: Valve bodies marked with steam ratings only are not required to use the symbol *S*, but when steam ratings and supplementary ratings are marked on the same part or on the same identification plate, then both must be accompanied by service symbols.

(*c*) *Service Ratings Other Than Steam* (*Primary*).—When the primary-service pressure rating is other than steam, numerals comprising the pressure in pounds per square inch supplemented by one or more of the symbols identifying the class of service shall be used.

(*d*) *Test Pressure.*—The inclusion in the marking of numerals indicating the maximum shell test pressure is prohibited, unless specifically requested or required by the purchaser.

Rule 3. Material Designation. (*a*) *Carbon-steel.*—All carbon-steel products shall be marked with the word "steel." All pressure-containing carbon-steel castings shall be marked with the melt number or melt identification. All forged or rolled carbon-steel flanges and flanged fittings shall be marked with the symbols designated in ASTM Specifications to indicate the grade of steel. A manufacturer may supplement these mandatory markings with any trade desig-

nation which he may wish to employ for his individual grade of steel, except that confusion with the symbols herein set up shall be avoided.

(*b*) *Alloy-steel.*—All alloy-steel products shall be marked with the word "steel." All pressure-containing alloy-steel castings shall be marked with the melt number or melt identification. All pressure-containing alloy-steel parts shall be marked with the symbols designated in ASTM Specifications to indicate the grade of steel. A manufacturer may supplement these mandatory markings with any trade designation which he may wish to employ for his individual grade of steel, except that confusion with the symbols herein set up shall be avoided.

(*c*) *Malleable Iron.*—Where required in this standard, malleable iron shall be indicated by the letters MI applied in such manner as may avoid confusion with other markings. M, MAL, or MALL are in common use and may be continued throughout the life of existing equipment.

(*d*) *Cast Iron.*—A material marking is not required for cast iron.

(*e*) *Nonferrous.*—Nonferrous alloys other than brass or bronze shall be indicated by an appropriate material symbol. Where applicable, the symbols listed in Rule 5, Valve Trim Identification, shall be used.

Rule 4. Temperature Designation (**Service Rating**). (*a*) *Products Marked with a Primary-service Pressure Other Than Steam.*—Every product which carries a primary-service pressure marking other than steam and which is recommended by the manufacturer for temperatures higher than normal atmospheric shall be marked with numerals comprising the temperature recommended followed by the letter F to indicate degrees Fahrenheit.

(*b*) *Steel Products, Carbon, and Alloy.*—No temperature marking is required on carbon-steel and alloy-steel products made of materials given in American Standard B16e which are marked with a primary steam pressure rating, unless the temperatures recommended by the manufacturer are different from those given in ASA B16e.

If the temperature recommendations are different from the foregoing, each product shall be marked with the numerals comprising the temperature recommended by the manufacturer (at the marked primary steam rating) followed by the letter F to indicate degrees Fahrenheit.

Certain types of carbon or alloy-steel products are furnished with various trim materials, for example, steel unions with bronze seats. The temperature at which such products may be used depends upon the materials of the trim rather than upon the material of the structure itself.

(*c*) *Malleable-iron Products.*—No temperature marking is mandatory on malleable-iron products[1] excepting as may be required by Rule 4*a*.

(*d*) *Cast-iron Products.*—No temperature marking is mandatory on cast-iron products excepting as may be required by Rule 4*a*.[1]

(*e*) *Nonferrous Products.*—All products made of nonferrous alloys recommended by the manufacturer for steam temperatures in excess of 550 F at the marked primary steam pressure shall be marked with the numerals comprising the recommended temperature followed by the letter F to indicate degrees Fahrenheit. Otherwise no temperature marking is mandatory on nonferrous products except as may be required by Rule 4*a*.[1]

[1] This standard marking system is not intended to specify or recommend allowable operating temperatures. Purchasers should consult Codes (such as ASME Boiler Code, the Pressure Piping Code, etc.) for limiting conditions under which various materials may be used for the particular service for which the product is intended.

(*f*) *Lubricated Plug Valves.*—No temperature marking shall be required on Lubricated Plug Valves because the temperature limitation on these valves is governed not only by the metals used, but more frequently by the temperature limitation placed on any one of a number of lubricants which may be used at different times in the same valve.

Rule 5. Valve Trim Identification.—See also Section I-5*b*2.

Valve Trim Symbols

Aluminum	AL
Alloy cast iron (for example, Ni-resist)	CIA
Bronze	B
Carbon steel	CS
Cast iron	CI
Cobalt-chromium-tungsten alloy (hard surface)	HF
Copper-nickel alloy (for example, inverted monel)	CU NI
Integral seats	INT
Malleable iron	MI
Nickel-copper alloy (for example, monel)	NI CU
Soft metal (for example, lead, babbitt, copper, etc.)	SM
Steel 13 chrome (for example, stainless steel)	CR 13
Steel 18 chrome	CR 18
Steel 28 chrome	CR 28
Steel 18–8 (18 Cr–8 Ni)	18 8
Steel 8–18 (8 Cr–18 Ni)	8 18
Surface-hardened steel (for example, nitralloy)	SH

The above listed symbols may be supplemented by manufacturers' individual code or trade designations except that confusion with the standard symbols shall be avoided. Other chrome-steel or chrome-nickel-steel alloys may be designated by using symbols representing the normal analysis.

(*a*) *Gate, Globe, Angle, and Cross Type Valves.*—When trim markings are specified in this standard for gate, globe, angle, and cross type valves, they shall consist of three symbols. The first shall indicate the material of the stem, the second shall indicate the material of the disc face, and the third shall indicate the material of the seat face. Symbols shall either be preceded by the words "STEM," "DISC," "SEAT," or used alone but must appear in the order given.

Examples.—(1) Steel gate valve, 13 % chrome-steel stem, cobalt-chromium-tungsten alloy disc face, 13 % chrome-steel seat face.

```
STEM CR 13                                  CR 13
DISC HF       or CR 13  HF  CR 13,    or HF
SEAT CR 13                                  CR 13
```

(2) "All-iron" globe valve, carbon-steel stem, cast-iron disc face, integral seat face.

```
STEM CS                           CS
DISC CI     or CS  CI  INT,    or CI
SEATS INT                         INT
```

(*b*) *Check Type Valves.*—When trim markings are specified in this standard for check and other types of valves having no stem, they shall consist of two symbols. The first shall indicate material of the disc face, and the second shall

indicate material of seat face. Symbols shall either be preceded by the words "DISC," "SEAT," or used alone but must appear in the order given.

[Example omitted.]

(*c*) Valve trim marking shall not be required for bronze or brass valves. Nonferrous valves other than bronze or brass shall have trim markings in accordance with Rules 5*a* and 5*b*.

(*d*) Valve trim markings shall not be required for cast-iron and malleable-iron valves unless trimmed with other than bronze.

(*e*) Valve trim markings shall not be required for Bar-stock Valves unless trimmed with different material than used in the bodies, in which case Rule 5*a* shall apply.

(*f*) Valve trim marking shall not be required for Lubricated Plug Valves unless the plugs are of different material than used in the bodies, in which case a single trim symbol shall be used to indicate the material of the plugs.

Rule 6. Thread Identification.—Fitings, flanges, and valve bodies when threaded with other than American Standard Pipe Thread, the API Line Pipe Thread or Hose Thread shall be marked to indicate the type of thread. The marking shall be cast or stamped on the products or marked on plates or tags securely fastened to the products. This rule does not apply to products of the lighter weights ordinarily used for steam pressures up to 300 pounds.

Marking for tubing, upset tubing, casing, and drill pipe threads shall include the following: (1) nominal pipe, tubing, or casing size, (2) outside diameter or upset diameter of pipe, tubing or casing, (3) name of thread, and (4) number of threads per inch.

Examples:	3 —$3\frac{1}{2}$	API	TBG	$11\frac{1}{2}$
	4 —$4\frac{1}{2}$	API	TBG	10
	$3\frac{1}{2}$ —$3\frac{1}{2}$	API	DP	8
	$6\frac{5}{8}$ —7	DBX	CSG	10
	$6\frac{5}{8}$ —$6\frac{5}{8}$	API	CSG	10
	$4\frac{3}{4}$ —5	API	UP CSG	10
	4 —$4\frac{3}{4}$	API	UP TBG	10

NOTE.—TBG stands for Tubing. DP stands for Drill Pipe. CSG stands for Casing. UP stands for Upset. API stands for American Petroleum Institute. DBX stands for Diamond B Casing with 10 threads.

Rule 7. Size Designation. (*a*) *General.*—Except for drilling valves as specified in Rule 7*b*, when size marking is required in this standard, it shall consist of numerals comprising the nominal pipe size in inches. Size marking may be omitted from reducing flanges and also from fittings and valves having more than one size of connecting ends.

(*b*) *Steel Drilling-thru Valves.*—(1) On screwed-end valves the nominal valve size is not required because covered by thread identification marking in Rule 6. (2) Flanged-end valves having American Standard or API Standard flanges shall be marked with numerals to indicate the standard flanges with which they are equipped.

Example: 8–900 designating 8 in. size of the 900-lb Steel Flange Standard.

CHAPTER V

HEAT INSULATION

By J. H. Walker[1]

The importance of insulation as a means of conserving heat is emphasized by recognizing the magnitude of losses from bare heated surfaces as compared with the relatively small losses from such surfaces when properly insulated. This is illustrated in Fig. 1 in which heat losses per degree temperature difference from bare surfaces are shown by the upper curve and from insulated surfaces by the lower curve. The area between the two curves represents the saving by insulation.

In addition to the saving of heat or fuel, insulation performs other valuable functions. Notable among these are the minimizing of undesirable temperature drop in superheated steam lines, hot-air ducts, etc., the maintaining of desired temperatures in equipment, the prevention of leaky flange joints, and the ensuring of more comfortable working conditions in the vicinity of heated surfaces. Keeping heat away from where it is not wanted is often quite as important as keeping it where it is useful.

Heat Losses from Bare Surfaces.—The rates of heat losses from bare surfaces at temperature differences up to 1000 F are shown in Fig. 2. These are average values for still-air conditions and, although there is some variation for different pipe sizes and for different absolute temperatures of surroundings, these variations are small as compared with those caused by comparatively low air velocities; therefore, these average values are sufficiently accurate for engineering purposes.

Heat losses expressed only in Btu are not usually as significant as when expressed in the equivalent actual costs in dollars. In Fig. 3 the equivalent losses in dollars per square foot per year, 8,760 hr, have been shown for various values of heat per 1,000,000

[1] Honorary member, National District Heating Association; member ASHVE, ASME: Superintendent of Central Heating, The Detroit Edison Company.

Btu. (1,000,000 Btu is approximately equivalent to 1,000 lb of steam.)

The value of heat per 1,000,000 Btu is either known or may be computed readily for a given fuel of known cost and a given efficiency; thus, this procedure renders Fig. 3 applicable to a wide

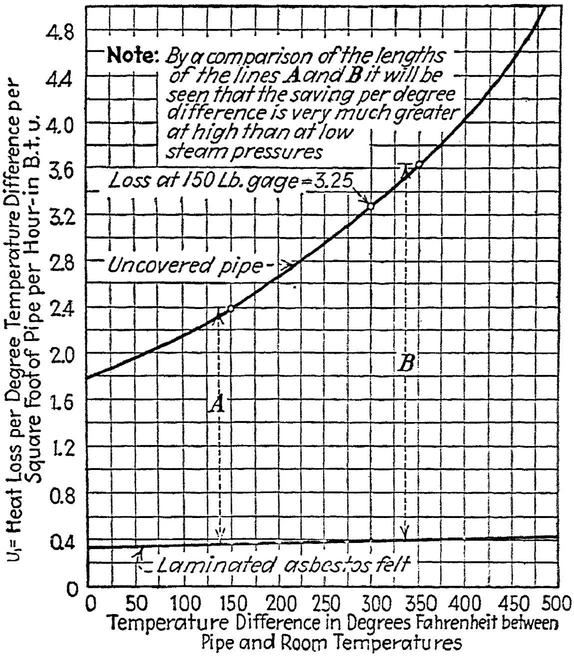


FIG. 1.—Chart showing saving in heat loss between insulated and bare pipe surfaces. (*Univ. Ill. Eng. Exp. Sta. Circ.* 7.)

range of fuels and conditions. Taking as a specific example a square foot of surface on an equipment operating 7,200 hr per year at a temperature of 300 F above that of the surrounding air, the loss per year, at a value of heat of 30 cents per 1,000,000 Btu, is \$2.60 × 7,200/8,760 = \$2.14. Suitable insulation saving upward of 90 per cent of this heat loss may be applied at a cost considerably less than one year's saving. This illustrates the desirability of insulating such surfaces as boiler heads, flanges,

and fittings which are frequently left uninsulated, even though adjacent piping is provided with effective insulation.

Effect of Air Velocity on Losses from Bare Surfaces.—The rate of heat loss from a surface maintained at constant temperature is greatly increased by air circulation over the surface. This is

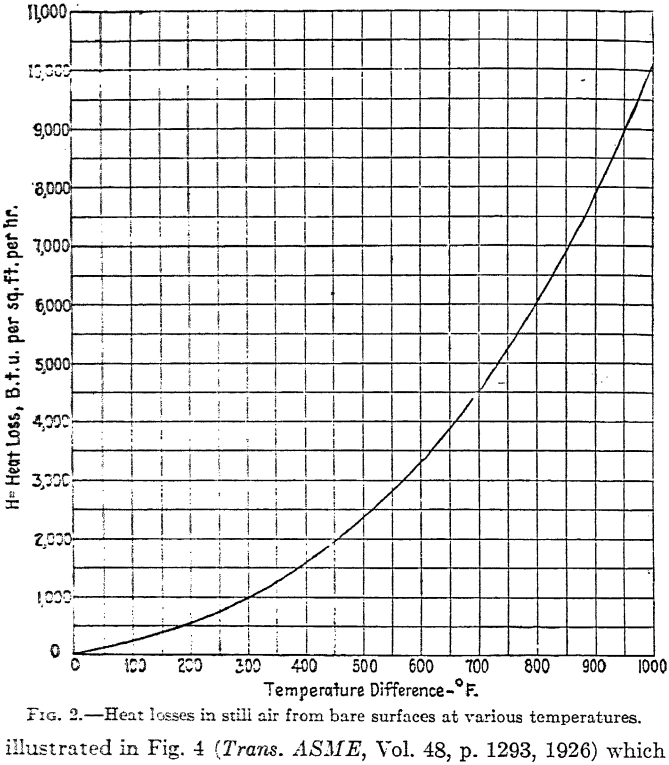


FIG. 2.—Heat losses in still air from bare surfaces at various temperatures.

illustrated in Fig. 4 (*Trans. ASME*, Vol. 48, p. 1293, 1926) which is based on Langmuir's equations (*Trans. Am. Electrochem. Soc.*, Vol. 23, 1913).

In the case of well-insulated surfaces the increases in losses caused by air circulation are very small compared with increases shown above for bare surfaces. The maximum increase in heat loss due to air velocity ranges from about 30 per cent in the case of 1-in. thick insulation to about 10 per cent in the case of 3-in.

thick insulations, provided that the insulation is thoroughly sealed so that air can flow only over the surface (see also page 707).

TABLE I*a*.—RADIATING SURFACE PER LINEAR FOOT OF STEEL PIPE

Nominal pipe size, inches	External surface, square feet	Nominal pipe size, inches	External surface, square feet	Nominal pipe size, inches	External surface, square feet
½	0.22	2	0.622	5	1.456
¾	0.275	2½	0.753	6	1.734
1	0.344	3	0.917	8	2.257
1¼	0.435	3½	1.047	10	2.817
1½	0.498	4	1.178	12	3.338

Radiating Surface of Pipes.—In order to determine heat losses per linear foot of pipe from known losses per square foot, it is

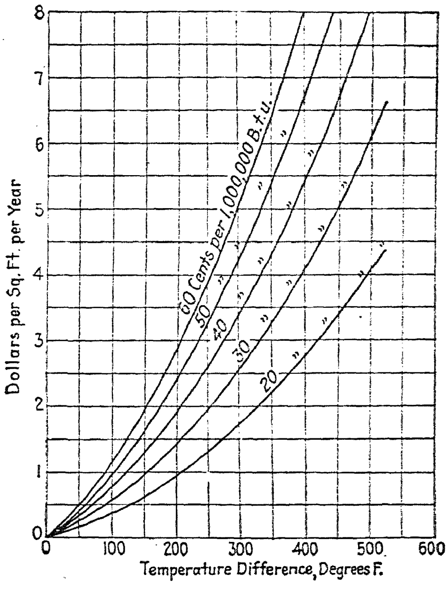


FIG. 3.—Cost of heat losses per year at various values of heat per 1,000,000 Btu.

necessary to know the number of square feet area per linear foot of pipe. Tables I*a* and I*b* give these areas for various standard pipe and tubing sizes.

TABLE Ib.—RADIATING SURFACE PER LINEAR FOOT OF COPPER TUBING

(Outside diameter ⅛ in. greater than nominal size)

Tubing size, inches	External surface, square feet	Tubing size, inches	External surface, square feet	Tubing size, inches	External surface, square feet
½	0.164	2	0.556	5	1.342
¾	0.229	2½	0.687	6	1.604
1	0.295	3	0.818	8	2.128
1¼	0.360	3½	0.949		
1½	0.426	4	1.080		

Heat Losses from Bare Fittings.—Very often, even where pipes are thoroughly insulated, flanges and fittings are left bare because of the belief that the losses from these parts are not large. However, the fact that a pair of 10-in. standard flanges having an area of 3.43 sq ft would lose at 100-lb steam pressure an amount of heat equivalent to more than 1 ton of coal per year shows the necessity for insulating such surfaces. Table II shows the areas of both 125-lb and 250-lb (or 150-lb and 300-lb steel) flanged fittings including the accompanying flanges bolted to the fitting.

TABLE II.—AREAS OF FLANGED FITTINGS,[1] SQUARE FEET

Nominal pipe size, inches	Flanged joint		90-deg ell		Long radius ell		Tee		Cross	
Pressure	125	250	125	250	125	250	125	250	125	250
1	0.320	0.438	0.795	1.015	0.892	1.083	1.235	1.575	1.622	2.07
1¼	0.383	[illegible]	0.957	1.098	1.084	1.340	1.481	1.925	1.943	2.53
1½	0.477	0.727	1.174	1.332	1.337	1.874	1.815	2.68	2.38	3.54
2	0.672	0.846	1.65	2.01	1.84	2.16	2.54	3.09	3.32	4.06
2½	0.847	1.107	2.09	2.57	2.32	2.76	3.21	4.05	4.19	5.17
3	0.945	1.484	2.38	3.49	2.68	3.74	3.66	5.33	4.77	6.95
3½	1.122	1.644	2.98	3.96	3.28	4.28	4.48	6.04	5.83	7.89
4	1.344	1.914	3.55	4.64	3.96	4.99	5.41	7.07	7.03	9.24
5	1.622	2.18	4.44	5.47	5.00	6.02	6.81	8.52	8.82	10.97
6	1.82	2.78	5.13	6.99	5.99	7.76	7.84	10.64	10.08	13.75
8	2.47	3.77	6.98	9.76	8.56	11.09	10.55	14.74	13.44	18.97
10	3.43	5.20	10.18	13.58	12.35	15.60	15.41	20.41	19.58	26.26
12	4.41	6.71	13.08	17.73	16.35	18.76	19.67	26.65	24.87	34.11
14 O.D.	5.39	8.30	16.38	22.31	20.17	25.70	24.81	33.63	31.48	43.15
16 O.D.	6.59	10.05	20.17	27.18	25.41	31.73	30.32	40.94	38.34	52.35

[1] Includes areas of adjoining companion flanges bolted on each outlet. Areas for 150- and 300-lb steel fittings are same as 125- and 250-lb cast iron, respectively.

Heat Transfer through Insulation.[1]—The rate of heat transfer through insulation is dependent upon the internal resistance offered by the insulation, the surface resistance, and the temperature of the hot surface and of the surrounding air.

The internal resistance to heat flow is dependent upon the thickness x and the conductivity k. In the case of flat surfaces

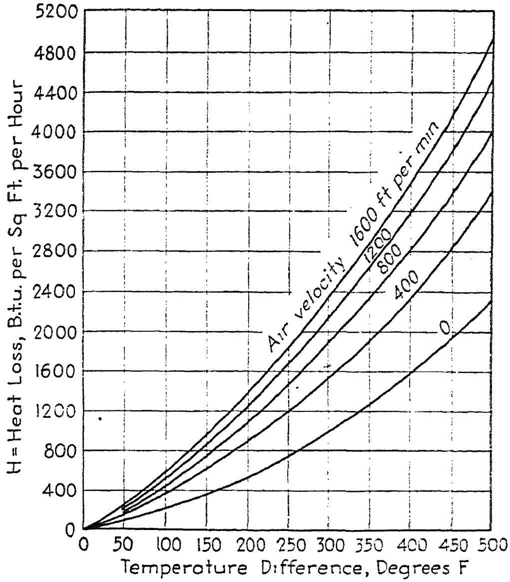


FIG. 4.—Effect of air velocity on rates of heat transfer from surfaces at various temperatures.

it varies directly as the thickness and inversely as the conductivity and is equal to x/k. Therefore, heat transfer through a material having flat surfaces is represented by the equation in which U is the over-all rate of heat transfer coefficient in Btu per square foot

[1] As an additional guide in heat transfer calculations, reference may be made to "Suggested Procedure for Calculating Heat Losses through Furnace Walls," Manual of ASTM Standards on Refractory Materials, prepared by ASTM Committee C-8, June, 1943.

per hour, per degree difference in temperature, and f is the rate of heat transfer from outer surface to air:

$$U = \frac{1}{\frac{x}{k} + \frac{1}{f}} \qquad (1)$$

The total heat transmitted is $H = UA(t_1 - t_2)$ in which

H = total heat transmitted, Btu per hr.

A = area of surface, sq ft.

t_1 = temperature of hot surface, deg F.

t_2 = temperature of surrounding air, deg F.

Conductivity.—Thermal conductivity is defined as rate of heat transfer in one direction (perpendicular to an area) per unit area, per unit temperature differential per unit thickness, per unit time (Btu per square foot, per degree temperature difference between surfaces per 1 in. thickness, per hour).

Conductivity is a specific property of a material. It is not dependent on the area, thickness, or shape of the material. It is a rate, not a quantity. The total quantity of heat transmitted is dependent upon the area, shape, and length of path (thickness of material), but conductivity is not. Conductivity is dependent upon temperature, but this is also true of other specific properties of material, density, for example. Standard test methods for determining conductivity are specified in ASTM Specification C177, Test for Thermal Conductivity of Materials by Guarded Hot Plate.

In Table III, page 701, are shown conductivities in Btu per square foot, per degree temperature difference between surfaces per 1 in. thickness, per hour for various types of insulation. In this table conductivities are shown as functions of the mean temperatures or the mean between the inner and outer surface temperatures of the insulation. This method of expressing conductivities permits their use in the calculation of heat transfer through materials whether used singly or in combination with other materials.

Surface Resistance.—The term $1/f$ in equation (1) represents the surface resistance. When heat flows through a solid material and then out into air (or any other fluid) a resistance to heat flow is encountered at the surface separating the solid from the air. Less heat will flow from the surface, therefore, than if no resistance were offered at this point.

TABLE III.—THERMAL CONDUCTIVITIES, DENSITIES, AND RECOMMENDED USE-TEMPERATURE LIMITS FOR VARIOUS TYPES OF HEAT-INSULATING MATERIALS

(Coefficient k, Btu per sq ft, per deg F, per 1-in. thickness, per hr)

Type of material (For descriptions, see pages 710-714)	Mean temperature of insulation, F									Approximate density, lb per cu ft	Use-temperature limit, F
	100 F	200 F	300 F	400 F	500 F	600 F	700 F	800 F	900 F		
1. High-temperature pipe covering and blocks (diatomaceous earth and asbestos)	0.624	0.659	0.694	0.729	0.764	0.799	0.834	0.869	.....	24 to 26	1600 to 1900
2. Laminated asbestos pipe covering and sheets (approximately 30 to 40 laminations per inch)	0.360	0.415	0.470	0.525	0.585	.....	.....	.....	.....	30 to 40	500
3. Laminated indented asbestos pipe covering and sheets (approximately 20 laminations per inch)	0.366	0.450	0.534	0.618	0.702	.....	.....	.....	.....	17 to 21	500
4. 85 per cent magnesia pipe covering and blocks	0.398	0.437	0.476	0.516	0.555	0.599	.....	.....	.....	15 to 17	600
5. Corrugated asbestos pipe covering and sheets (4 plies per 1 in. thickness)	0.496	0.621	0.746	0.871	.....	.....	.....	.....	.....	11 to 15	400
6. Corrugated asbestos pipe covering and sheets (8 plies per 1 in. thickness)	0.506	0.601	0.696	0.791	.....	.....	.....	.....	.....	21	400
7. Mineral wool (rock wool, slag wool, glass wool):											
(a) Loose	0.261[1]	0.353	0.445	0.537	0.629	0.721	0.813	0.905	.....	6 to 10	1200
(b) Blankets	0.300	0.368	0.436	0.504	0.575	0.640	0.708	0.776	.....	8 to 15	800
(c) Block	0.347	0.393	0.439	0.485	0.531	0.577	0.623	0.669	.....	15 to 23	1200 to 1800
8. Amosite asbestos pipe covering	0.402	0.450	0.500	0.550	0.596	0.652	0.690	0.738	.....	22	1200
	0.350	0.405	0.460	0.515	0.570	0.625	0.680	.....	.....	13	700
9. Expanded vermiculite (mica) pipe covering and blocks	0.880	0.930	0.980	1.030	1.080	1.130	1.180	1.230	.....	20	1200
10. Aluminum foil (3 crumpled layers per inch)	0.358	0.423	0.488	0.553	0.619	0.684	0.749	0.815	.....	0.5	1200
11. Cork pipe covering and blocks	0.30 to 0.39	.....	.....	.....	.....	.....	.....	.....	.....	7 to 14	200
12. Hair-felt sheet	0.300	.....	.....	.....	.....	.....	.....	.....	.....	6 to 11	150
13. Tar-lined wool-felt pipe covering	0.300	.....	.....	.....	.....	.....	.....	.....	.....	18	150
14. Insidline blocks	.....	.....	.....	0.630	0.720	0.800	0.880	0.965	1.050		

NOTES.—When a range of values is given for the Approximate Density or Use-temperature Limit, it indicates that these properties vary among the products of different manufacturers. The thermal conductivity values given were selected as a result of a comprehensive survey of published and unpublished data from numerous sources. The majority of the values given are based upon tests by R. H. Heilman, Mellon Institute (private communication).

[1] May vary from 0.240 to 0.350 at 100 F, depending upon the quality and density of the product.

In the case of good conductors of heat, surface resistance is the greater part of the total resistance to heat flow. In connection with efficient insulating materials, however, surface resistance is small compared with the internal resistances of the materials.

Numerically, surface resistance is the reciprocal of the rate of heat transmission from surface to air. That is, if the rate of heat transmission from surface to air is 2.0 Btu per square foot per degree temperature difference per hour, the surface resistance is 0.5. A higher rate of heat transmission from surface indicates a lower surface resistance and vice versa.

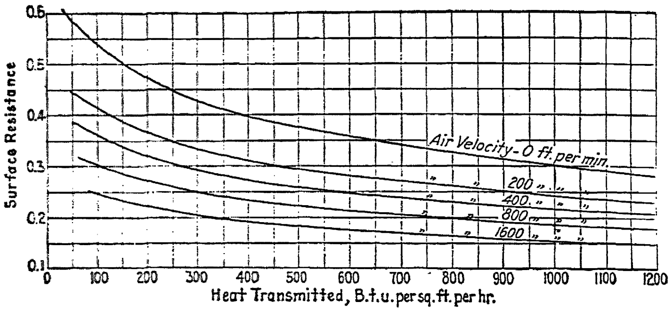


FIG. 5.—Surface resistance $1/f$ at various air velocities.

Surface resistances are very materially reduced by air velocity, and Fig. 5 shows values of surface resistances sufficiently accurate for use in insulation calculations where the surface resistance is usually less than 25 per cent and frequently less than 10 per cent of the total resistance.

Effect of Thickness of Insulation on Rate of Heat Transfer.—The manner in which the rate of heat transfer varies as the thickness of the insulation is increased is shown in Fig. 6 which is based on the conditions of equation (1). It will be noted that the curve appears relatively flat beyond 2 in. thickness. This is by no means an indication, however, that the additional savings by thicker insulations will not be large where temperatures and value of heat demand the use of such greater thickness. The curves show losses per degree temperature difference and the total losses are determined by multiplying the values from the curve by the temperature difference. For example, the saving by the addi-

tional inch of thickness from 3 to 4 in. looks small as compared with that from 1 to 2 in., but the greater thicknesses are used at the higher temperatures; therefore, actually at 500 F temperature difference, the saving, by increasing the thickness from 3 to 4 in., is much greater than that which would result from increasing it from 1 to 2 in. where the temperature difference was 100 F.

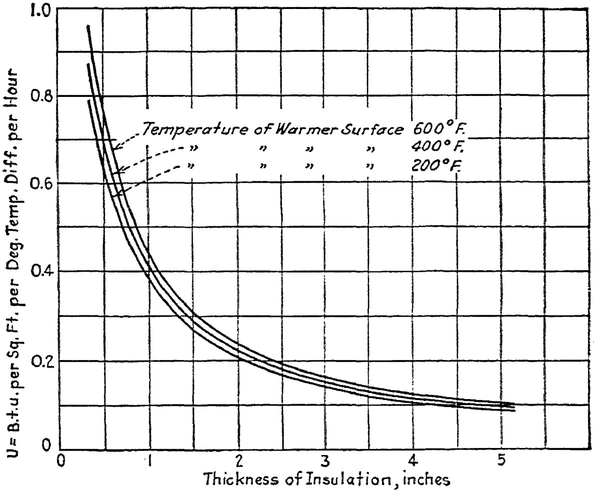


Fig. 6.—Effect of thickness on heat transfer U. Flat surfaces.

The heat transfer coefficient through a combination of any number of materials on a flat surface may be determined through the use of the following equation, in which

$$U = \frac{1}{\frac{1}{f_1} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \cdots + \frac{1}{f_2}}. \tag{2}$$

x_1, x_2, x_3, etc., are the thicknesses and k_1, k_2, k_3, etc., are the conductivities of the respective materials, f_1 is the rate of heat transfer to the warmer surface, and f_2 is the rate of heat transfer from the cooler surface.

The inside-surface resistance, $1/f_1$, is not used when the temperature of the warmer surface is known. Also, its magnitude

is often negligible where effective insulation is placed directly against a heated surface, the temperature of which is known. It is included in these general equations, however, in order that it be not neglected in cases where it should be taken into account.

Example.—Suppose it is required to calculate the heat transmission through a composite wall consisting of 4 in. of a material (brick, for example) with a conductivity of 5.0, and 2 in. of material (insulation) with a conductivity of 0.5, and that the other conditions are: inside-surface temperature, 200 F; outside-air temperature, 70 F; and rate of heat transfer from surface to air, 2.0 Btu per sq ft per degree temperature difference per hour.

Substituting in equation (2) gives

$$U = \frac{1}{\frac{4}{5} + \frac{2}{0.5} + \frac{1}{2}} = \frac{1}{5.3} = 0.189.$$

Heat transmission $= H = AU(t_1 - t_2)$.

For 1 sq ft, $H = 0.189(200 - 70) = 24.5$ Btu.

The percentages of the total insulating value contributed by the various items are as follows:

4-in. material (brick):
0.8/5.3 = 15.1 per cent of total
2-in. material (insulation):
4.0/5.3 = 75.5 per cent of total
Surface resistance:
0.5/5.3 = 9.4 per cent of total.

It is apparent, therefore, that the 2-in. material of low conductivity contributed five times as much insulating value as did the 4-in. material with relatively high conductivity and the surface resistance is to be credited with less than 10 per cent of the total insulating value. This illustrates the relative importance of the various items.

Cylindrical Surfaces.—Except on flat surfaces, the internal resistance of a material does not vary directly as the thickness. In the case of cylindrical surfaces, increasing the thickness supplies additional resistance through which the heat must flow, but at the same time increases the area of the path through which the heat may flow. This is illustrated in Fig. 7 which shows the areas of paths for flat and cylindrical surfaces. It is clearly apparent from this diagram that the heat transfer per unit of area of inner surface will be greater for insulation on a curved than on a flat surface, and that the smaller the radius of curvature, the greater will be the rate of heat transfer per unit of inner-surface area.

The heat loss per hour per degree difference in temperature per square foot of *outer surface* of the insulation on a pipe or other cylindrical surface is given by the equation in which r_1 is the external radius of the pipe or cylinder, r_2 is the radius of the outer surface of the insulation, and the other terms are as defined under equation (1).

$$U_2 = \frac{1}{\dfrac{r_2 \log_e \dfrac{r_2}{r_1}}{k} + \dfrac{1}{f}}. \tag{3}$$

The loss per square foot of *pipe surface* per degree difference in temperature is given by the equation in which U_1 and U_2 represent

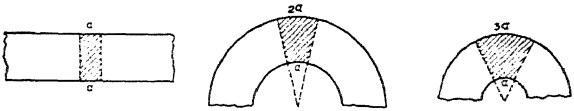


FIG. 7.—Areas of paths for heat transfer through insulations on flat and cylindrical surfaces.

the rates of heat transfer per hour per degree difference in temperature per square foot of pipe surface and outer surface of insulation, respectively.

$$U_1 = \frac{r_2}{r_1} \times U_2. \tag{3a}$$

The loss per linear foot of pipe per degree is found by multiplying the rate of heat transfer per hour per square foot of pipe surface U_1 by the square feet of external surface per foot of pipe given in Table I, pages 697–698.

The effect of pipe size on the rates of heat transfer through insulation per square foot of pipe surface under the conditions covered by equations (3) and (3a) is illustrated in Fig. 8. It will be noted that the rate of heat transfer through 2-in.-thick insulation on ½-in. pipe is more than twice as great as that through the same thickness of insulation on 12-in. pipe.

The heat transfer coefficient through combinations of two or more insulations on a pipe or other cylindrical surface may be

calculated from equations (4) and (4*a*)

$$U_s = \frac{1}{\dfrac{r_s}{r_1} \times \dfrac{1}{f_1} + \dfrac{r_s \log_e \dfrac{r_2}{r_1}}{k_1} + \dfrac{r_s \log_e \dfrac{r_3}{r_2}}{k_2} + \dfrac{r_s \log_e \dfrac{r_4}{r_3}}{k_3} + \cdots + \dfrac{1}{f_2}}. \tag{4}$$

$$U_1 = \frac{r_s}{r_1} \times U_s, \tag{4a}$$

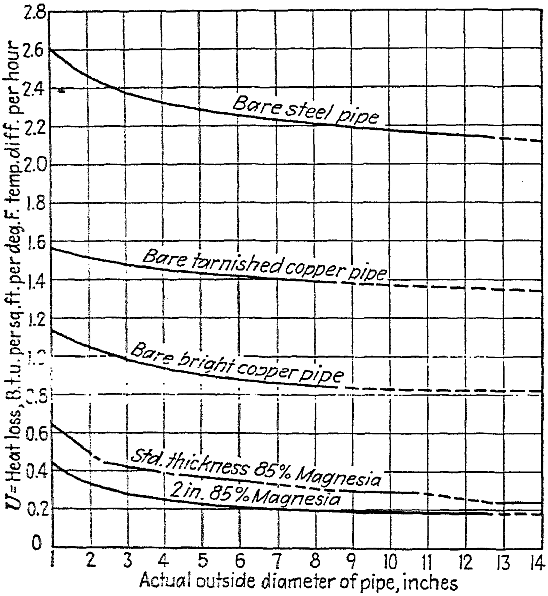


FIG. 8.—Variation[1] with pipe size of rate of heat loss from bare-steel and copper pipe compared with various thicknesses of 85 per cent magnesia insulation. Pipe temperature 210 F, surrounding air at 70 F. Dash lines indicate interpolated or extrapolated data.

in which r_s is the radius of the outer surface of insulation and U_s is rate of heat transfer per hour per degree difference in temperature per square foot of this surface. Other terms have the same significance as in previous equations.

[1] Heat loss based on data taken from Chap. 43, "ASHVE Guide," 1943.

Effect of Air Velocity on Heat Losses from Insulated Surfaces.— In the case of well-insulated surfaces the increases in heat losses due to air velocity are very small as compared to the increases previously shown for bare surfaces. This is due to the fact that air flowing over the surface of the insulation can increase only the rate of heat transfer from surface to air and cannot change the internal resistance to heat flow inherent in the insulation itself. The effect of the air circulation, therefore, is to cool the surface of the insulation to a temperature lower than it would have under still-air conditions, thereby increasing the temperature drop through the insulation.

In the case of surfaces located out of doors, the combined effect of wind and rain may bring the surface temperature of the insulation practically down to the air temperature, yet even in this extreme case the increase in heat loss through the insulation is not so great as might be expected. This is illustrated by the following example, based on a flat surface insulated with 2-in. thick material, having a conductivity of 0.5 Btu per sq ft per degree temperature difference per 1 in. thickness per hour and a rate of heat transfer from its surface to air under still-air conditions of 1.8 Btu per sq ft per degree temperature difference per hour.

$$\text{Internal resistance of insulation} = \frac{2}{0.5} = 4.0$$

$$\text{Surface resistance} = \frac{1}{1.8} = 0.556$$

$$\text{Total resistance} = 4.556$$

Rate of heat transfer $= \dfrac{1}{4.556} = 0.22$ Btu per sq ft per degree temperature difference per hour.

If the surface resistance were completely eliminated, owing to the cooling action of wind and rain, the internal resistance of 4.0 would still remain, and the rate of heat transfer would be $1/4.0 = 0.25$ Btu per sq ft per degree temperature difference per hour. Therefore, the increase in loss due to wind and rain, eliminating surface resistance, would be

$$\frac{0.25 - 0.22}{0.22} = 13.6 \text{ per cent.}$$

In like manner, it may be shown that the increase for 1 in. thickness of the same material under the same conditions is 27.9 per cent, and in the case of 3 in. thickness, 9.2 per cent. It is, therefore, apparent that the thicker or the more efficient an insulation is, the less its rate of heat transfer will be affected by air circulation.

Figure 9 shows graphically the relative increases in rates of heat transfer due to air circulation in the case of a bare surface

maintained at 400 F and the same surface insulated with 1 and 2 in. thickness of an insulation with a conductivity of 0.48 Btu per sq ft, per degree temperature difference per 1 in. thickness per hour. In these curves, the effect of air velocity on rate of heat transfer from surface to air is based on Langmuir's equations.

All of the above discussion as to effect of air circulation on losses through insulation is based on flow of air over the surface of the insulation, and applies to cases where the insulation is tightly

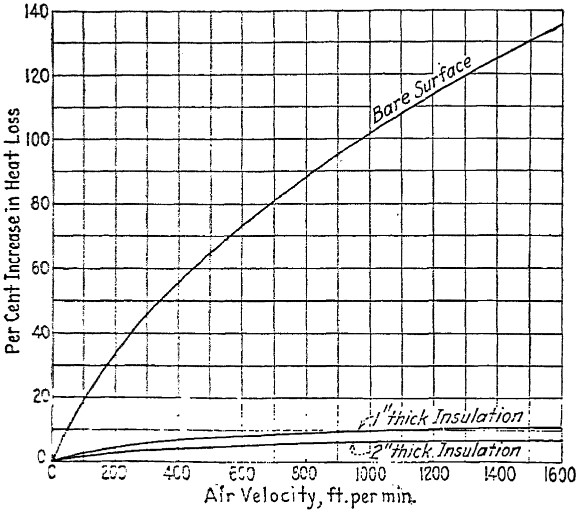


FIG. 9.—Increase in heat losses due to air circulation.

sealed. If the condition of the insulation is such that the air may circulate through cracks and crevices in the insulation, the increases may be far greater than those given above. It is essential, therefore, that all insulation be sealed as tightly as possible; and this is most particularly true of insulation located out of doors. On the latter, effective weatherproofing should be provided which should be so designed and applied as to seal the insulation against infiltration of air, as well as protecting it from the weather.

Cold Pipes.—Insulation on cold surfaces also should be sealed thoroughly against the penetration of moist air. The temperature within the insulation frequently will be, and in the case of insulated

cold water, brine, and ammonia piping usually will be, below the dew point of the surrounding air. If air leaks into the insulation, therefore, the air will be chilled below its dew point, and moisture will be deposited in the insulation. Aside from damage which may result from the consequent drippage of water on equipment, goods, etc., such condensation is undesirable since the wetted pipe surface promotes increased heat transfer from the cooling fluid, and the cold effect of the latter is reduced through absorption of the latent heat liberated by the condensing moisture, thus reducing the useful cooling effect of the refrigerant. Also, water soaking will seriously impair the effectiveness of the insulation, not only because of the increased heat transfer through the insulation, but also because of actual mechanical damage to the material itself. Where the surfaces insulated are below 32 F, moisture that enters the insulation freezes, and frequently the expansive forces due to the formation of frost are sufficient to disintegrate the insulation. In connection with insulation on such cold surfaces, therefore, the provision and maintenance of a moisture-proof seal are two of the more essential requirements.

The thickness of insulation to prevent pipe sweating or condensation of moisture on the outside of the covering on cold pipes may be determined from the chart of Fig. 10. Moisture will be deposited on a surface whenever its temperature falls to that of the dew point of the surrounding air. The maximum permissible temperature drop is indicated on Fig. 10 at the point where the guide line passes through the horizontal scale at the left center of the chart.

Inside Insulation.—Although it is customary to apply insulation to the external surface of piping, an insulation has been developed whereby part or all of it is placed on the inside of the pipe and held in place by a metal liner. Since this insulation reduces pipe wall temperatures, ordinary carbon steel pipe can handle fluids having temperatures up to 1500 F which occur in some new industrial processes, thus eliminating the need for expensive alloy pipe of special thicknesses. Expansion joints with inside insulation also are available for use in such lines. This type of insulation has been widely used in refinery cracking plants. (For further data, see Type 14 Table III, and Baldwin-Hill[1] pamphlet on Insidline insulation.)

[1] Baldwin-Hill Co., 501 Klagg Ave., Trenton, N.J.

Heat-insulating Materials.—The composition, method of manufacture, and form in which heat-insulation materials are used in pipe covering are described in the following paragraphs:

High-temperature Type.—The usual high-temperature insulation is a molded form of calcined diatomaceous silica and asbestos fiber with bonding clay and cementing materials. Some varieties contain a small percentage of carbonate of magnesia. Its weight

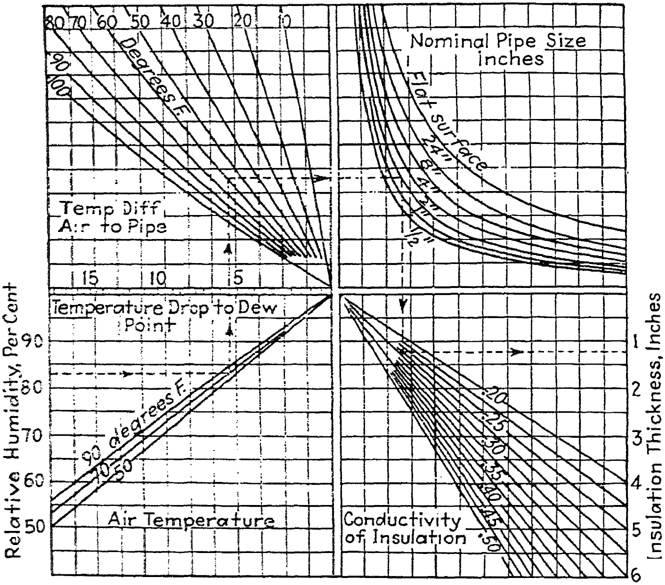


FIG. 10.—Thickness of pipe insulation to prevent sweating. Solve problems by drawing lines as indicated by dotted line, entering chart at lower left-hand scale. (*Reproduced by permission from p. 39, ASHVE Guide*, 1938.)

is approximately 23 lb per cu ft. This material, which is recommended as a first-layer insulation on steam lines 600 to 1200 F, is furnished in split sections 3 ft long, 1 to 3 in. thick, for pipe sizes ½ to 12 in., inclusive, and in flat block or segmental form for larger sizes. An alternate type of high-temperature pipe covering is described under Amosite asbestos.

Laminated Asbestos.—This is a fabricated insulation made up of 20 to 40 layers per inch of felted asbestos paper, sometimes

containing small particles of spongy cellular material, the layers being cemented together at intervals with binding strips and silicate of soda. The weight of covering having 40 laminations per inch is approximately 30 lb per cu ft. Although some manufacturers recommend it for lines up to 700 F, others limit its use to 400 to 500 F. This type may be used interchangeably with 85 per cent magnesia covering as a second layer over high-temperature insulation. The 40 layers per inch covering costs approximately 50 per cent more than the 85 per cent magnesia, but it is less expensive to apply and withstands rough handling better. It is furnished in split sections 3 ft long, 1 to 3 in. thick in ½-in. increments, for pipe sizes ½ to 30 in., inclusive, with canvas jacket and brass lacquered bands.

85 *Per Cent Magnesia.*—This is a molded insulation consisting of approximately 85 per cent by weight of basic carbonate of magnesia and 10 to 15 per cent of asbestos fiber with additions of clay or other cementing materials and weighing approximately 17 lb per cu ft. This insulation is recommended for temperatures up to 500 to 600 F and as a second layer over high-temperature insulation. It is furnished in split sections 3 ft long, all thicknesses for pipe sizes ½ to 8 in., inclusive, and single layer for 10-in. pipe size with canvas jacket and brass-lacquered bands. Double-layer insulation for 10-in. pipe size and thicknesses up to 2 in. for 12-in. pipe size may be obtained in sectional form. Insulation for larger

TABLE IV.—STANDARD THICKNESS 85 PER CENT MAGNESIA COVERING

Nominal pipe size, inches	Standard thickness, inches	Nominal pipe size, inches	Standard thickness, inches
½	⅞	3½	1 1/32
¾	⅞	4	1⅛
1	⅞	5	1⅛
1¼	⅞	6	1⅛
1½	⅞	8	1¼
2	1 1/32	10	1¼
2½	1 1/32	12 and larger	1½
3	1 1/32		

The term "standard thickness," as applied to 85 per cent magnesia covering, refers to the above thickness.

"Double standard thickness" of 85 per cent magnesia insulation is just two times standard thickness.

It should be noted that these thicknesses are referred to for preformed glass wool, vermiculite, and Amosite asbestos pipe covering as well as for 85 per cent magnesia covering but do not apply to such products as laminated asbestos, corrugated asbestos, cork, or high-temperature pipe covering.

pipe is furnished in segmental or flat-block form. For standard thicknesses of 85 per cent magnesia covering, see Table IV.

Corrugated Asbestos.—A fabricated cellular insulation consisting of alternate layers of corrugated and plain asbestos paper cemented together with silicate of soda (water glass). Various constructions are offered in four, six, and eight plies to the inch. Some varieties are intermediate between the corrugated- and laminated-asbestos types. The weight of the four ply per inch construction is approximately 12 lb per cu ft. It is recommended for service in the temperature range 100 to 300 F. For some varieties the asbestos paper is waterproofed to reduce absorption of moisture and consequent shrinkage when heated. Corrugated-asbestos insulation is available in thicknesses of ½, ¾, and 1 in. It is furnished in split cylindrical sections 3 ft long for pipe ½ to 30 in., inclusive. Sections are furnished either canvas jacketed or asbestos-paper jacketed with brass-lacquered steel bands.

Mineral Wool.—This insulation is made by blowing steam through fused clayey limestone or furnace slag to fiberize it. Although it is most extensively used in the form of bats or loose fill as a house-insulating material and for insulation inside the metal jackets of house-heating boilers, furnaces, and storage water heaters, it is also used as pipe insulation. For low-temperature insulation, a molded form of mineral wool bonded with an asphaltic compound is available, whereas for high-temperature service a flexible pipe covering composed of loose wool or bats secured to metal lath or wire netting and finished with an outer casing is sometimes used. The weight of the flexible pipe covering is about the same as an equivalent thickness of 85 per cent magnesia. Molded blocks, composed of loose or nodulated mineral-wool fibers, asbestos, clay, and chemical binders, are used for large-sized pipes, bends, valves, flanges, fittings, and on irregular contours where sectional pipe covering is not applicable. Loose fill, bats, and flexible pipe covering are recommended by the manufacturers for service at temperatures up to 1000 or 1200 F, depending upon the quality of the wool, whereas molded blocks are recommended for temperatures up to 1600 or 1800 F.

Glass Wool.—This insulation is made by blowing steam through streams of molten glass to fiberize it. The composition of the glass batch and the fiber diameters determine the type of service and use-temperature limit for which various types of glass wool are recommended by the manufacturer. It is claimed to be

unaffected by water and impervious to other destructive agencies. Loose fill, bats, and flexible blankets, usually installed at 4 to 8 lb per cu ft density, are used similarly to mineral wool products and in addition molded sectional pipe covering and blocks composed of glass wool and chemical binders are available for temperatures up to about 600 F. Turbine pads composed of glass cloth enclosing a loose glass wool filler are used frequently.

Expanded Mica.—One of the new insulating materials used for pipe covering is made by heating vermiculite (the mineralogical name for one form of mica) to a temperature sufficient to drive out its water of crystallization. The expanded mica which results is combined with asbestos fiber, bonding clay, and cement to form a molded insulation. It also is used in several varieties of insulating cements.

Hair Felt.—Hair felt made from cattle hair is furnished in felted sheets or as sectional covering having alternate layers of tar paper and felt. It usually is applied in two layers so that joints may be staggered. Because of its organic nature and combustibility, hair felt is not suitable as a permanent insulation on really hot surfaces, although it is used with success on domestic hot-water storage tanks at temperatures of 150 to 200 F.

Wool Felt.—This covering is made by felting rags and other fibrous materials into sections having a thickness of ¾ to 2 in. A protective liner of waterproof material is used when insulating cold lines. An inner liner of asbestos paper is built into the covering where it is intended for insulation of hot surfaces. Wool felt should not be used above about 250 F.

Cork.—Cork pipe insulation is made from granulated cork pressed into molds and baked at a moderate temperature. Baking liberates the resin in the cork which cements the granules together, no other binding agent being required. The molded insulation is coated with a waterproofing material to seal against air and water. Cork covering for pipe is furnished in half sections. Joints are sealed with waterproof cement.

Asbestos Blankets.—Asbestos blankets are made of two layers of asbestos cloth woven from long-fiber asbestos, between which mineral wool or loose-fill white Chrysotile asbestos or brown Amosite asbestos filling is quilted. The quilting is done with wire-inserted asbestos cord or wire on about 6-in. centers. Edges of the blankets are beveled for a lap joint. Monel-metal hooks are fastened to the edges of the blankets and lacing is ordinarily done

with monel-metal or stainless steel wire. Asbestos blankets are recommended for service at various limiting temperatures up to about 900 F, depending upon the quality of the asbestos cloth and the kind of filler used. These blankets are used extensively for insulation of steam turbines and may be used for other uneven contours as found in valves, fittings, pipe bends, etc.

Amosite Asbestos.—Long-fiber brown Amosite asbestos insulation is available in two grades of sectional pipe covering. The high-temperature (1200 F) material, which has a density of about 20 lb per cu ft, may be used as an alternate for high-temperature diatomaceous silica and asbestos insulation; the low-temperature (750 F) material, which has a density of about 12 lb per cu ft, may be used as an alternate for laminated asbestos, 85 per cent magnesia, and glass-wool pipe covering. It is furnished in split sections 3 ft long, 1 to 4 in. thick for pipe sizes ½ to 40 in. inclusive.

Committee C16 of the ASTM had in process of formulation a number of specifications for insulating materials which have been issued in the form of emergency specifications. These probably will be converted to the regular specifications after the war. The emergency specifications issued are as follows:

ES-14. Blanket Thermal Insulation for Building Purposes.
ES-15. Blanket Thermal Insulation for Industrial Purposes.
ES-16. Blanket Thermal Insulation for Refrigeration Purposes.
ES-17. Preformed Pipe Covering Thermal Insulation.
ES-18. Preformed Block (Thermal Insulation).
ES-19. Structural Insulating Board (Thermal Insulation).

Insulating Cements.—Insulating cements may be classed as: high-temperature cements; 85 percent magnesia cements; mineral-wool insulating cements; vermiculite insulating cements; semi-refractory insulating cements; high-, medium-, and low-grade asbestos insulating cements, and hard-finish cements. The high-temperature and 85 per cent magnesia cements have the same ingredients and are subject to the same temperature limitations as the molded insulations of the corresponding types.

Mineral-wool insulating cement is composed of mineral wool (in nodulated form in some makes), asbestos fiber, clay, and chemical binders. Vermiculite insulating cement consists of expanded mica, asbestos fiber, clay, and, in some makes, a chemical binder. Semi-refractory cements are composed of asbestos fiber and refractory-

clay binder. High- and medium-grade insulating cements consist of long-fiber asbestos and clay binders. Low-grade insulating cement is made from short mill-fiber asbestos. Hard-finish cements are composed of long-fiber asbestos, hard-setting clay binder, and, in some makes, Portland cement.

The following ASTM specifications on thermal insulating cements have been issued:

C193. 85 Per Cent Magnesia Thermal Insulating Cement.
C194. Long Fiber Asbestos Thermal Insulating Cement.
C195. Mineral Wool Thermal Insulating Cement.
C196. Expanded or Exfoliated Vermiculate Thermal Insulating Cement.
C197. Diatomaceous Silica Thermal Insulating Cement.

Selection of Insulation.—Typical selections of insulation for power-plant piping are given in Table V. There are many other combinations that might be used, but these materials have been found satisfactory. Similar selections of insulation for oil-refinery piping are given in Table VI. Alternate types are indicated in this table for each temperature range. Table VII gives typical selections for the various services encountered in underground steam distribution piping. The insulation of low-pressure steam and hot-water piping is given in Table VIII. Typical selections for insulation of low-temperature refrigerating and air-conditioning piping are given in Table IX. Where an analysis of economic thickness is desired, the chart of Fig. 11 is of great convenience.

In order to use the chart, start at the lower left-hand corner and proceed to the right to a point representing the given number of hours of operation per year; then proceed vertically to the line representing the given value of heat; thence horizontally, to the right, to the line representing the given temperature difference; thence vertically to the line representing the conductivity of the given material; thence horizontally, to the left, to the line representing the given discount on that material; thence vertically to the curve representing the required percentage return on the investment; thence horizontally, to the left, to the curve representing the given pipe size: thence vertically to the scale at the top of the sheet where the economical thickness may be read off directly. The cost of applying insulation varies from 10 to 50 per cent of the list price of the insulation depending on local labor

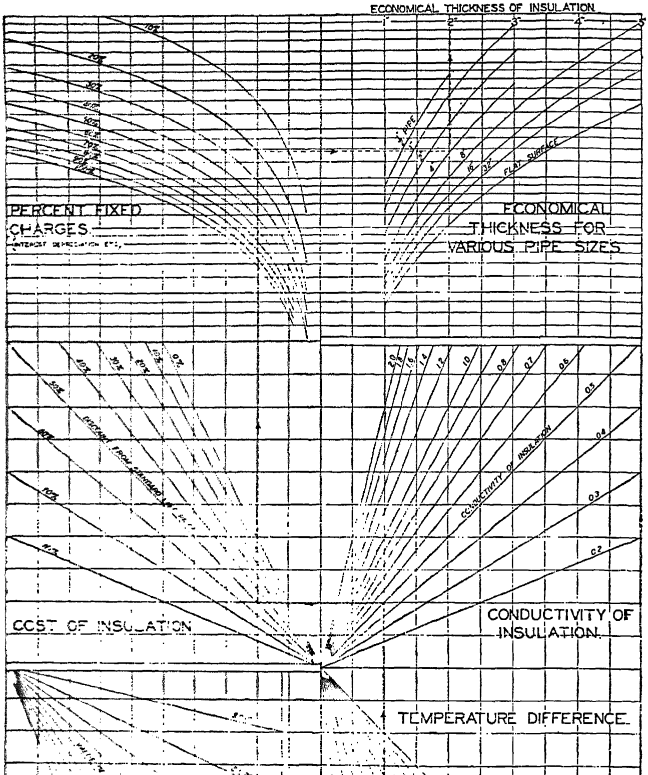


FIG. 11.—Chart for determining economical thicknesses of insulation.

conditions, type of construction, and the type of insulation used. These items should be taken into account in the cost of the insulation by reducing the discount from list price by the proper percentage, in most cases between 10 and 50 per cent. The dotted line on the chart illustrates its use in solving a typical example.

TABLE V.—TYPICAL SELECTION OF INSULATION FOR POWER-PLANT PIPING

Thickness of insulation, inches

Nominal pipe size, in.	100 to 300 F		300 to 500 F		500 to 700 F		700 to 1000 F		
	Pipe	Valves and fittings	Pipe	Valves and fittings	Pipe	Valves and fittings	Pipe: Inner layer	Pipe: Outer layer	Valves and fittings
½	Standard thickness[1]	1	2	2	2	2	2	None	2
¾	Standard thickness	1	2	2	2	2	2	None	2
1	Standard thickness	1	2	2	2	2	2	None	2
1¼	Standard thickness	1	2	2	2	2	2	None	2
1½	Standard thickness	1	2	2	2	2	2	None	2
2	Standard thickness	1	2	2	2	2	1½	1½	3[2]
2½	Standard thickness	1	2	2	2	2	1½	1½	3[2]
3	Standard thickness	1	2	2	2	2	1½	1½	3[2]
4	Standard thickness	1	2	2	2	2	1½	1½	3[2]
6	Standard thickness	1	2	2	3	3[2]	2	2	4[2]
8	Standard thickness	1	2	2	3	3[2]	2	2	4[2]
10	Standard thickness	1	2	2	3	3[2]	2	2	4[2]
12	Standard thickness	1	2	2	3	3[2]	2	2	4[2]
14	Standard thickness	1	2	2	3	3[2]	2	2	4[2]
16	Standard thickness	1	2	2	3	3[2]	2	2	4[2]
18	Standard thickness	1	2	2	3	3[2]	2	2	4[2]
20	Standard thickness	1	2	2	3	3[2]	2	2	4[2]

[1] See Table IV for standard thicknesses.
[2] Two layers, each one-half total thickness.

NOTES.—All pipe insulation of preformed sectional type. Single layer or outer layer to be of 85 per cent magnesia or Amosite asbestos. Inner layer of molded high-temperature insulation or high-temperature Amosite asbestos. Valves and fittings to be covered with high-temperature flat-block insulation. All voids to be filled with mineral wool insulating cement, block surface to be smoothed with ¼-in. finishing cement.

For drips, by-passes, and instrument lines 1¼ in. and smaller, use standard thickness high-temperature insulation.

On superheated steam lines 10 in. and larger, provide an expansion joint every 9 ft of length of covering filled, when pipe line is at operating temperature, with curly glass wool.

The above thicknesses are based on a coal cost of approximately $4 per ton for 13,000 Btu per lb of coal.

TABLE VI.—TYPICAL SELECTION OF INSULATION FOR OIL-REFINERY PIPING[1]

Nominal pipe size, inches	Thickness of insulation, inches: 250 to 350 F, Molded 85 per cent magnesia	250 to 350 F, Laminated asbestos felt	385 to 500 F, Molded 85 per cent magnesia	385 to 500 F, Laminated asbestos felt	600 to 700 F, combination, Inner layer, high-temperature[3]	600 to 700 F, combination, Outer layer, 85 per cent magnesia	600 to 700 F, Laminated asbestos felt	800 to 1000 F, combination, Inner layer, high-temperature[3]	800 to 1000 F, combination, Outer layer, Molded 85 per cent magnesia	800 to 1000 F, combination, Outer layer, Laminated asbestos felt
½	Standard thickness[2]	1	Standard thickness	1	2	None	2	2	None	None
¾	Standard thickness	1	Standard thickness	1	2	None	2	2	None	None
1	Standard thickness	1	Standard thickness	1	2	None	2	2	None	None
1¼	Standard thickness	1	Standard thickness	1	2	None	2	2	None	None
1½	Standard thickness	1	Standard thickness	1	2	None	2	2	None	None
2	Standard thickness	1	1½	1½	1¼	1⅛	2½	1 9/16	1⅛	1
2½	Standard thickness	1	1½	1½	1 5/16	1⅛	2½	1 13/16	1⅛	1
3	Standard thickness	1	1½	1½	1 9/16	1⅛	2½	2 1/16	1¼	1
4	Standard thickness	1	1½	1½	1 9/16	1¼	2½	2 1/16	1¼	1
6	Standard thickness	1	2	2	1½	1½	3	2 1/16	1½	1½
8	Standard thickness	1	2	2	1½	1½	3	2 1/16	1½	1½
10	Standard thickness	1	2	2	1 9/16	1½	3	2⅛	1½	1½
12	Standard thickness	1½	2	2	1 9/16	1½	3	2⅛	1½	1½
14	Standard thickness	1½	2	2	1½	1½	3	2⅛	1½	1½
16	Standard thickness	1½	2	2	1½	1½	3	2	1½	1½
18	Standard thickness	1½	2	2	1½	1½	3	2	1½	1½
20	Standard thickness	1½	2	2	1½	1½	3	2	1½	1½

[1] Alternate types of insulation are indicated for each temperature range. Either a combination of high-temperature insulation and 85 per cent magnesia or laminated asbestos felt may be used from 600 to 700 F. Also, equivalent thicknesses of low-temperature Amosite asbestos covering may be used as an alternate for temperatures up to 700 F, and high-temperature Amosite [illegible]

[illegible] form for larger [illegible].

The above thicknesses are based on a fuel cost of approximately 15 cents per million Btu. Where fuel costs are appreciably different from this figure, approximately ½ in. greater or less thickness should be used.

Table VII.—Typical Selection of Insulation for Underground Steam-distribution Piping (See also Pages 1016 to 1018)

Nominal pipe size, inches	Thickness of insulation, inches					
	Surface mains and feeders[1]	Lines in tunnels and feeders	Service connection, buried construction[1]	Service connection, tunnels and tunnel shafts	Surface and tunnel return lines in basements and areaways	Drip connections in tunnels, manholes, basements, and areaways[7]
¾	....	....	....	....	....	1
1	....	....	....	....	....	1
1¼	....	....	....	....	....	1
1½	....	....	....	....	....	1
2	....	....	....	1½[2]	1[1,5]	1
2½	....	....	1	1½[2]	1[1,5]	
3	....	....	1	1½[2]	1[1,5]	
4	1	1½[2]	1	1½[2]	1[1,5]	
6	1	1½[2]	1	1½[2]	1[1,5]	
8	1	1½[2]	1	1½[2]	1[1,5]	
10	1	3[3]	1	3[3]	1[1,5]	
12	1½	3[3]	1½	3[4]	1½[1,6]	
14 O.D.	1½	3[3]	1½	3[4]	1½[1,6]	
16 O.D.	1½	3[3]	1½	3[4]	1½[1,6]	

[1] Laminated asbestos with waterproof jacket.
[2] Eighty-five per cent magnesia pipe covering.
[3] Eighty-five per cent magnesia, 1½-in. inner layer and 1½ × 3 × 36 in. flat-block outer layer asbestos cement, and 9-oz duck.
[4] Double-layer flat block, 1½ × 3 × 36 in. with asbestos cement and 9-oz duck.
[5] Standard thickness, 85 per cent magnesia, except for damp conditions where laminated asbestos is used. See Table IV for standard thicknesses.
[6] Single-layer flat block, 1½ × 3 × 36 in.
[7] Laminated asbestos with roofing cement coating.
Note.—Amosite asbestos sectional pipe covering, with or without waterproof jacket, may be used as an alternate for any of these services and in the same thicknesses as designated in this table.

Table VIII.—Typical Selection of Insulation for Low-pressure Steam and Hot-water Piping

Pipe sizes	Insulation, hot water	Steam	
		25 psi, maximum	100 psi, maximum
½ to 30 in.	Air-cell type, ¾-in.	Air-cell type, ¾ in.	85% magnesia, standard thickness

TABLE IX.—TYPICAL SELECTION OF INSULATION FOR LOW-TEMPERATURE REFRIGERATING AND AIR-CONDITIONING PIPING

Service	Temperature range, deg. F	Insulation	Number of layers	Total thickness (nominal), inches	Waterproofed
Cold water (antisweat).	50 to 75	Hair or wool felt	1	5 in. and less—¾ 6 in. and over—1	Inside and outside
Ice water............	32 to 50	Hair or wool felt	1 2	3 in. and less—¾ 4 in. and over—1½	Inside and outside
		Cork	1	Ice water thick[1]	Seams and edges
Brine and ammonia....	25 to 35	Cork Hair felt	1 2	Ice water thick[1] 2	Seams and edges Inside and outside
Brine and ammonia....	0 to 25	Cork Hair felt	1 3	Brine thickness[1] 3	Seams and edges Inside and outside
Brine and ammonia....	−25 to 0	Cork Hair felt	1 4	Heavy brine th.[1] 4	Seams and edges Inside and outside

[1].—"Ice water thickness" cork is approximately 1½ in. thick.

"Standard brine thickness" cork is approximately 2 in. thick on pipes ½ to 1 in., 2½ in. thick on pipes 1¼ to 3 in., and 3 in. thick on larger sizes.

"Heavy brine thickness" cork is approximately 3 in. thick on pipes ½ to 3 in., 3½ in. thick on pipes 3½ to 4 in., and 4 in. thick on larger sizes.

NOTE 1.—Application:

Correct application of insulation on cold piping is absolutely essential to the maintenance of lasting efficiency. The insulation must be thoroughly sealed against admission of moisture from the air. The lower the pipe temperature, the more imperative the necessity for perfect sealing, because at pipe temperatures below freezing the slightest opening at the joints will allow moisture to enter, condense, and freeze, and this formation of frost tends to damage or destroy the insulation.

NOTE 2.—Where water pipes are exposed to air temperatures below 32 F, they are usually insulated to prevent freezing. A common specification consists of three layers of standard hair felt (total nominal thickness 3 in.) protected by means of a weatherproof roofing jacket lapped and sealed at all joints. This specification is suitable where water circulation is maintained continuously or where circulation is interrupted for only brief periods. Where conditions of exposure are more severe or where periods of no flow are more prolonged, the line is usually protected by means of a small steam line alongside of the water line, and both pipes are then insulated together by wrapping with two plies of ¼-in. asbestos air-cell paper and two layers of standard hair felt finally protected with a weatherproof roofing jacket sealed at all joints.

For additional data on insulation and velocities of water to prevent freezing, see Chap. 18 on Pipe Insulation, "1944 Guide of the American Society of Heating and Ventilating Engineers," and "Handbook on Welded Steel Pipe," published by the California Corrugated Culvert Co., Berkeley, Calif. See also p. 1071.

NOTE 3.—Glass-wool sectional pipe covering may be used as an alternate for low-temperature refrigerating and air-conditioning piping. Recommended thicknesses for each type of service should be obtained from the manufacturer.

CHAPTER VI

HANGERS AND SUPPORTS

Support of piping and regulation of its motion are important items in design, otherwise the stresses and thrusts occasioned by expansive movement and dead weight may exceed safe working values for the pipe material, connected equipment, or building structure. The design of pipe hangers and anchors is closely related to the study of flexibility in pipe lines.[1] In fact, it is necessary to determine the points of fixed and sliding anchorage and the means of supporting a line before calculating stresses.

Anchors.—Providing anchors to guide or control expansive travel is of prime importance and the necessity for having them should not be overlooked. Anchors or bracing are needed to prevent the disengagement of expansion joints of the slip type, and of certain kinds of pipe joints such as bell-and-spigot, particularly where there is a change in the direction of the line. No loop header or other section of large pipe having small branches and subject to expansion should be left floating in an indeterminate fashion owing to the likelihood of having the expansive travel take place in a direction that will overload the small branch connections. Good practice dictates that any piping system subject to thermal expansion shall be anchored and guided in such a manner that the travel at any point is not left to conjecture or be apt to cause damage to branch connections or equipment. Solid anchors are used where it is desired to fix the piping with reference to motion in three-dimensional space, that is, in the longitudinal, transverse, and vertical directions, and are usually located at important junction points in the piping system (see Fig. 8). Such anchors serve to divide a piping system into isolated sections with respect to expansive movement, and each section should contain suitable provision to take up the thermal elongation occurring between the solid anchors at its two extremities. Sliding anchors are provided to prevent buckling and guide the expansion movement.

[1] See Chap. VII, p. 754.

Anchors may be connected to pipe or fittings by means of bolted clamps, welded attachments, or cast pads in the case of cast fittings. Where welded attachments are used for anchors or hangers, attention is called to the rules abstracted on page 725.

Hangers.—Supports should be designed to prevent excessive stresses in operation or too large variations in loading with change in temperature, and to guard against shock or possible resonance with imposed vibrations. The complete release of support at any one point through the failure of a spring or other part must be prevented, since so drastic a shift in load might result in failure of adjacent supports or of the line itself. For the same reason, complete disengagement of supporting members through expansive movement of the pipe line must be avoided.

Materials.—Steel is commonly used for permanent hangers, supports, and anchors and their accompanying appurtenances such as pipe and beam clamps, turnbuckles, straps, etc. The Code for Pressure Piping[1] permits the use of cast iron for roller-bearing bases, roller guides, anchor bases, brackets, and other parts of piping supports where the loading is mainly compression. Malleable-iron castings of suitable design are permitted for pipe clamps, beam clamps, hanger flanges, clips and bases, swivel rings, and similar parts of piping supports where the operating temperature of the pipe line does not exceed 450 F. The Piping Code requires that materials must be capable of meeting the physical and chemical requirements and tests of the specifications listed.

Permanent supports are required to be of noncombustible materials where used in tunnels and buildings of fireproof construction, except that wood and wire may be used for rigging and temporary supports. For nonfireproof buildings and locations outside of buildings, wooden structures may be used to support piping except that, for conveying fluids at temperatures in excess of 230 F, piping should be spaced or insulated from such supports to prevent dangerous overheating.

The Piping Code requires that all parts of supporting equipment, except springs, shall be designed with a factor of safety of five based on the tensile strength of the material. Springs should be designed as recommended on page 729. Where the supporting equipment is intended to function at temperatures of 650 F or higher, the calculated stress should not exceed the allowable

[1] Copies may be obtained from the American Society for Mechanical Engineers, 29 W. 39th St., New York 18, N.Y.

stresses permitted by the Code for corresponding materials at the desired temperature as given on page 43 for power and district heating, pages 1212, 1213 for gas and air, pages 1155 to 1161 for oil, and page 1237 for refrigeration piping. For high-temperature service, the clamps of Type I hangers (see Fig. 4) may be extended beyond the insulation so that the hanger rod and bolt are outside of the high-temperature range to enable the use of a higher allowable stress commensurate with the lower temperature.

MATERIAL SPECIFICATIONS FOR HANGERS AND SUPPORTS
ASA B31.1

Material	Specification	
Steel, hot rolled bar	ASTM	A 107
Steel, welded and seamless pipe for ordinary uses	ASTM	A 120
Steel, structural	ASTM	A 7
Steel, structural (plates)	ASTM	A 78
Steel, structural (rivets)	ASTM	A 141
Wrought iron (refined bars)	ASTM	A 41
Wrought iron (plates)	ASTM	A 42
Wrought iron (extra-refined bars)	ASTM	A 84
Cast iron (ordinary gray-iron castings)	ASTM	A 48
Cast iron (higher grade gray-iron castings such as used for valves, flanges, and pipe fittings)	ASTM	A 126
Malleable-iron castings	ASTM	A 47
Malleable-iron castings (cupola process)	ASTM	A 197
Brass (rods and bars for structural use)	ASTM	B 21
Bronze (manganese bronze castings)	ASTM	B 54
Chains	ASTM	A 56
Springs, helical (for use on spring hangers)	ASTM	A 125
Springs, elliptical (for use on spring hangers)	ASTM	A 62

Loading.—Safety requires that supporting equipment, except springs, should be designed to include weight of the pipe, valves, and fittings, weight of the fluid transported or that used for testing,

whichever is the heavier, and the weight of insulation, if used. Weight calculations for springs are based on normal operating conditions to secure springs that are properly proportioned for normal operation rather than the seldom encountered test conditions. Where the possibility of lines becoming full of water or other fluid is remote, such as for large gas or air piping, exhaust steam or safety-valve relief piping, the Piping Code permits design of supporting equipment on the basis of normal operating conditions. For exterior piping, wind pressure and snow and ice loads should be considered in the calculations of strength and structural features of supports.[1] Earthquake shock should be provided for in design in localities where such shocks are known to occur. In California, provision of a side thrust equal to one-fifth of the dead load due to gravity is considered to be ample for buildings that are public gathering places, and one-tenth of the gravity load for other buildings. Design stresses and safe loadings for pipe when used as structural columns are given in Chap. VII on pages 759 to 763.

Corrosion Resistance.—Where conditions exist that would cause excessive corrosion of ordinary steel or wrought iron, nonferrous metals or corrosion-resisting steel alloys should be used for supports. Cast or malleable iron may be used to avoid excessive corrosion subject to the limitations mentioned on page 722. Under conditions causing atmospheric rusting or slight corrosion, corrosion-resistant materials need not be used, but a protective coating such as hot-dipped galvanizing, a weather-resistant paint, or other suitable protection shall be applied after fabrication to all parts where required. Even under conditions causing no material corrosion, it is recommended that a suitable paint or other durable protective coating be applied to all parts after installation. Under any conditions, exposed screw threads should be greased or painted immediately after installation unless corrosion-resistant materials are used.

With steam lines and other lines carrying hot fluids, or with lines carrying refrigerants or extremely cold fluids, the supports must be carefully worked out and more elaborate provisions made. For simple longitudinal expansion, the supports must permit the pipe to elongate or contract without undue restriction. When the line is supported from below, travel can be accommodated with rollers

[1] See "Design of Modern Industrial Piping Systems, Support of Pipe Lines," by F. L. Snyder and T. E. Bridge, *Heating, Piping and Air Conditioning*, April, 1936, pp. 189–195.

or slides; and if hung from above, the hangers may be flexible or articulated.

The Code for Pressure Piping has designated the following dimensional limitations for supporting equipment used for piping within its jurisdiction:

(a) *Straps.*—Straps shall be limited to the minimum dimensions of $\frac{1}{8}$ in. thickness by 1 in. width for use in locations protected from the weather. The minimum strap thickness for locations exposed to the weather shall be $\frac{1}{4}$ in. As exceptions to this, where straps are to be used for supporting pipe of 1 in. nominal size or smaller, the minimum strap dimensions shall be $\frac{1}{16}$ in. thickness by $\frac{3}{4}$ in. width in locations protected from the weather, and $\frac{1}{8}$ in. thickness in locations exposed to the weather.

(b) *Hanger Rods.*—Hanger rods shall be limited to a minimum diameter of $\frac{3}{8}$ in. for supporting pipe of 2 in. nominal size or smaller. For supporting pipe of $2\frac{1}{2}$ in. nominal size and larger, rods used in the fabrication of hangers which are to be located in places protected from the weather, shall be limited to a minimum diameter of $\frac{1}{2}$ in. Where rods are to be exposed to the weather or other corrosive elements, they shall be fabricated from corrosion-resistant metals or protected by a suitable protective coating. The minimum rod sizes for these corrosive conditions shall be as given above for the noncorrosive condition.

(c) *Chain.*—Chain used for hangers shall be limited to stock having a minimum diameter of $\frac{3}{16}$ in. or equivalent area for supporting pipe of 2 in. nominal size or smaller. For supporting pipe of $2\frac{1}{2}$ in. nominal size and larger, chain used in the fabrication of hangers which are to be located in places protected from the weather shall be limited to stock having a minimum diameter of $\frac{3}{8}$ in. or equivalent area. Rods for chain hangers shall be fabricated in accordance with the preceding paragraph. Where such composite chain hangers are to be exposed to the weather or other corrosive elements, castings and chain stock as well as the rods and all parts shall be fabricated, where applicable, from corrosion-resistant metals or protected with a suitable coating, according to conditions. The minimum sizes of chain stock for these corrosive conditions shall be as given above for the noncorrosive condition.

(d) *Bolted Plate Clamps.*—Bolted plate clamps, used in connection with rod or chain hangers, shall have a minimum thickness of $\frac{3}{16}$ in. for weather-protected locations and $\frac{1}{4}$ in. for places where these parts are exposed to the weather. The bolts used for these clamps shall be of the same diameter as the hanger rod, or $\frac{3}{8}$ in. diameter as minimum when clamps are used with chain.

(e) *Welded Hanger Supports.*—Lugs, plates, angle clips, etc., used as a part of a hanger assembly for the support of pipe may be welded directly to the pipe provided the design is adequate for the load. Welded joints shall be proportioned so that the stresses caused therein by the pipe loading shall not exceed the following values: shear on the section through the weld throat, 13,600 psi; tension on the section through the weld throat, 15,600 psi; and compression on the section through the weld throat, 18,000 psi. Welding shall be done by operators qualified in accordance with the rules of Chap. 4 of the Code, see abstract on page 496.

Hangers that are adjustable under load should be used for supporting pipe lines 2 in. nominal size and larger; except where it is

desired to maintain an exact grade, rigid hangers may be used. If adjustable-type hangers are used, the turnbuckle or adjusting nut should make use of its complete thread and the amount of adjustment should be visible. Suitable locking devices should be provided for such devices.

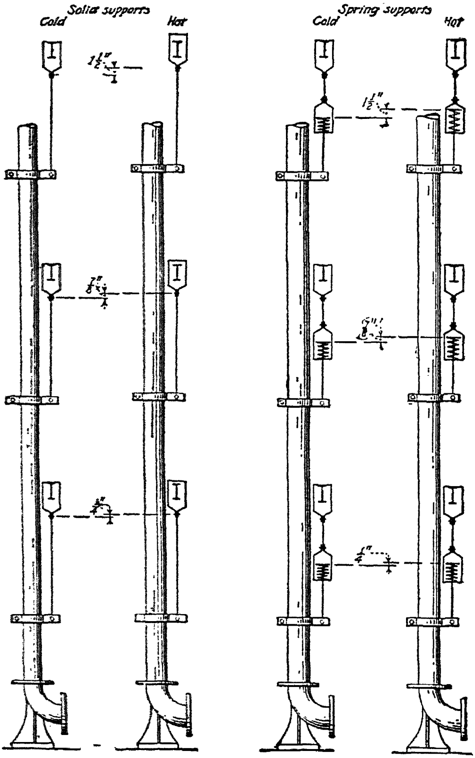


FIG. 1.—Action of vertical pipe with and without spring supports.

The case of a vertical pipe subjected to expansion is somewhat less simple. If the pipe is supported at intervals by ordinary rod hangers or other rigid supports, each of them, if properly adjusted, will carry its share of the load when the pipe is cold; but as the pipe becomes heated and lengthens it will tend to rise clear of the

upper supports and the weight will be largely or wholly transferred to the bottommost support. This may be permissible if the support is adequate, but such is not always the case, and severe strains may be imposed upon fittings or upon the flanges of equipment to which the piping is connected.

The remedy is to use hangers that are more or less elastic and will partially maintain their supporting action as the pipe elongates. This is best accomplished by the use of helical springs, properly selected, although pivoted counterweights, or counterweights hung on cables running over sheaves, sometimes are used. Figure 1 illustrates this condition.

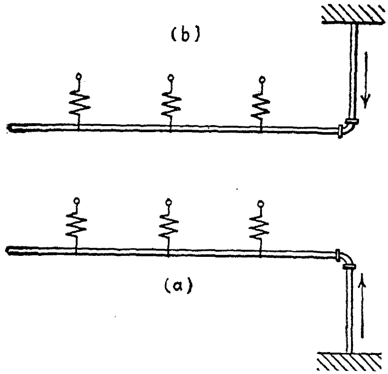


FIG. 2.—Conditions where spring supports are required.

Flexible supports are called for in certain other cases. A horizontal pipe may be subjected to vertical movement by the action of some other part of the piping system (see Fig. 2). Spring supports will permit it to move up or down without unduly disturbing the distribution of load. With rigid supports excessive stresses might be set up.

These precautions are especially necessary with piping connected to steam turbines or other apparatus which would be affected by expansion stress. Each case must be thoroughly analyzed.

The selection of springs for pipe hangers preferably should be done by a rational method rather than by guess. The principles of spring design are so simple that a spring suitable for the exact purpose usually can be selected by reference to a handbook table. A convenient table for determining helical spring sizes is given in

"Machinery's Handbook."[1] In selecting a hanger spring, one should be chosen that is not too stiff to give a reasonable deflection under load and yet will not compress under load to the point where its coils are in contact. The term "free length" is applied to the length of a helical spring when not under load, and "solid length" to the condition when all the coils are in contact under a compressive load. For hangers, a reasonable spring deflection under normal working load would be about one-half the difference between the free and the solid length. Likewise the opposing springs in sway braces should be compressed about 30 per cent when in the normal position.

A typical selection of springs for use under average conditions in spring hangers such as those shown in Figs. 5 and 6, or for the sway braces shown in Fig. 9, is given in Table I. The stiffness of spring selected for any particular hanger application should depend, of course, on the spacing of hangers and the load to be carried on each, while the difference between the free and the closed length should be chosen to suit the expansive lift.

Material for the springs shown in Table I may be ordered in accordance with ASTM Specification A125 for heat-treated steel helical springs. The springs are designed so that, at the maximum load with the spring compressed solid, the stress in the spring wire will approximate 100,000 psi. This stress does not take into account the Wahl factor, which provides for increase in stress resulting from a decrease in ratio of mean diameter of the spring to wire diameter, because (1) the working load is intended to be from 40 to 60 per cent of the maximum load which results in a spring fiber stress of only 40,000 to 60,000 psi and (2) for these springs, the Wahl factor is relatively low, not exceeding about 1.25 since the spring diameter is large with respect to the wire diameter. The stress at the maximum load or that at the working load may be corrected to include the Wahl factor by multiplying the stress by the Wahl factor, which is available from "Machinery's Handbook." In general, the springs given in Table I are suitable for hangers used for nominal pipe sizes the same as the spring size, but for pipe heavier than Schedule 80, springs should be checked to determine acceptability for the actual loading and service con-

[1] See also (a) "Design of Modern Industrial Piping Systems, Support of Pipe Lines," by F. L. Snyder and T. E. Bridge, *Heating, Piping and Air Conditioning*, February, 1936, pp. 93–99. (b) "How to Select and Install Spring Hangers for Piping," by W. F. Fischer, *ibid.*, August, 1939, pp. 491–494.

ditions. This is particularly the case for lines above 700 F where the large pipe movement occasioned by the higher temperatures probably will require springs capable of greater movement between free and solid lengths.

TABLE I.—TYPICAL SELECTION OF SPRINGS FOR SPRING HANGERS[1] AND SWAY BRACES

Spring number and nominal pipe size	Outside diameter spring, inches	Diameter wire, inches	Free length, inches	Solid length, inches	Maximum load at solid length, pounds	Number of active coils[2]	Deflection per turn, inches	Weight of spring, pounds
2	$1\frac{7}{8}$	$\frac{1}{4}$	$8\frac{1}{2}$	$4\frac{3}{16}$	370	15.3	0.283	1.20
3	$2\frac{7}{8}$	$\frac{3}{8}$	$8\frac{1}{2}$	$4\frac{3}{16}$	810	9.7	0.445	2.76
4[3]	$2\frac{7}{8}$	$\frac{7}{16}$	$8\frac{1}{2}$	$4\frac{13}{16}$	1,350	9.5	0.388	3.58
6	$3\frac{3}{8}$	$\frac{9}{16}$	$8\frac{1}{2}$	$5\frac{5}{16}$	2,490	7.9	0.402	5.87
8[4]	$3\frac{7}{8}$	$\frac{5}{8}$	$8\frac{1}{2}$	$5\frac{1}{4}$	2,890	6.9	0.470	7.45
10[4]	$4\frac{7}{8}$	$\frac{3}{4}$	$8\frac{1}{2}$	$5\frac{1}{16}$	3,980	5.2	0.654	11.0
12[4]	$5\frac{7}{8}$	$\frac{7}{8}$	$8\frac{1}{2}$	5	5,250	4.2	0.833	15.2

[1] Based on a hanger spacing of 15 to 18 ft.

[2] Total coils equals active coils plus two. Additional spring travel may be provided by extra coils when unusual expansive movement requires it.

[3] Can be used also for sway braces on 6-in. nominal pipe size.

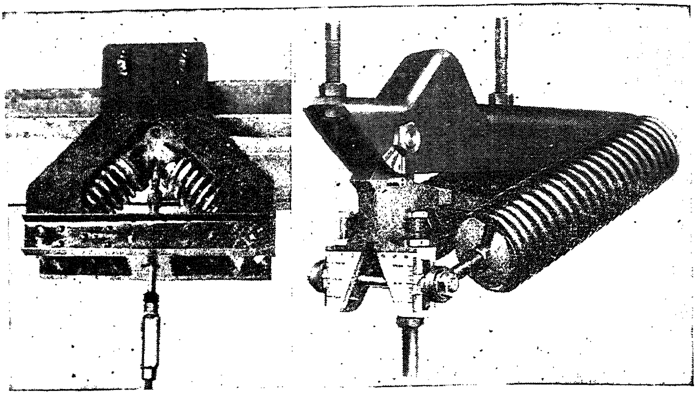
[4] Can be used also for sway braces on 50 per cent larger pipe diameters unless shock conditions are unusually severe.

The stiffness of a helical spring under load depends on the diameter of the rod from which the spring is made, the number of coils, and the diameter of the coils. Where the hanger design is such as to impose limitations on one or another of these features, the others can be selected to give the characteristics wanted in the spring.

Where the amount of vertical movement to be absorbed is large, some form of constant support should be provided. Although weighted counterbalances hung on levers or on sheaves and cables have been used in some cases, the obvious drawbacks to such devices have led to the development of hangers containing springs mounted in a toggle action tending to give an approach to constant support over a considerable range of travel.[1] Two types of toggle hangers designed to provide essentially constant support over a range of travel of 3 to 6 in. are illustrated in Fig. 3, the springs in the hanger in (*a*) being loaded in compression while those in the hanger in (*b*) are loaded in tension. Suitable limit stops are pro-

[1] See article on "The Balanced Support of High-temperature Pipe Lines," by Joseph Kaye Wood, *Engineering*, Dec. 3 and Dec. 10, 1937.

vided in the latter type so that failure of a spring is not of serious consequence. The load-*vs*.-travel characteristic of such a support is shown in the diagram in Fig. 3*c*. The load required to compress a helical spring to its solid length is a function of the size rod used and the diameter of the coil and is independent of the number of



(*a*) (*b*)

FIG. 3.—Constant support hangers utilizing toggle-action springs. Type (*a*) was invented by Paul E. Todd while employed by The Detroit Edison Company. Type (*b*) was invented by Joseph Kaye Wood and is manufactured in this country by the Grinnell Company, Inc.

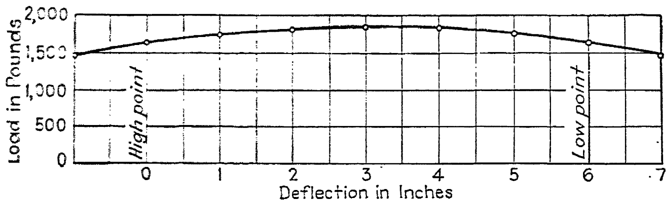


FIG. 3*c*.—Load-deflection curve for type (*a*).

coils in the spring. In other words, if the load to close solid a given spring having 5 coils is 1,000 lb, a similar spring having 10 coils will also close under a load of 1,000 lb. This is because the load supported by each coil is the same in either case. The over-all change in length of the coil will, of course, be twice as great. For example, the 10-coil spring under a load of, say, 400 lb would be the more flexible of the two because it would have twice as many coils. On

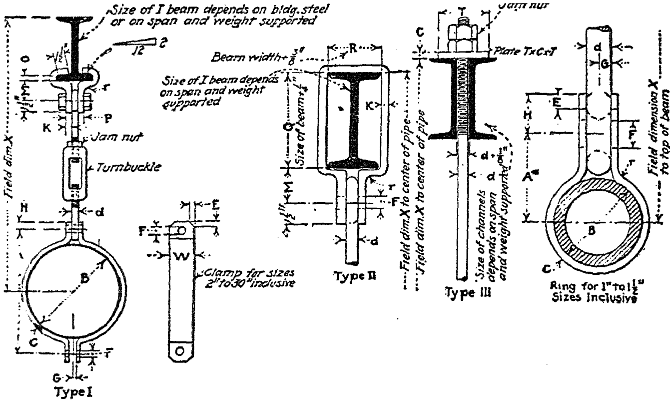


FIG. 4.—Details of rod hanger, designed for 12-ft spacing.

TABLE II.—DIMENSIONS OF PIPE HANGERS SHOWN IN FIG. 4
(All dimensions in inches)

Nominal pipe size, inches	*A*	*B*	*C*	*W*	*d*	*G*	Bolts	*r*	*E*	*F*	*H*	Developed length of ring or half clamp	*K*	*M*	Max.[2] load per hanger, pounds
1	1 3/8[1]	1 5/16	3/16	1	3/8	3/16	3/8 × 1 1/2	1/8	3/8	7/16	5/8	7 1/8[1]	1/4	1	1,325
1 1/4	1 3/4[1]	1[illegible]	[illegible]	1	3/8	3/16	3/8 × 1 1/2	1/8	3/8	7/16	5/8	8 7/8[1]	1/4	1	1,325
1 1/2	1 13/16[1]	1[illegible]	[illegible]	1	3/8	3/16	3/8 × 1 1/2	1/8	3/8	7/16	5/8	9 1/2[1]	1/4	1	1,325
2	4 1/2	2[illegible]	[illegible]	[illegible]	3/8	3/16	3/8 × 1 1/2	1/4	3/8	7/16	5/8	7 1/2	3/8	1 1/4	1,325
2 1/2	5	2[illegible]	[illegible]	[illegible]	1/2	1/4	1/2 × 2	1/4	1/2	9/16	3/4	9	3/8	1 1/4	2,355
3	5 1/2	3 1/2	1/4	2	1/2	1/4	1/2 × 2	1/4	1/2	9/16	3/4	11	3/8	1 1/4	2,355
4	6 3/4	4 1/2	3/8	2	5/8	5/16	5/8 × 2 1/2	3/8	5/8	11/16	1	12	3/8	1 1/4	3,670
6	9 1/4	6 5/8	3/8	2 1/2	3/4	3/8	3/4 × 2[illegible]	[illegible]	[illegible]	[illegible]	1[illegible]	16	1/2	1 1/2	5,305
8	11 1/4	8 5/8	3/8	3	3/4	3/8	3/4 × 2[illegible]	[illegible]	[illegible]	[illegible]	1[illegible]	18	1/2	1 3/4	5,305
10	13 1/2	10 3/4	3/8	3	7/8	7/16	7/8 × 3[illegible]	[illegible]	[illegible]	[illegible]	1[illegible]	22	1/2	1 3/4	7,215
12	15 3/4	12 3/4	3/8	3 1/2	7/8	7/16	7/8 × 3[illegible]	[illegible]	[illegible]	[illegible]	1[illegible]	27	3/4	2 1/4	7,215
14	17 1/2	14	1/2	3 1/2	1	1/2	1 × 3[illegible]	[illegible]	1[illegible]	1[illegible]	1[illegible]	31	3/4	2 1/4	9,425
16	19 1/2	16	1/2	3 1/2	1	1/2	1 × 3[illegible]	[illegible]	1[illegible]	1[illegible]	1[illegible]	36	3/4	2 1/4	9,425
18	22	18	5/8	3 1/2	1 1/8	9/16	1 1/8 × 4[illegible]	[illegible]	1[illegible]	1[illegible]	2	37	3/4	2 1/2	11,930
20	24	20	5/8	3 1/2	1[illegible]	9/16	1[illegible] × 4[illegible]	[illegible]	1[illegible]	1[illegible]	2	40	3/4	2 1/2	11,930
22	26	22	5/8	3 1/2	1[illegible]	9/16	1[illegible] × 4[illegible]	[illegible]	1[illegible]	1[illegible]	2	44	3/4	2 1/2	11,930
24	28 1/2	24	5/8	4	1[illegible]	[illegible]	1[illegible] × 4[illegible]	[illegible]	1[illegible]	1[illegible]	2[illegible]	47	3/4	2 3/4	14,725
26	30 1/2	26	5/8	4	1[illegible]	[illegible]	1[illegible] × 4[illegible]	[illegible]	1[illegible]	1[illegible]	2[illegible]	51	3/4	2 3/4	14,725
28	32 1/2	28	5/8	4	1[illegible]	[illegible]	1[illegible] × 4[illegible]	[illegible]	1[illegible]	1[illegible]	2[illegible]	54	3/4	2 3/4	14,725
30	34 1/2	30	5/8	4	1[illegible]	[illegible]	1[illegible] × 4[illegible]	[illegible]	1[illegible]	1[illegible]	2[illegible]	57	3/4	2 3/4	14,725

NOTES.—Bolts in beam clamp and pipe clamp are of the same size. All holes are to be punched if possible. Beam clamp and pipe clamp bars are of the same width. All eyebolts are to have welded or upset and drilled eyes.

[1] See Fig. 4, right-hand detail.

[2] Based on ASTM A107 hot-rolled bar-stock material having a 60,000 psi tensile strength, factor of safety of 5.

the other hand, the *change* in spring force per inch of movement will be only half as great. For this reason, it is desirable to select a spring with enough turns to give a reasonable cushioning effect, and a satisfactory small departure from constant force.

A reasonable figure for the difference between open and solid lengths is 4 in. for small helical springs, increasing to; say, 10 in. for springs on large pipes. This allowance is sufficient for hangers in horizontal runs of pipe and provides enough follow-up in the spring to care for a small amount of expansion in the case of a short vertical run of pipe. A much more liberal allowance should be made for springs used in a long vertical run subject to expansion. Springs in any case should be provided with means to prevent misalignment, buckling, eccentric loading, and with stops to prevent excessive travel. Supports of either the rigid or spring types should be capable of taking the full load resulting from expansion or contraction, from failure of springs in spring supports, or from excessive loads encountered during erection.

In the case of important hangers or spring supports, it is sometimes desirable to have the springs calibrated in a compression testing machine in which the actual load producing a given deflection can be measured. This will serve to detect any errors in the design or manufacture of the spring or weakness in the material. Methods for testing helical springs are described in ASTM Specifications A61 and A125.

Figure 4 shows a simple rod hanger, and Table II gives the dimensions of the various parts. This hanger is designed for heavy duty, as in a steam power plant. Hangers should be spaced at about 12-ft intervals for small pipe, while larger pipes may have a greater distance between supports, up to 40 to 50 ft having been used on some occasions. The thickness of the pipe as regards its ability to act as a beam (see pages 744 to 753) and the strength of the hanger should be considered in determining frequency of hanger spacing. When beam clamps of the design illustrated with Type I hangers are used for heavy loads hung from wide beams, a clamp made from heavier stock may be required to avoid buckling. For light, simple piping systems, such as ordinary heating pipes, there are available cheaper and lighter commercial designs of hangers. In Fig. 5 is shown a spring hanger for a 6-in. pipe.

Where it is necessary to provide for a large amount of horizontal movement and the headroom is not sufficient to permit the use of long rods for the required swing, the type of hanger shown in Fig.

6 is useful. It consists of a solid or spring hanger with a roller which runs on an I beam.

It sometimes is desirable to provide a hanger that will permit free movement in any direction in a horizontal plane. The multiple rolling action of the spring hanger shown in Fig. 7 is expressly designed for this purpose. In this type of hanger, it is desirable to design the supporting building steel so that the greater movement is provided by the top rollers thus permitting smaller

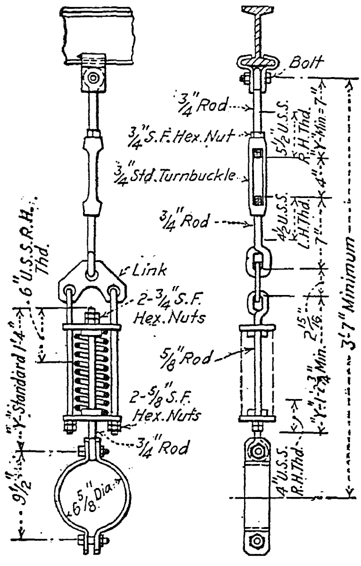


FIG. 5.—Spring hanger for 6-in. pipe.

carriages. Special hangers, such as illustrated in Figs. 6 and 7, are costly and should be avoided wherever possible. Correct methods of supporting important lines, such as the main-steam system of a large power plant, are essential, however, and a special type of hanger, expressly designed for the purpose, sometimes offers the best solution for a troublesome situation. Application of several types of hangers with anchors and sway bracing in the support of a main steam system for a power plant is shown *schematically* in Fig. 8. The number of hangers required in an actual installation depends on the size of the pipe, length of the line to be supported,

temperature of the fluid conveyed, and other factors which are not taken into consideration in this schematic representation.

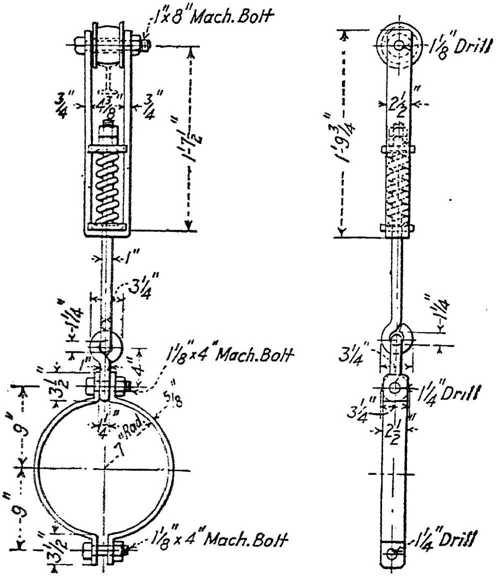


FIG. 6.—Spring hanger, with roller, for 14-in. pipe.

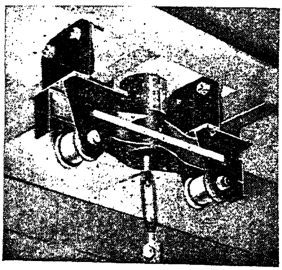


FIG. 7.—Multiple-roller-type spring hanger.

At points marked A, a constant-support type of hanger is used so that as the pipe expands vertically, the entire load is not transferred back to the anchor point as was explained in connection

with Fig. 1. In the case illustrated, the stress in the pipe already was high so that it was desirable to maintain constant support at points A through use of a toggle-type hanger. Where the stress

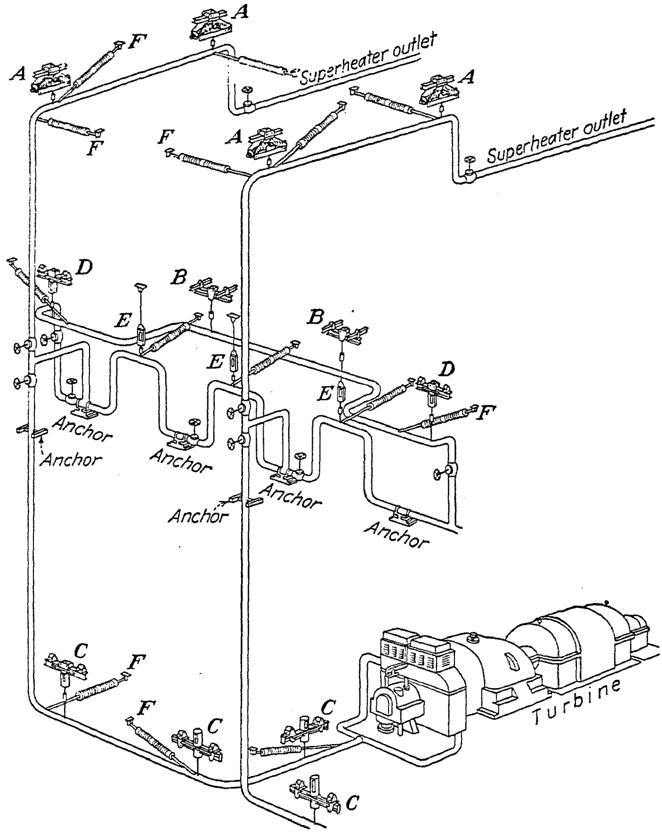


FIG. 8.—Typical application of hangers, anchors, and sway bracing for a main steam piping system.

is within reasonable limits as at points C or D, ordinary springs of ample length in proportion to the amount of expansion may be used. At points B multiple-roller solid-rod hangers are used since the small amount of upward expansion is cared for by the spring

hangers at points *D* near the ends of the loop, and it is necessary to provide only for the horizontal movement. Roller hangers are essential at points *B* since considerable bowing of the line is to be expected here in a horizontal plane as expansion takes place. Multiple-roller spring hangers rather than toggle hangers are used at points *C* since the downward expansion of the pipe increases the spring load and decreases the tension on the anchor. The additional bending moment exerted on the pipe was of no significance in this case since the stress in the pipe was low. If the pipe stress were high, it again would be desirable to use constant support hangers at points *C* to maintain as nearly as possible the original loading conditions on the line. Hangers shown at *E* are the ordinary spring type on rods since it is unnecessary to provide for much horizontal or vertical movement at these points. Several sway braces, such as indicated at points *F*, have been provided to dampen vibration and cushion shock. In any case, the individual line to be hung should be studied and hangers applied to suit the conditions.

Guides.—It is occasionally necessary to guide the expansion of a pipe or the direction of movement of some selected point in the system, such as an important fitting. These are special cases for which no specific rules can be given as to the design. In general, a cross-head type of guide is to be preferred, and the parts should be generously proportioned because the forces acting are not usually capable of computation.

Sway Braces.—In the case of pipe lines closely associated with equipment and conveying steam, feed water, compressed air, oil, or other fluids, some kind of control device often is required to cushion vibration or shock. Two types of sway braces that may be used for this purpose are shown in Fig. 9. The cushioning effect is provided by opposing springs which should be preloaded equally in the normal operating position about 30 per cent of the difference between the open and the closed lengths of the springs so that any tendency to move in the direction of either is resisted by a force that increases in proportion to the amount of travel. If the location for the sway brace has been properly selected, this amount of preloading usually permits sufficient spring travel so that the spring will not be compressed solid when the pipe is cold, thus avoiding possible overstressing of the line in the cold condition. The preloading eliminates lost motion and acts to dampen vibration or to prevent slamming back and forth under shock con-

ditions. To be most effective, such dampeners must be applied in two or three directional planes and at points where their action will not be disturbed unduly by expansive movement of the pipe line (see Fig. 8). It has been found by experience that springs used for sway-bracing any size pipe can well have about the same characteristics as the springs used in support hangers for the respective sizes as shown in Table I. Where severe water hammer is encountered it may be advisable to use the spring corresponding to the next larger pipe size.

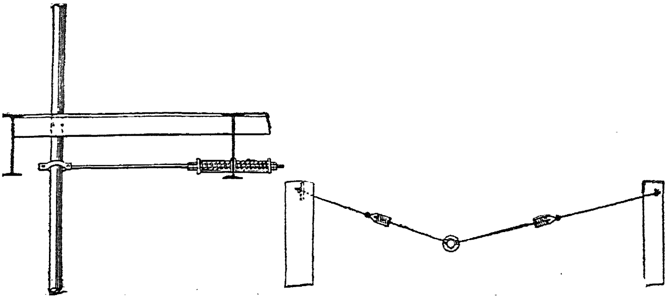


FIG. 9.—Sway braces.

Fabrication of Hangers.—On a large piping job considerable money may be saved by standardizing hanger design in the drafting department and ordering the hangers made up in advance ready for the erection crew, rather than having the field blacksmith do the job from pencil sketches made by the pipe fitters. In standardizing hanger design, it is advisable to distinguish between parts common to practically all hangers for a given size pipe, and parts that must be made to dimension for each particular hanger location. For instance, in the case of rod hangers, the clamps that bolt around the pipe and the diameter of the supporting rods can be made identical for all pipes of the same size, while the length of the rods must be ordered to suit the requirements of each individual location. Similarly, the style of clamp for attachment to the structural work in the building must suit the particular size of beam, angle, channel, etc. In making up a standard drawing for hangers, each style should be illustrated with lettered dimensions referring to a table placed on the same sheet which gives for each size pipe hanger the dimension in inches corresponding to the letter in the illustration. Dimensions such as X, the length of rod,

TABLE III.—APPROXIMATE WEIGHTS OF FL

Nominal pipe size, inches	ASA primary-service rating, psi	ASA pipe schedule number	Weight, pounds per linear foot of pipe						Weight, pounds per item of one[2]		
			Pipe	Water contents	Insulation[1]				One flange		
					100–300 F	300–500 F	500–700 F	700–1000 F	Slip-on and screwed	Lapped	Welding-neck
1¼	125 CI[3]	40	2.28	0.65	0.77	2.56	2.56	3.52	2.2 CIS		
	150 St	40	2.28	0.65	0.77	2.56	2.56	3.52	3.0 St	3.0	3.0
	250 CI	80	3.00	0.55	0.77	2.56	2.56	3.52	4.0 CIS		
	300 St	80	3.00	0.55	0.77	2.56	2.56	3.52	4.0 St	4.0	6.0
	400	80	3.00	0.55	0.77	2.56	2.56	3.52	6.0	6.0	7.0
	600	80	3.00	0.55	0.77	2.56	2.56	3.52	6.0	6.0	7.0
	900	80	3.00	0.55	0.77	2.56	2.56	3.52	10	10	10
	1,500	160	3.76	0.46	0.77	2.56	2.56	3.52	10	10	10
1½	125 CI	40	2.72	0.88	0.85	2.72	2.72	3.74	2.9 CIS	...	...
	150 St	40	2.72	0.88	0.85	2.72	2.72	3.74	3.0 St	3.0	4.0
	250 CI	80	3.64	0.76	0.85	2.72	2.72	3.74	6.0 CIS	...	...
	300 St	80	3.64	0.76	0.85	2.72	2.72	3.74	6.0 St	6.0	8.0
	400	80	3.64	0.76	0.85	2.72	2.72	3.74	7.0	7.0	10
	600	80	3.64	0.76	0.85	2.72	2.72	3.74	7.0	7.0	10
	900	80	3.64	0.76	0.85	2.72	2.72	3.74	14	14	14
	1,500	160	4.86	0.61	0.85	2.72	2.72	3.74	14	14	14
2	125 CI	40	3.66	1.45	1.23	3.04	3.04	6.38	4.8 CIS	...	...
	150 St	40	3.66	1.45	1.23	3.04	3.04	6.38	5.0 St	5.0	6.0
	250 CI	80	5.03	1.28	1.23	3.04	3.04	6.38	7.4 CIS	...	...
	300 St	80	5.03	1.28	1.23	3.04	3.04	6.38	7.0 St	7.0	9.0
	400	80	5.03	1.28	1.23	3.04	3.04	6.38	9.0	9.0	12
	600	80	5.03	1.28	1.23	3.04	3.04	6.38	9.0	9.0	12
	900	80	5.03	1.28	1.23	3.04	3.04	6.38	25	25	25
	1,500	160	7.45	0.77	1.23	3.04	3.04	6.38	25	25	25
2½	125 CI	40	5.80	2.07	1.41	3.31	3.31	7.04	6.8 CIS	...	...
	150 St	40	5.80	2.07	1.41	3.31	3.31	7.04	7.0 St	7.0	8.0
	250 CI	80	7.67	1.83	1.41	3.31	3.31	7.04	11 CIS	...	...
	300 St	80	7.67	1.83	1.41	3.31	3.31	7.04	10 St	10	12
	400	80	7.67	1.83	1.41	3.31	3.31	7 04	13	12	18
	600	80	7.67	1.83	1.41	3.31	3.31	7.04	13	12	18
	900	80	7.67	1.83	1.41	3.31	3.31	7.04	36	35	36
	1,500	160	10.0	1.53	1.41	3.31	3.31	7.04	36	35	36

ANGED, WELDED, AND SCREWED PIPE LINES

Weight, pounds per item of one[2]

90° ell		Tee (full size)		Check valve		Globe valve		Gate valve	
Flanged	Welding and screwed	Flanged	Welding and screwed	Flanged	Welding and screwed	OS and Y Flanged	Welding and screwed	OS and Y Flanged	Welding and screwed
7.0 CI	0.60 B	11 CI	1.1 B		2.5 S		3.5 S	15 CI	4.3 S
7.0 St	1.3 S	11 St	1.6 S		2.8 S		5.5 S	15 St	7.6 S
13 CI	0.80 B	22 CI	1.4 B		4.1 S		8.0 S	37 CI	7.6 S
13 St	2.10 S	22 St	2.7 S	29 St	18 S	29 St	17 S	28 St	14 S
......	0.80 B	....	1.4 B						
16	2.10 S	24	2.7 S	29	18 S	36	27 S	28	14 S
......	0.80 B	....	1.4 B						
16	2.10 S	24	2.7 S	29	18 S	36	27 S	28	14 S
......	0.80 B	....	1.4 B						
26	2.10 S	43	2.7 S	29	18 S	76	27 S	115	35 B
......	1.0 B	....	1.7 B						
26	......	43	5.0 S	......	......	76	......	115	90 B
9.3 CI	0.94 B	15 CI	1.4 B		3.4 S		4.9 S	25 CI	6.1 S
9.3 St	1.80 S	15 St	2.2 S		3.7 S		11 S	25 St	11 S
20 CI	1.1 B	29 CI	1.8 B		5.5 S		11 S	43 CI	11 S
20 St	2.9 S	29 St	3.9 S	50 St	26 S	43 St	25 S	60 St	48 B
......	1.1 B	....	1.8 B						
24	2.9 S	33	3.9 S	50	26 S	43	25 S	60	48 B
	1.1 B	....	1.8 B						
24	2.9 S	33	3.9 S	50	26 S	43	25 S	60	48 B
......	1.1 B	....	1.8 B						
39	2.9 S	57	3.9 S	......	......	114	41 S	135	120 B
......	1.7 B	....	3.0 B						
39	......	57	8.5 S	......	......	114	......	135	120 B
16 CI	1.6 B	23 CI	2.1 B	22 CI	14 CIS	30 CI	26 CI	42 CI	36 CIS
17 St	2.9 S	27 St	3.5 S	25 St	15 B	48 St	36 B	52 St	40 B
22 CI	2.1 B	33 CI	3.0 B	49 CI	39 CIS	61 CI	49 CIS	48 CI	42 CIS
24 St	4.7 S	36 St	5.8 S	62 St	38 B	80 St	69 B	68 St	55 B
......	2.1 B	....	3.0 B						
38	4.7 S	46	5.8 S	68	52 B	105	105	105	105
......	2.1 B	....	3.0 B						
38	4.7 S	46	5.8 S	68	52 B	105	90 B	96	80 B
......	2.1 B	....	3.0 B						
64	4.7 S	100	5.8 S	140	100 B	200	70 S	185	145 B
......	3.2 B	....	4.5 B						
64	......	100	15 S	140	100 B	200	160 B	185	145 B
21 CI	3.1 B	34 CI	3.9 B	27 CI	20 CIS	41 CI	31 CIS	55 CI	41 CIS
28 St	7.0 S	41 St	10 S	31 St	20 B	62 St	53 B	65 St	50 B
31 CI	4.0 B	48 CI	5.3 B	60 CI	55 CIS	84 CI	72 CIS	80 CI	73 CIS
35 St	9.0 S	42 St	14 S	88	50 B	100 St	85 B	92 St	85 B
......	4.0 B	....	5.3 B						
45	9.0 S	70	14 S	108	85 B	128	105 B	138	115 B
......	4.0 B	....	5.3 B						
45	9.0 S	70	14 S	108	85 B	128	105 B	138	115 B
......	4.0 B	....	5.3 B						
100	9.0 S	130	14 S	245	180 B	325	265 B	262	200 B
......	6.0 B	....	7.0 B						
100	......	130	24 S	245	180 B	325	265 B	262	200 B

TABLE III.—

Nominal pipe size, inches	ASA primary-service rating, psi	ASA pipe schedule number	Weight, pounds per linear foot of pipe: Pipe	Water contents	Insulation[1]: 100–300 F	300–500 F	500–700 F	700–1000 F	Weight, pounds per item of one[2]: One flange: Slip-on and screwed	Lapped	Welding-neck
3	125 CI[3]	40	7.58	3.19	1.63	3.84	3.84	7.78	8.1 CIS	...	...
	150 St	40	7.58	3.19	1.63	3.84	3.84	7.78	8.0 St	8.0	10
	250 CI	80	10.3	2.86	1.63	3.84	3.84	7.78	15 CIS	...	...
	300 St	80	10.3	2.86	1.63	3.84	3.84	7.78	13 St	13	15
	400	80	10.3	2.86	1.63	3.84	3.84	7.78	16	15	23
	600	80	10.3	2.86	1.63	3.84	3.84	7.78	16	15	23
	900	80	10.3	2.86	1.63	3.84	3.84	7.78	31	30	32
	1,500	160	14.3	2.35	1.63	3.84	3.84	7.78	48	47	48
4	125 CI	40	10.8	5.51	2.22	4.48	4.48	9.05	14 CIS	...	...
	150 St	40	10.8	5.51	2.22	4.48	4.48	9.05	13 St	13	15
	250 CI	40	10.8	5.51	2.22	4.48	4.48	9.05	24 CIS	...	...
	300 St	40	10.8	5.51	2.22	4.48	4.48	9.05	22 St	22	25
	400	40	10.8	5.51	2.22	4.48	4.48	9.05	26	25	35
	600	80	15.0	4.98	2.22	4.48	4.48	9.05	37	36	42
	900	80	15.0	4.98	2.22	4.48	4.48	9.05	53	51	51
	1,500	160	22.6	4.02	2.22	4.48	4.48	9.05	73	75	73
6	125 CI	40	19.0	12.5	3.0	6.04	10.0	17.1	19 CIS	...	...
	150 St	40	19.0	12.5	3.0	6.04	10.0	17.1	19 St	19	24
	250 CI	40	19.0	12.5	3.0	6.04	10.0	17.1	37 CIS	...	...
	300 St	40	19.0	12.5	3.0	6.04	10.0	17.1	39 St	39	42
	400	40	19.0	12.5	3.0	6.04	10.0	17.1	44	42	57
	600	80	28.6	11.3	3.0	6.04	10.0	17.1	80	78	81
	900	80	28.6	11.3	3.0	6.04	10.0	17.1	108	105	110
	1,500	160	45.3	9.16	3.0	6.04	10.0	17.1	164	170	164
8	125 CI	40	28.6	21.7	4.30	7.45	12.3	20.4	30 CIS	...	...
	150 St	40	28.6	21.7	4.30	7.45	12.3	20.4	30 St	30	39
	250 CI	40	28.6	21.7	4.30	7.45	12.3	20.4	58 CIS	...	...
	300 St	40	28.6	21.7	4.30	7.45	12.3	20.4	58 St	58	67
	400	40	28.6	21.7	4.30	7.45	12.3	20.4	67	64	89
	600	60	35.7	20.8	4.30	7.45	12.3	20.4	115	112	117
	900	80	43.4	19.8	4.30	7.45	12.3	20.4	172	188	187
	1,500	160	74.7	15.8	4.30	7.45	12.3	20.4	258	286	273

TABLE III.—

Nominal pipe size, inches	ASA primary-service rating, psi	ASA pipe schedule number, or wall thickness, in	Weight, pounds per linear foot of pipe						Weight, pounds per item of one[2]		
			Pipe	Water contents	Insulation[1]				One flange		
					100–300 F	300–500 F	500–700 F	700–1000 F	Slip-on and screwed	Lapped	Welding-neck
10	125 CI[3]	40	40.5	34.2	5.20	8.93	14.4	23.0	41 CIS	...	...
	150 St	40	40.5	34.2	5.20	8.93	14.4	23.0	43 St	43	52
	250 CI	40	40.5	34.2	5.20	8.93	14.4	23.0	82 CIS	...	...
	300 St	40	40.5	34.2	5.20	8.93	14.4	23.0	81 St	91	91
	400	40	40.5	34.2	5.20	8.93	14.4	23.0	91	112	126
	600	60	54.8	32.3	5.20	8.93	14.4	23.0	177	195	189
	900	80	64.4	31.1	5.20	8.93	14.4	23.0	245	277	268
	1,500	160	116	24.6	5.20	8.93	14.4	23.0	436	485	454
12[4]	125 CI	0.375	49.5	48.9	7.50	10.3	16.5	27.2	62 CIS	...	...
	150 St	0.375	49.5	48.9	7.50	10.3	16.5	27.2	64 St	64	80
	250 CI	0.375	49.5	48.9	7.50	10.3	16.5	27.2	117 CIS	...	...
	300 St	0.375	49.5	48.9	7.50	10.3	16.5	27.2	115 St	139	138
	400	0.375	49.5	48.9	7.50	10.3	16.5	27.2	129	152	177
	600	0.500	65.4	46.9	7.50	10.3	16.5	27.2	215	240	226
	900	80	88.6	44.0	7.50	10.3	16.5	27.2	326	371	372
	1,500	160	161	34.9	7.50	10.3	16.5	27.2	667	749	690

[1] Based on insulation thickness given in Tables IV and V, pages 711 and 717. Weight of 85
22 lb per cu ft. See Table III, page 701, for density of insulating materials.

[2] Weights of valves and fittings were compiled from manufacturer's catalogues. Because of
but the weights given herein are reasonably representative. Socket-welding fittings have approxi-
pages 611 to 615.

[3] CI = cast iron; St = steel; S = screwed; B = butt welding; CIS = cast iron screwed.

[4] For sizes larger than 12 in.: see pages 360 and 362 for pipe weights; pages 611 to 615 for

must, of course, be left indeterminate. Then, in writing up the order for a hanger it can be described, for instance, as, "1 Type II hanger for 4-in. pipe, dimension X 36 in., beam clamp to fit 6-in. I beam." The exact manner in which this is done will depend on the office routine in any particular drafting room.

Pipe clamps are frequently made in a forge shop with a steam hammer operated with either steam or compressed air. A medium-sized hammer may be used with the dies illustrated in Fig. 10. A bottom die is placed on the anvil of the steam hammer. The bar stock for a half clamp is laid on this die and an upper die is held

(Concluded)

Weight, pounds per item of one[2]

90° ell		Tee (full size)		Check valve		Globe valve		Gate valve	
Flanged	Welding and screwed	Flanged	Welding and screwed	Flanged	Welding and screwed	*OS* and *Y* Flanged	Welding and screwed	*OS* and *Y* Flanged	Welding and screwed
194 CI	82 B	275 CI	81 B	475 CI	......	526 CI	......	515 CI	
228 St	140 S	310 St	190 S	500 St	415 B	......	......	610 St	525 B
305 CI	82 B	440 CI	81 B	700CI	520 S			895 CI	
325 St	140 S	531 St	190 S	748 St	465 B	......	......	1,000 St	700 B
......	82 B	....	81 B						
423	140 S	584	190 S	900	820 B	......	......	1,425	1,350 B
......	108 B	....	101 B						
630	230 S	915	312 S	1,100	875 B	......	......	1,750	1,550 B
......	111 B	....	101 B						
859	230 S	1,344	312 S					2,400	
......	270 B	....	265 B						
295 CI	120 B	385 CI	111 B	685 CI	......	......	......	720 CI	
330 St	245 S	435 St	355 S	830 St	700 B	......	......	850 St	715 B
435 CI	120 B	615 CI	111 B	1,000 CI	740 B			1,210 CI	
504 St	245 S	783 St	355 S	1,105 St	700 B	......	......	1,536 St	1,130 B
......	120 B	....	111 B						
650	245 S	868	355 S	1,375	1,150 B	......	......	2,036	1,800 B
......	158 B	....	140 B						
800	295 S	1,120	586 S	1,600	1,375 B	......	......	2,530	2,200 B
......	158 B	....	140 B						
1,200	295 S	1,890	586 S					3,500	
......	460 B	....	489 B						

per cent magnesia assumed as 16 lb per cu ft. Weight of high-temperature insulation assumed as

differences in design, considerable variation exists between products of different manufacturers, mately the same weight as screwed fittings. For [illegible] of 125-lb cast-iron fittings see

Weights of fittings and valves without symbols are for steel.
125-lb CI fittings and manufacturers' catalogues.

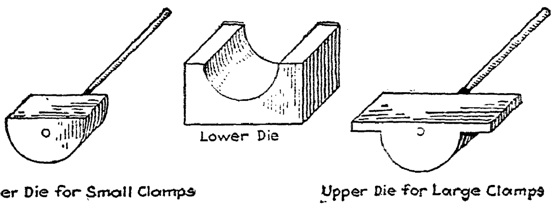


FIG. 10.—Dies for fabricating clamps.

on top of the bar. One blow from the hammer is sufficient to bend bars into shape for small clamps. Bar material less than $\frac{3}{8}$ in. thick may be bent cold, while it is necessary to heat heavier stock. Pipe clamps up to 12 in. nominal size can easily be made with dies of this sort.

Weights.—Table III gives the weights of the pipe, flanges, etc., for the various pressure standards, from which the total weight of the pipe line may be computed.

Spacing Supports.—Hangers or other supports for piping must be spaced with respect to three considerations: (1) ability to place a support at some desired location, (2) keeping sag in the line within limits that will permit drainage, and (3) avoiding excessive bending stresses from the uniform and concentrated loads between supports. Owing to the necessity of meeting requirements (2) and (3), it may be necessary at times to use some ingenuity in overcoming obstacles encountered in connection with (1).

Design formulas for calculating bending stress and deflection between supports are derived from the usual beam formulas which depend upon the method of support and the type of loading. For a given beam and a given system of supports the deflection and bending stress resulting from a combination of loads is equal to the sum of the deflections and bending stresses that would result from each of the loads applied separately. The formulas given in Table IV for bending stress and deflection have been made up on this basis from standard formulas for uniformly distributed and concentrated loads. The nomenclature used in Table IV, however, is taken from the list of symbols given herewith rather than in conformance to the symbols usually found in beam for-

TABLE IV.—EQUATIONS FOR MAXIMUM BENDING STRESS AND DEFLECTION BETWEEN SUPPORTS

Type of support	Maximum bending stress,[1] $S =$	Position of maximum bending stress	Maximum deflection $y =$	Position of maximum deflection
Single span, free ends	$\frac{(0.75wL^2 + 1.5w_fL)D}{I}$	Mid-point of span	$\frac{22.5wL^4 + 36.0w_fL^3}{EI}$	Mid-point of span
Continuous line.....	$\frac{(0.5wL^2 + 0.75w_fL)D}{I}$	At supports	$\frac{4.5wL^4 + 9.0w_fL^3}{EI}$	Mid-point of span

[1] In the case of a single span with free ends the maximum bending stress is compression in the top of the pipe and tension in the lower fibers. In the case of the continuous line the maximum bending stress occurs at the supports where it is tension in the top of the pipe and compression in the lower fibers.

mulas. As shown in Table IV, the type of end attachment assumed makes an important difference in the calculated amounts of bending stress and deflection.

List of Symbols.—The following symbols are used in the formulas for maximum deflection and bending stress:

y = maximum deflection, in.
w = $w_p + w_w + w_c$, lb per ft of pipe.
w_p = weight of pipe, lb per ft = $2.67(D^2 - d^2)$.
w_w = weight of water, lb per ft of pipe = $0.341d^2$.
w_c = weight of covering, lb per ft of pipe = $37.7y'(Dt + t^2)$, (see pages 738–743).
w_f = weight of concentrated load at center of span, lb.
D = outside diameter of pipe, in.
d = inside diameter of pipe, in.
E = modulus of elasticity, psi (see pages 296, 344).
y' = density of covering, lb per cu in. (see page 701).
t = thickness of covering, in.
S = maximum stress, psi.
L = length of span, ft.
I = moment of inertia of pipe, in.[4] (see page 858).

Deflection of Pipe Lines.—In order to prevent pocketing of condensation in a pipe line, it is necessary to pitch the line in the direction of flow of the condensation and to support the pipe in spans of such length that the pipe in any span will not sag below the elevation of the support at the low end of the span. Assuming a single span with free ends and no concentrated loads as a basic case, the deflection, or sag, between supports can be determined from the conventional formula $y = 22.5wL^4/EI$, noting that y is in inches, L is in feet, and w is in pounds per foot, and a close approximation of the necessary gradient to prevent pockets can be derived from the properties of the circle.

Referring to Fig. 11 in which the span of pipe is assumed to deflect as the arc of a circle, it is obvious that a horizontal line of supports would cause pocketing. To eliminate the pocket it would be necessary to lower one of the supports a distance indicated as $2e$ so that the tangent $f'g'$ is horizontal. From the properties of the circle it may be shown that $2e = 144L^2y/(36L^2 - y^2)$ where the span L is measured in feet, and the deflection y and the distance e are given in inches. Or if $2e/L$, the average gradient, is represented as $1/G$ (*i.e.*, 1 in. in G ft) the relationship among deflection, span, and average gradient becomes $y = -72GL$

$+ 6L\sqrt{144G^2 + 1}$. The simple expression $y = L/4G$ is an almost exact approximation of the same thing.

In Fig. 12, page 749, the calculated deflections $y = 22.5wL^4/EI$ are plotted as straight lines on log-log coordinates with a slope of 4 to 1. Superimposed on these lines are straight lines at 45-deg slope giving the relationship between deflection and average gradient from $y = L/4G$. Maximum allowable spans can be determined from Fig. 12 with reference to deflection limits which will permit drainage of condensation. The design limitations of the chart shown in Fig. 12 are stated in the explanatory note below the chart. According to these limitations, the deflection of pipes

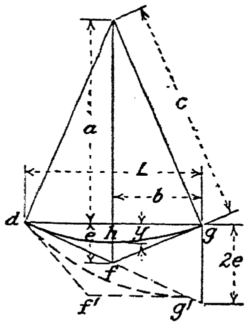


FIG. 11.—Minimum pitch required to avoid pocketing condensation as derived from the properties of a circle.

filled with water, or of pipes carrying concentrated loads, or of pipes supported as continuous lines, or of pipes having other wall thickness cannot be read directly from the chart. Hence Table V has been prepared as an adjunct to Fig. 12 to cover these other cases as demonstrated under Deflection Examples.

Deflection Examples.—The following examples serve to illustrate the combined use of Fig. 12 and Table V in determining the deflection of a pipe line under various conditions of loading and support.

Example 1.—A 4-in. Schedule 40 welded line is supported as a simple beam on hangers spaced 15 ft apart. The line is covered with standard-thickness 85 per cent magnesia. There are no concentrated loads between supports. Find the deflection between supports and the minimum gradient to prevent pocketing.

Solution.—Read vertically from 15-ft support spacing to 4-in. diameter pipe. Read deflection = 0.069 in. on scale at left. This deflection and support spacing falls between gradients of 1 in. in 50 ft and 1 in. in 60 ft. Use gradient = 1 in. in 50 ft.

Example 2.—Assume there is a concentrated load of 300 lb at the mid-point of the span in the above example. What is the deflection and what gradient would be needed?

Solution.—From Section A of Table V the uniform weight between supports is $15 \times 13.07 = 196$ lb. Concentrated load ratio $R = 300/196 = 1.53$. Interpolating from Section B of Table V the deflection factor would be 3.45. From the chart for 4-in. pipe on 15-ft span the deflection = 0.068 in. With concentrated load the deflection $= 0.068 \times 3.45 = 0.235$ in. For 15-ft span and 0.235-in. deflection the minimum gradient is found to be 1 in. in 16 ft.

Example 3.—Assume the same 4-in. Schedule 40 pipe, but filled with water and carrying a concentrated load of 300 lb, and assume that each span is considered as a part of a continuous line. Find the deflection and required gradient.

Solution.—From Section A of Table V the total uniform weight of pipe, covering, and water between supports is $18.57 \times 15 = 278$ lb. The concentrated load ratio is $300/278 = 1.08$. The deflection factor for this load ratio as interpolated from Section B of Table V for a continuous line is 0.63. Observe also that with no concentrated load the deflection factor for a continuous line would have been 0.20. Thus the deflection to be expected with the added concentrated load is $0.63/0.20 = 3.15$ times as much as for the line with only its uniform load.

Section A of Table V also gives deflection multiplication factors for a pipe filled with water. As a continuous line of 4-in. diameter the deflection of the pipe filled with water (but without concentrated load) would be 0.28 times as much as shown on Fig. 12. Combining this factor with the one developed above for the effect of the concentrated load gives an over-all factor of $3.15 \times 0.28 = 0.882$. The chart deflection for 4-in. pipe on 15-ft span (Fig. 12) = 0.068 in. The corrected deflection for this problem is $0.068 \times 0.882 = 0.060$ in. The minimum gradient necessary for 0.060-in. deflection between 15-ft supports is (from Fig. 12) 1 in. in 61 ft. More conservative design would be to use a gradient of 1 in. in 50 ft, as in Example 1.

Example 4.—Assuming the same 4-in. Schedule 40 pipe to be empty, what is the maximum distance between supports if the average gradient is to be 1 in. in 20 ft?

Solution.—On Fig. 12 locate the intersection of the deflection line for 4-in. pipe with the 1 in. in 20-ft gradient line. Vertically below this intersection is the maximum span which is read as 21.2 ft. The corresponding deflection read at the left is 0.266 in.

Example 5.—What will be the deflection in a single 20-ft span of 4-in. Schedule 160 pipe, covered but empty and with no concentrated load?

Solution.—On Fig. 12 locate the intersection of the 20-ft span line with the deflection line for 4-in. pipe. The deflection read at the left is 0.215 in. for pipe having 0.237 in. wall thickness. From Section A of Table V the deflection factor for 4-in. Schedule 160 pipe is 1.038. Therefore, the expected deflection is $0.215 \times 1.038 = 0.223$ in.

Bending Stresses in Pipes Due to Load between Supports.—Figure 13 shows, for common sizes of pipe of the thickness indicated, the bending stress for different length spans under the conditions stated below the chart. Like Fig. 12, the chart is made up for pipe with free ends supported in a single span except

TABLE V.—DEFLECTION FACTORS FOR USE WITH FIG. 12

Section A.—Deflection factors for fluid contents and heavier pipe

Nominal pipe size, inches	Pipe wall thickness, inches,[1] of Figs. 12 and 13	Weight, pounds per foot, with std thickness covering		Deflection factors			
				For pipe filled with water		For heavier pipe	
		Empty[2]	Filled with water	Single span with free ends	Continuous line	Schedule 80	Schedule 160
1/2	0.109	1.38	1.51	1.10	0.22	0.998	1.030
3/4	0.113	1.73	1.96	1.13	0.23	0.975	1.010
1	0.133	2.35	2.72	1.16	0.23	1.004	1.046
1 1/4	0.140	3.07	3.72	1.21	0.24	0.991	1.019
1 1/2	0.145	3.63	4.51	1.24	0.25	0.990	1.020
2	0.154	4.94	6.39	1.29	0.26	0.986	1.013
3	0.216	9.25	12.44	1.35	0.27	1.003	1.036
4	0.237	13.07	18.57	1.42	0.28	0.994	1.038
5	0.258	17.30	25.97	1.50	0.30	0.995	1.042
6	0.280	22.09	34.60	1.57	0.31	0.996	1.043
8	0.322	32.95	54.63	1.66	0.33	0.994	1.048
10	0.365	45.85	80.04	1.75	0.35	1.000	1.069
12	0.375	57.39	106.3	1.85	0.37	0.979	1.048
14 O.D.	0.375	63.09	122.9	1.95	0.39	0.993	1.052
16 O.D.	0.375	72.20	151.4	2.10	0.42	0.983	1.038
18 O.D.	0.375	81.80	183.0	2.25	0.45	0.978	1.040
20 O.D.	0.375	90.41	216.6	2.40	0.48	0.987	1.052
24 O.D.	0.375	108.6	292.6	2.70	0.54	0.978	1.048
30 O.D.	0.500	174.8	387.8	2.90	0.58		
36 O.D.	0.500	210.3	691	3.29	0.66		
48 O.D.	0.625	346.4	1,205	3.48	0.70		
60 O.D.	0.750	510	1,855	3.64	0.73		
72 O.D.	0.875	707	2,647	3.74	0.75		

Section B.—Deflection factors for concentrated loads

Load ratio[3] *R*	Single span with free ends	Continuous line	Load ratio[3] *R*	Single span with free ends	Continuous line	Load ratio[3] *R*	Single span with free ends	Continuous line
0	1.00	0.20	0.70	2.12	0.48	1.4	3.24	0.76
0.10	1.16	0.24	0.80	2.28	0.52	1.5	3.40	0.80
0.20	1.32	0.28	0.90	2.44	0.56	1.6	3.56	0.84
0.30	1.48	0.32	1.00	2.60	0.60	1.7	3.72	0.88
0.40	1.64	0.36	1.1	2.76	0.64	1.8	3.88	0.92
0.50	1.80	0.40	1.2	2.92	0.68	1.9	4.04	0.96
0.60	1.96	0.44	1.3	3.08	0.72	2.0	4.20	1.00

[1] Schedule 40 (standard weight) in sizes 1/2 to 10 in., inclusive.
[2] Deflection chart Fig. 12 is drawn for this condition.
[3] Ratio *R* is the ratio of concentrated load (assumed at mid-point of span) to total uniform load between supports.

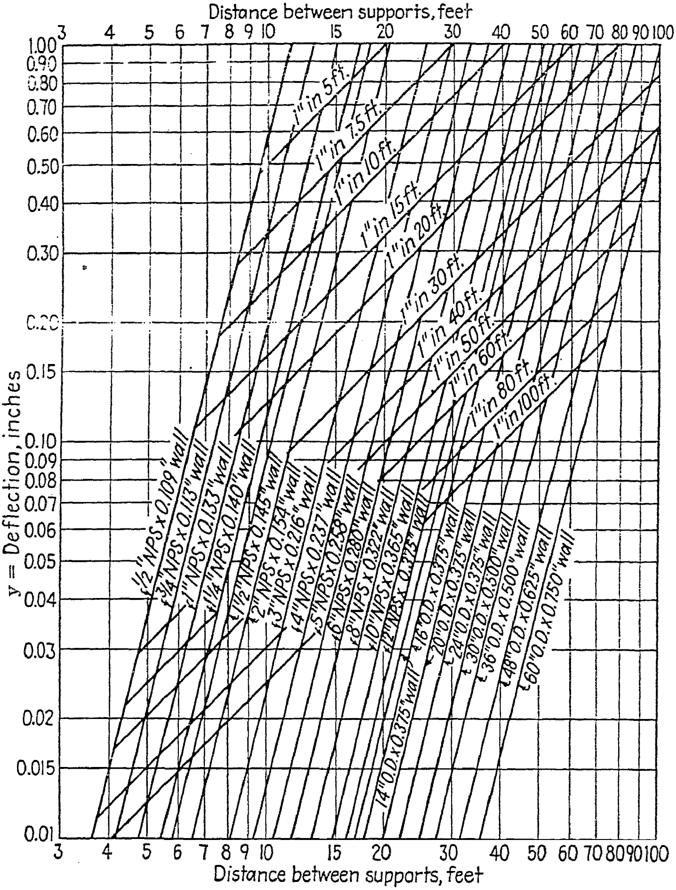


FIG. 12.—Deflection of pipes between supports.

Deflection of Pipe between Supports.—Deflections are for *empty* pipe of the thickness shown in Table V when covered with standard-thickness 85 per cent magnesia supported in a single span with free ends and carrying no concentrated load. The deflection of heavier walled pipe may be computed from Fig. 12 by applying the deflection factors given in Table V. The line is assumed *empty* because this approximates the condition for which pocketing of condensation must be avoided in air, gas, or steam piping. The assumption of a single span with free ends is the least favorable condition of support. If the ends of the

that, where the deflection chart assumed the pipe to be empty, Fig. 13 assumes it to be filled with water, and the load between supports to include a pair of 125-psi standard cast-iron flanges also. Thus the bending stress chart approaches the worst condition of stress that is apt to be encountered. The assumption that a steam pipe may be full of water is made on the supposition that the hangers should be spaced so as to support the pipe adequately if filled with water, either accidentally or for the purpose of a hydrostatic test. Where the ends are restrained or where there are several spans so that each may be considered as one of a continuous line, the bending stresses will be less than those shown in Fig. 13.

For other pipe thickness schedules the bending stresses will differ somewhat from those shown in Fig. 13. As illustrated in the examples, the *bending stress factors* for Schedules 80 and 160 pipe given in Table VI can be applied to the values read from Fig. 13 so as to adjust these values to the heavier pipe.

Table VII gives the bending stress in a continuous line of bare pipe filled with water, for the wall thicknesses and under the conditions stated below the table.

TABLE VI.—BENDING STRESS FACTORS FOR USE WITH FIG. 13[1]

Pipe size	Bending stress factors		Pipe size	Bending stress factors	
	Schedule 80	Schedule 160		Schedule 80	Schedule 160
1/2	1.596	1.534	6	1.184	1.163
3/4	1.363	1.285	8	1.195	1.110
1	1.374	1.259	10	1.138	1.110
1 1/4	1.287	1.203	12	1.049	1.079
1 1/2	1.323	1.209	14	0.967	1.017
2	1.382	1.174	16	0.879	0.945
3	1.189	1.253	18	0.865	0.896
4	1.193	1.161	20	0.821	0.836
5	1.237	1.246	24	0.810	0.739

[1] The pipe is assumed to be filled with water, covered with standard-thickness 85 per cent magnesia and carrying a concentrated load at mid-point of span approximately equal to a pair of slip-on 900-lb flanges on Schedule 80 pipe, and 1,500-lb flanges on Schedule 160 pipe.

span were not free or if it were considered part of a continuous line, the computed deflection would be only one-fifth as much as shown on the chart. Nevertheless the supports should be spaced frequently enough to cover such contingencies as concentrated load, settlement, improper adjustment of hangers, and poor alignment of pipe. Beyond this, in order to ensure adequate drainage, the line should be pitched more than enough merely to avoid pocketing condensation. Hence the chart spacing is recommended.

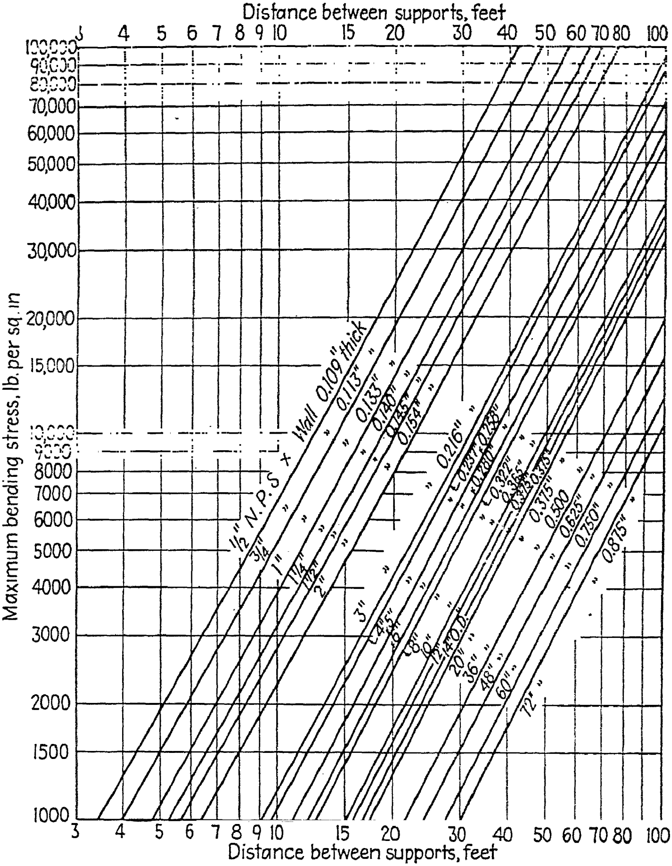


FIG. 13.—Bending stress in covered pipes due to load between supports, single span with free ends.

Bending Stress in Pipe Due to Load between Supports.—The maximum bending stresses shown in the chart are caused by the weight of pipe, standard-thickness 85 per cent magnesia covering, 125-lb standard cast-iron flanges, and fluid contents. The flanges are assumed at the middle of the span, and the pipe is assumed to be filled with water. Wall thicknesses for nominal pipe diameters ½ to 10 in., inclusive, are for Schedule 40 (standard-weight) pipe (see Table V). Wall thicknesses of pipe 12 to 72 in., outside diameter, were computed for 200 to 400 psi steam service pressure depending on pipe size and material. The chart

TABLE VII.—BENDING STRESS IN BARE PIPES DUE TO WEIGHT OF PIPE AND CONTENTS, CONTINUOUS LINE[1]

Nominal pipe size, inches	Maximum bending stress in pipe wall, psi			
	Distance between supports, feet			
	10	15	20	25
2	957	2,100	3,700	5,750
2½	763	1,699	3,000	4,670
3	644	1,435	2,595	3,940
4	528	1,172	2,073	3,220
5	534	1,122	1,923	2,940
6	460	967	1,662	2,545
8	362	765	1,320	2,025
10	301	638	1,102	1,690
12	274	579	1,008	1,527
14 O.D.	268	562	965	1,475
16 O.D.	254	533	911	1,390
18 O.D.	238	501	860	1,313
20 O.D.	231	484	828	1,246
24 O.D.	219	457	780	1,190

assumed to be full of water and without covering. Schedule 40 (standard-weight) steel [illegible]

puted for continuous beam conditions by formula in Table IV.

Stress Examples.—The following examples serve to illustrate the combined use of Fig. 13 and Table VI in determining the stress due to bending moment in the worst case of loading, *i.e.*, a single span of covered pipe, filled with water and carrying a concentrated load corresponding to the weight of a pair of flanges at the mid-point of the span.

Example 1.—A 4-in. Schedule 40 pipe covered with standard-thickness 85 per cent magnesia and full of water is supported on hangers spaced 20 ft apart. Find the bending stress due to the load between supports.

Solution.—Referring to Fig. 13, read vertically from 20-ft support spacing to 4-in.-diameter pipe. On the left-hand scale read bending stress, 4,000 psi.

Example 2.—A 6-in. Schedule 160 pipe covered with standard-thickness 85 per cent magnesia and full of water is supported on hangers spaced 20 ft apart. Find the bending stress due to the load between supports.

Solution.—Referring to Fig. 13, read vertically from 20-ft support spacing to 6-in.-diameter pipe. On the left-hand scale read bending stress, 2,700 psi. From Table VI read bending stress multiplication factor for 6-in. Schedule 160 pipe = 1.163. From which the bending stress in 6-in. Schedule 160 pipe on 20-ft support spacing = 2,700 × 1.163 = 3,140 psi.

was computed from the formula of Table IV for bending stress for pipe supported in a single span with free ends.

Spacing of Supports for Thin-wall Pipe.—Spacing of supports for large-diameter thin-wall pipe cannot be determined by the rules used for medium-to-heavy wall pipe given herein. Maximum safe spans for thin-wall pipe of various diameters and gage numbers when carried on saddle supports are given in Table VIII. These spans were computed by the method of analysis outlined by Prof. Roark[1] and are reproduced here in condensed form by permission from the "Handbook of Welded Steel Pipe" published by the California Corrugated Culvert Co. For information on safe spans where ring-girder stiffener angles and plates are used at points of support, reference may be made to the "Handbook of Welded Steel Pipe" or to a paper by Schorer.[2] A notable increase in permissible spans is obtained through reinforcement where the D/t ratio is above 100.

Table VIII.—Recommended Safe Spans for Lightweight Steel Pipe of Diameters and Manufacturers' Gauge Thicknesses Shown[1]
(Without reinforcement at supports)

Nominal pipe diameter, inches	Maximum span, feet					
	16 gauge	14 gauge	12 gauge	10 gauge	7 gauge	3 gauge
4	30	32	36	38		
5	31	33	37	40		
6	32	35	39	42		
8	16	25	41	45	49	
10	..	17	30	46	51	
12	..	12	22	34	53	58
14	..	..	16	26	43	60
16	..	..	13	21	34	57
18	..	..	10	16	28	46
20	..	..	8	14	23	38
22	..	..	7	11	20	32
24	..	..	6	10	16	28
30	..	..	..	6	11	19
36	..	..	..	4	8	13
42	..	..	..	..	6	10
48	..	..	..	..	4	8

[1] The above table is based on strength of the pipe as a beam and resistance to crushing at the supports, these supports being 12 in. wide and extending around one-third of the pipe periphery. For decimal equivalents of gauge numbers, see column for AISI inch equivalent for U.S. sheet steel thickness in Table III on page 22.

[1] See "A Study of Circumferential Bending of Pipes and Cylindrical Containers," by Prof. Raymond J. Roark, Univ. Wisconsin Engineering Experiment Station *Bull.* 69, 1929.

[2] See "Design of Large Pipe Lines," by Herman Schorer, *Trans. ASCE*, Vol. 98, 1933.

CHAPTER VII

EXPANSION AND FLEXIBILITY

ELONGATION DUE TO CHANGE IN TEMPERATURE

General.—When the temperature of a pipe is raised or lowered, there is a corresponding increase or decrease in both length and diameter. The amount of this expansion or contraction is directly proportional to the dimensions of the pipe and to the change in temperature. If the pipe has not been stressed enough to take a permanent set or exposed to high enough temperature for "creep" to occur, it will again resume its original dimensions when normal temperature is restored. In ordinary piping practice, the change in length between atmospheric temperature during erection and operating temperature in service is of interest rather than any incidental change in diameter, since the latter is too small, even with extreme temperature changes, to be of much consequence. The change in length between erection and service conditions may be either an expansion in the case of pipes carrying hot fluids or a contraction in case the fluids are cold, as in refrigerating lines.

Coefficient of Expansion.—The amount of linear expansion (or contraction) per unit length of a material per degree change in temperature is termed the "coefficient of linear expansion" of that material or, commonly, the "coefficient of expansion." The coefficient of expansion is not the same for different materials and for a given material usually varies appreciably at different temperatures or, in other words, it is not constant throughout an extensive temperature range. For this reason, it is desirable to use an average coefficient of expansion for each material which applies to the particular temperature range obtaining. When the average coefficient of expansion is known, the change in length of a pipe line i computed by the formula $\Delta = 12CL(t_2 - t_1)$, where Δ is the increase in length in inches; C the average coefficient of expansion for the temperature range t_1 to t_2, expressed as the decimal fraction which a unit length changes per degree Fahrenheit

increase (or decrease) in temperature; L the length of the line in feet at temperature t_1; t_1 the initial temperature and t_2 the final temperature in degrees Fahrenheit; and 12 the number of inches in 1 ft.

According to tests made by Holborn and Day and others,[1] the length of a pipe at temperatures from 32 to 1800 F can be computed from the formula

$$L_t = L_0\left[1 + a\left(\frac{t - 32}{1,000}\right) + b\left(\frac{t - 32}{1,000}\right)^2\right]$$

where L_t is the length at t F; L_0 the length at 32 F (L_t and L_0 may be expressed either in feet or inches provided both are in the same unit); t the final temperature in degrees Fahrenheit; and a and b constants as shown below:

Metal	a	b
Cast iron	0.005441	0.001747
Steel	0.006212	0.001623
Wrought iron	0.006503	0.001622
Copper	0.009278	0.001244
Brass	0.010188	0.003776

The elongations of pipe in inches per 100 ft given in Table I for cast iron, steel, wrought iron, and copper were computed by the above formula for temperatures from 0 to 1000 F. The source of the data for 4 to 6 per cent chromium steel, 12 per cent chromium steel, 18 per cent chromium, 8 per cent nickel, etc., is indicated in Table I. The elongations from 0 F to −200 F were computed from data based on the thermal expansion of pure metals.[1] Values below 0 F are given as negative so that in all cases the algebraic difference between the elongation corresponding to the higher and lower temperature will give the total expansion between the two temperatures. For example, the expansion of carbon steel per 100 ft from −40 F to 700 F is found as follows:

Reading from Table I, 6.084 − (−0.288) = 6.084 + 0.288 = 6.372 in. per 100 ft. For the more usual case where both temperatures are above 0 F, such as expansion from 60 F to 700 F, the elongation is found as follows:

Reading from Table I, 6.084 − 0.448 = 5.636 in. per 100 ft.

[1] See source reference 2(c) below Table I.

TABLE I.—THERMAL EXPANSION OF PIPE IN INCHES PER 100 FEET

Saturated steam, vacuum in. Hg below 212 F, pressure, psi gauge above 212 F	Temperature, degrees Fahrenheit	Cast iron[1]	Carbon and carbon molybdenum steel[1]	Wrought iron[1]	4–6% Cr alloy steel[7]	12% Cr stainless steel[8]	18 Cr-8 Ni stainless steel[7]	Copper[1]	Brass[3]
	−200	−1.058	−1.282	−1.289	−1.250	−1.170	−2.030	−1.955	−2.065
	−180	−0.982	−1.176	−1.183	−1.150	−1.070	−1.850	−1.782	−1.890
	−160	−0.891	−1.066	−1.073	−1.030	−0.970	−1.670	−1.612	−1.705
	−140	−0.797	−0.948	−0.955	−0.970	−0.870	−1.480	−1.428	−1.508
	−120	−0.697	−0.826	−0.833	−0.800	−0.750	−1.300	−1.235	−1.308
	−100	−0.593	−0.698	−0.705	−0.700	−0.630	−1.090	−1.040	−1.098
	− 80	−0.481	−0.563	−0.570	−0.550	−0.520	−0.880	−0.835	−0.888
	− 60	−0.368	−0.428	−0.435	−0.430	−0.400	−0.670	−0.630	−0.673
	− 40	−0.248	−0.288	−0.295	−0.290	−0.270	−0.450	−0.421	−0.452
	− 20	−0.127	−0.145	−0.152	−0.145	−0.130	−0.225	−0.210	−0.227
	0	0	0	0	0	0	0	0	0
	20	0.128	0.148	0.154	0.140	0.140	0.223	0.238	0.233
	32	0.209	0.230	0.249	0.234	0.234	0.356	0.366	0.373
	40	0.263	0.285	0.313	0.280	0.280	0.446	0.451	0.466
Vacuum inches of Hg: 29.39	60	0.391	0.448	0.468	0.430	0.430	0.669	0.684	0.690
28.89	80	0.522	0.580	0.628	0.600	0.600	0.892	0.896	0.920
27.99	100	0.660	0.753	0.787	0.750	0.750	1.115	1.134	1.150
26.48	120	0.799	0.910	0.958	0.900	0.900	1.338	1.366	1.390
24.04	140	0.924	1.064	1.113	1.050	1.050	1.545	1.590	1.625
20.27	160	1.073	1.223	1.275	1.220	1.220	1.784	1.804	1.865
14.63	180	1.218	1.383	1.445	1.370	1.370	2.000	2.051	2.100
6.45	200	1.368	1.546	1.626	1.520	1.520	2.230	2.296	2.340
Pressure, psi gauge: 0	212	1.451	1.643	1.721	1.600	1.600	2.361	2.428	2.467
2.5	220	1.507	1.707	1.784	1.675	1.675	2.460	2.516	2.580
10.3	240	1.653	1.875	1.958	1.825	1.825	2.680	2.756	2.830
20.7	260	1.804	2.038	2.127	2.000	2.000	2.920	2.985	3.070
34.5	280	1.958	2.205	2.313	2.150	2.150	3.130	3.218	3.315
52.3	300	2.106	2.374	2.478	2.320	2.320	3.375	3.461	3.565
74.9	320	2.268	2.545	2.648	2.470	2.470	3.615	3.696	3.820
103.3	340	2.416	2.717	2.836	2.625	2.625	3.840	3.941	4.065
138.3	360	2.573	2.884	3.023	2.820	2.780	4.075	4.176	4.320
180.9	380	2.732	3.066	3.198	2.980	2.980	4.346	4.424	4.560
232.4	400	2.881	3.230	3.369	3.140	3.130	4.560	4.666	4.825
293.7	420	3.055	3.421	3.568	3.300	3.300	4.800	4.914	5.080
366.1	440	3.218	3.595	3.748	3.470	3.470	5.045	5.154	5.340
451.3	460	3.384	3.784	3.944	3.650	3.650	5.335	5.408	5.600
550.3	480	3.556	3.955	4.128	3.800	3.800	5.540	5.651	5.925
664.3	500	3.720	4.151	4.325	4.000	4.000	5.800	5.906	6.120
795.3	520	3.893	4.342	4.525	4.150	4.150	6.050	6.148	6.380
945.3	540	4.063	4.525	4.714	4.350	4.340	6.320	6.410	6.650
1,115	560	4.238	4.715	4.905	4.540	4.500	6.572	6.64	6.920
1,308	580	4.414	4.906	5.116	4.740	4.640	6.835	6.919	7.170
1,525	600	4.598	5.102	5.303	4.920	4.850	7.100	7.184	7.440
1.768	620	4.769	5.292	5.508	5.100	5.020	7.370	7.432	7.715
2,041	640	4.955	5.482	5.698	5.280	5.180	7.630	7.689	7.980
2,346	660	5.133	5.686	5.915	5.470	5.350	7.900	7.949	8.240
2,705	680	5.315	5.875	6.108	5.670	5.550	8.170	8.196	8.515
3,080	700	5.502	6.084	6.329	5.850	5.700	8.425	8.472	8.780
	720	5.681	6.280	6.521	6.050	5.900	8.670	8.708	9.050
	740	5.879	6.490	6.747	6.220	6.010	8.932	8.999	9.324
	760	6.073	6.688	6.948	6.430	6.280	9.220	9.256	9.600
	780	6.262	6.901	7.162	6.600	6.480	9.480	9.532	9.870
	800	6.460	7.105	7.356	6.830	6.680	9.750	9.788	10.150
	820	6.652	7.319	7.605	7.030	6.890	10.020	10.068	10.425

TABLE I.—(*Concluded*)

Saturated steam, vacuum in. Hg below 212 F, pressure, psi gauge above 212 F	Temperature, degrees Fahrenheit	Cast iron[1]	Carbon and carbon molybdenum steel[1]	Wrought iron[1]	4–6% Cr alloy steel[7]	12% Cr stainless steel[8]	18 Cr–8 Ni stainless steel[7]	Copper[1]	Brass[3]
	840	6.843	7.517	7.800	7.200	7.090	10.270	10.308	10.690
	860	7.049	7.743	8.043	7.400	7.300	10.540	10.610	10.975
	880	7.248	7.953	8.248	7.580	7.500	10.820	10.971	11.250
	900	7.452	8.168	8.487	7.770	7.720	11.075	11.156	11.545
	920	7.668	8.400	8.715	7.970	7.950	11.350	11.421	11.815
	940	7.862	8.610	8.937	8.170	8.140	11.620	11.707	12.120
	960	8.073	8.830	9.148	8.360	8.350	11.900	11.976	12.420
	980	8.279	9.051	9.395	8.560	8.550	12.150	12.269	12.720
	1000	8.490	9.276	9.624	8.760	8.750	12.432	12.543	13.080

NOTE.—Values for cast iron, carbon and carbon-molybdenum steel, wrought iron, and copper above 0 F were computed from the Holborn and Day formula, p. 755. For basis of other materials and for temperatures below 0 F, see source references.

Source References for Tables I and II

[1] "Smithsonian Physical Tables," 7th rev. ed., p. 218; "Mechanical Engineers' Handbook," by Marks, 4th ed., p. 297; Hütte, 25th ed., Vol. 1, p. 439. Constants adjusted to English units.

[2] "The Thermal Expansion of Pure Metals," by F. C. Nix and D. MacNair, *Phys. Rev.*, Vol. 60, Oct. 15, 1941, and Vol. 61, Jan. 15, 1942.

[3] "Thermal Expansion of Some Industrial Copper Alloys," by Peter Hidnert and Geo. Dickson, *Bur. Standards J. Research*, Vol. 31, RP 1550, p. 77, August, 1943.

[4] "Thermal Expansion Characteristics of Some Nickel Cast Irons," by T. J. Wood, *Trans. ASM*, Vol. 23, p. 455, 1935.

[5] "Thermal Expansion of Metals," by N. L. Mochel, Symposium on Effect of Temperature on Metals, ASTM-ASME, 1931, p. 453.

[6] "Thermal Expansion of Magnesium and Some of Its Alloys," by P. Hidnert and W. T. Sweeney, *Bur. Standards J. Research*, Vol. 1, RP 29, p. 771, 1928.

[7] "Thermal Expansion of Heat Resisting Iron Alloys," by J. B. Austin and R. H. H. Pierce, Jr., *Ind. Eng. Chem.*, Vol. 25, No. 7, p. 778, July, 1933.

[8] "Thermal Expansion of Alloys of the 'Stainless Iron' Type," by P. Hidnert and W. T. Sweeney, Bureau of Standards *Sci. Paper* 570, Vol. 22, p. 639, 1927–1928.

[9] "Nickel-Molybdenum-Iron and Related Alloys," by F. T. McCurdy, *Proc. ASTM*, Vol. 39, p. 698, 1939.

[10] "Studies Relating to Use of Saran for Water Pipes in Buildings and for Service Lines," by F. M. Dawson and A. A. Kalinske, *Jour. AWWA*, August, 1945, p. 1058.

[11] Haynes Stellite Co.

[12] "Thermal Expansion of Nickel, Monel Metal, Stellite, Stainless Steel and Aluminum," by W. H. Souder and P. Hidnert, Bureau of Standards *Sci. Paper* 426, Vol. 17, p. 497, 1922.

[13] "Thermal Expansion of Molybdenum," by P. Hidnert and W. B. Gero, Bureau of Standards *Sci. Paper* 488, Vol. 19, p. 429, 1923–1924.

[14] "Thermal Expansion of Tungsten," by P. Hidnert and W. T. Sweeney, Bureau of Standards *Sci. Paper* 515, Vol. 20, p. 483, 1924–1926.

[15] "Coefficient of Expansion of Carbon-Moly Pipe," by W. A. Tucker, Westinghouse Report to Edison Electric Institute, 1938.

[16] "Thermal Expansion of Beryllium and Aluminum Beryllium Alloys," by P. Hidnert and W. T. Sweeney, Bureau of Standards *Sci. Paper* 565, Vol. 22, 1927–1928.

[17] "Pure Zinc at Normal and Elevated Temperatures," by J. R. Freeman, Jr., F. Sillers, Jr., P. F. Brandt, Bureau of Standards *Sci. Paper* 522, Vol. 20, p. 661, 1924–1926.

TABLE II.—MEAN COEFFICIENTS OF LINEAR EXPANSION, INCHES PER INCH PER DEGREE FAHRENHEIT × 10^{-6}

Example.—$11.70 \times 10^6 = 0.0000117$

Material*	−148 to 32 F	32 to 212 F	32 to 392 F	32 to 572 F	32 to 752 F	32 to 932 F	32 to 1112 F	32 to 1292 F	Source†
Aluminum	11.70	13.30	13.70	13.90	14.50	14.80	15.50	15.70	2
Beryllium	6.20ᵉ	6.78	7.30	7.73	8.23	8.62	8.90	9.30ᵉ	16
Brass	9.20	9.70	10.00	10.30	10.50	10.70	11.10	11.40	3
Bronze	9.20	9.70	9.85	10.30	10.50	10.70	11.10	11.40	3
Colmonoy	.....	.....	.....	.....	.....	.....	.....	8.68	Mfr.
Copper	8.70	9.20	9.40	9.60	9.90	10.10	10.30	10.50	2,3
Hastelloy (60% Ni, 20% Mo, 20% Fe)	.....	6.1							11
Iron, pure	5.70	6.70	6.95	7.40	7.70	7.90	8.10	8.20	2
Iron, wrought	5.70ᵉ	6.70	6.95	7.40	7.70	7.90	8.10	8.20	1
Iron, cast:									
Gray	4.60	5.80	6.10	6.50	6.90	.....	.....	.....	1,5
Malleable	4.40	5.00	5.50	6.00	6.60	.....	.....	.....	5
5% nickel	4.20ᵉ	4.30	5.30	6.20	6.70	.....	.....	.....	4
Ni-resist (14% Ni, 6% Cu)	7.60ᵉ	10.30	10.40	10.60	10.60	.....	.....	.....	4
[illegible]% nickel	3.40ᵉ	4.10	5.10	6.80	7.30	.....	.....	.....	4
36% nickel	2.10ᵉ	2.30	3.70	4.90	5.70	.....	.....	.....	4
Duriron (14% Si)	.....	4.00	.....	.....	.....	.....	.....	.....	Mfr.
Lead	15.50	16.20	16.50ᵉ						2
Magnesium	14.0	14.50	15.00	15.50	16.10				
Molybdenum	2.60	2.72	2.80	2.85	2.90	3.06	3.10ᵉ	3.15ᵉ	2,13
Monel	7.70	8.05	8.25	8.45	8.60	8.70	8.84	8.90ᵉ	12
Nickel, pure	6.30	7.20	7.50	7.80	8.20	8.33	8.50	8.60	2,12
Nickel-chrome-iron (62% Ni, 15% Cr, 23% Fe)	7.3ᵉ	7.62	7.75	7.95	8.15	8.33	8.50	8.70	Mfr.
Nickel-molybdenum-iron (60% Ni, 20% Mo, 20% Fe)	5.50	6.10	6.40	6.60	6.90	7.20	7.40	7.70	9
Saran (vinylidene chloride plastic)		88.00							10
Silver	7.30	10.50							2
Steel, alloy:									
C-0.50% Mo	5.90ᵉ	6.50	6.90	7.20	7.50	7.90	8.00	8.10ᵉ	15
1% Cr 0.50% Mo	6.40ᵉ	6.70	7.00	7.33	7.67	7.95	8.12	8.17	7
5% Cr-0.50% Mo	5.80ᵉ	6.12	6.42	6.73	7.00	7.22	7.38	7.44	7
36% Ni (invar)	0.83ᵉ	1.66	3.33	5.97	5.97	6.8	7.40	7.45	5
Steel, carbon:									
0.10-0.20% C	5.90ᵉ	6.50	6.90	7.20	7.50	7.90	8.00	8.10ᵉ	5
0.30-0.40% C	5.85	6.40	6.60	7.00	7.40	7.80	7.90	8.05ᵉ	5
0.50-0.60% C	5.80ᵉ	6.20	6.55	6.90	7.30	7.70	7.80	8.00ᵉ	5
Steel, stainless:									
12% C	5.40	6.10	6.40	6.70	6.80	7.20	7.30	7.40	8
17% Cr	5.60ᵉ	5.85	6.23	6.40	6.55	6.73	6.84	6.95	7
25-30% Cr	5.85ᵉ	5.90	5.95	6.00	6.17	6.28	6.38	6.56	7
18% Cr-8%	9.00ᵉ	9.28	9.53	9.78	10.06	10.28	10.45	10.60	7
25% Cr-12% Ni	8.05ᵉ	8.33	8.72	9.12	9.33	9.55	9.78	9.90	7
Stellite (60% Co, 30% Cr, 4% W)	5.70	6.10	6.50	6.90	7.20	7.30	7.50	7.60	5,11
Tungsten	2.35	2.37	2.43	2.50	2.50	2.55	2.60ᵉ	2.65ᵉ	14
Zinc, pure	21.2ᵉ	21.50	21.8	21.8					17

ᵉ = extrapolated values.

* For chemical abbreviations, see p. 347.

† For source references, see Table I, p. 757.

The elongation of carbon-molybdenum steel pipe is given as the same as carbon steel in Table I. Dilatometer measurements of the expansion of carbon-molybdenum steel have given values from slightly less to approximately 5 per cent greater than carbon steel but the differences are within the range of values found for different samples.

The mean coefficients of linear expansion of more than 40 ferrous and nonferrous materials are given in Table II. The source reference is given in each case. Extrapolated values are so marked. An effort has been made to select data that are consistent with the known expansion characteristics of each class of material. For example, as the chromium content of a steel is increased, other factors remaining constant, the mean coefficient of expansion should decrease. The mean coefficient of expansion for a given temperature range such as 32 to 752 F is found by dividing the sum of the instantaneous values at 32 F and at 752 F by two. For practical purposes the mean value represents the average change in length per unit of length per degree Fahrenheit change in temperature over the range.

Effect of Constraint.—Where piping is not free to expand or contract with changes in temperature, tensile or compressive stresses are set up in the material in proportion to the amount of constraint. As shown by Table III these stresses may be very high where much change in temperature is involved and the piping is fully restrained from expansive movement. Failure to provide for such expansion may cause joints to leak or fail, and other damage to the piping and supports may ensue. It is customary, therefore, to provide elements for taking up expansion, if and when required, as described in the next section.

There are borderline cases, however, where the need for providing expansion elements may be questioned. For instance, gas, oil, sewer, and water piping, either buried or aboveground, may be subjected to seasonal as well as cyclical temperature variations in the order of 10 to 100 F depending on exposure to atmospheric conditions and similar influences (see pages 761, 762). In arriving at a rational solution for such cases where conditions can be estimated or predetermined, consideration should be given to the probable behavior of the pipe as a long column in compression with attendant likelihood of buckling, and to the magnitude of the reactions at anchor points or equivalent restraints.

The American Institute of Steel Construction has established formulas[1] (see notes, Table IV) for computing safe stresses in steel pipes axially loaded as columns. The allowable compressive stresses computed for various slenderness ratios by these formulas are given in Table IV. By definition, the slenderness ratio of a column is l/k, *viz.*, its length divided by its radius of gyration. Equivalent formulas and values for cast-iron pipe taken from the building code of the State of New York are given also in Table IV.

For convenience in solving such problems, the radius of gyration, cross-sectional area of metal, and moment of inertia for each diameter of Schedule 40 (standard-weight) and Schedule 80 (extra-strong) pipe are given in Table V. Two purposes are served by Table IV. First, it can be used in connection with Table III to determine whether a straight run of pipe without expansion joints between anchors will buckle from changes in atmospheric temperature or from a small range of operating temperature. Second, Table IV can be used to determine safe loadings for different slenderness ratios where pipe is used for structural members such as columns.

A comparison of the compressive stress set up by temperature change in a pipe fully restrained from elongating as given in Table III with the allowable compressure stresses given in Table IV gives a direct check on whether the possibility of buckling or the provision of sidewise support need be considered. For example, a temperature change of 70 F in a steel pipe fully restrained from expansion causes a compressive stress of 13,650 psi, as given in Table III. Reference to Table IV shows that a column having an l/k ratio greater than 80 would not be considered safe for a compressive stress of 13,650 psi. Hence, if a 10-in. Schedule 40 pipe with a radius of gyration of 3.67, as given in Table V, were so loaded, the maximum unsupported length should not exceed $l = 80 \times 3.67 = 294$ in. or 24.5 ft.

Where expansive movement is fully restrained, the thrusts at anchor points or equivalent restraints can be computed as follows by using the values of Table III in connection with the cross-sectional areas of pipe metal given in Table V.

$$F = SA$$

[1] American Institute of Steel Construction Manual, 3d ed., 1939, p. 127, 101 Park Ave., New York 16, N.Y.

where F = reacting force at anchor point, lb.
S = tensile or compressive stress in pipe material, psi, from Table III.
A = cross-sectional area of pipe metal, sq in.

In a similar manner the total load that may safely be supported by pipe acting as a column can be computed using the values for safe compressive stress given in Table IV. In the foregoing example, the thrust on the anchor $F = SA = 13{,}650 \times 11.91 = 162{,}500$ lb also represents the safe axial load for a 10-in. Schedule 40 pipe column approximately 25 ft long.

TABLE III.—LONGITUDINAL TENSILE OR COMPRESSIVE STRESS SET UP IN PIPE MATERIAL WHEN THERMAL EXPANSION IS FULLY RESTRAINED

Temperature change, deg F	S = stress in pipe material, psi[1]								
	Steel	Wrought iron	Cast iron	Brass or bronze	Copper	Lead	Cement asbestos	Concrete	Wood
10	1,950	1,876	696	1,358	1,472	405	231	195	53
20	3,900	3,752	1,392	2,716	2,944	810	462	380	105
30	5,850	5,628	2,088	4,074	4,416	1,215	694	585	158
40	7,800	7,504	2,784	5,432	5,888	1,620	925	780	210
50	9,750	9,380	3,480	6,790	7,360	2,025	1,156	975	263
60	11,700	11,256	4,176	8,148	8,832	2,430	1,387	1,170	315
70	13,650	13,132	4,872	9,506	10,304	2,835	1,618	1,365	368
80	15,600	15,008	5,568	10,864	11,776	3,240	1,850	1,560	420
90	17,550	16,884	6,264	12,222	13,248	3,645	2,081	1,755	473
100	19,500	18,760	6,960	13,580	14,720	4,050	2,312	1,950	525
	Modulus of linear elasticity, 1,000,000 psi								
	30	28	12	14	16	2.5	3.4	3.0	1.5
	Coefficient of linear expansion per deg F $\times 10^{-6}$								
	6.5	6.7	5.8	9.7	9.2	16.2	6.8	6.5	3.5

[1] Computed from relation: $S = CE(t_2 - t_1)$
where S = longitudinal tensile or compressive stress in pipe material, psi.
C = coefficient of linear expansion per degree F, see Table II, p. 758.
E = modulus of elasticity for pipe material, psi, see Table I, p. 344.
t_1 = initial temperature, F.
t_2 = final temperature, F.

Outdoor, or buried, cold lines carrying gas, oil, sewage, or water within the ordinary range of atmospheric temperatures often extend for a long way in the same direction without any bend or

TABLE IV.—ALLOWABLE COMPRESSIVE STRESS IN STEEL AND CAST-IRON PIPE COLUMNS

Slenderness ratio, l/k	Allowable compressive stress, S, psi: Steel[1]	Allowable compressive stress, S, psi: Cast iron[2]	Slenderness ratio, l/k	Allowable compressive stress, S, psi, steel[1]	Slenderness ratio, l/k	Allowable compressive stress, S, psi, steel[3]
0	17,000	9,000				
10	16,950	8,600	110	11,130	210	3,340
20	16,810	8,200	120	10,020	220	3,060
30	16,560	7,800	130	9,280	230	2,780
40	16,220	7,400	140	8,620	240	2,580
50	15,790	7,000	150	8,000	250	2,370
60	15,250	6,600	160	7,430	260	2,180
70	14,620	6,200	170	6,910	270	2,030
80	13,900		180	6,430	280	1,880
90	13,070		190	5,990	290	1,800
100	12,150		200	5,590	300	1,640

[1] The values of S in the above table were computed by the following formulas: *Steel pipe columns* (From American Institute of Steel Construction Manual, 3d ed., 1939, p. 127):
Main members, l/k up to 120,

$$S = 17,000 - 0.485 \frac{l^2}{k^2}$$

Secondary members, l/k 120 to 200,

$$S = \frac{18,000}{1 + [illegible]}$$

[2] *Cast-iron pipe columns* (From New York Building Law, 1929):
For values of l/k up to 70,

$$S = 9,000 - 40 \frac{l}{k}$$

[3] For values of l/k greater than 200,

$$S = \frac{\pi^2 E}{2} \left(\frac{k}{l}\right)^2.$$

This is Euler's formula for hinged ends with a factor of safety of 2 inserted.

DEFINITIONS:

A = cross-sectional area of pipe metal, sq in.
l = length of column, in.
k = radius of gyration = $\sqrt{I/A}$.
S = allowable compressive stress, psi.
$F = S \times A$ = allowable axial load, lb.

dip which would enable the inherent flexibility of the line to care for its own expansion. Where lines are made up with bell joints, compression sleeve couplings, or equivalent devices affording some slip, this usually suffices for taking up the relatively small amount of thermal expansion involved. Solid-welded lines and those having other types of fixed joints, however, may need some positive relief such as alternating the fixed joints with compression sleeve couplings or similar joining devices providing slip (see page 1199).

TABLE V.—DATA FOR COMPUTING ALLOWABLE AXIAL LOAD ON STEEL PIPE

Nominal pipe size, in.	Schedule 40 steel pipe					Schedule 80 steel pipe				
	Wall thickness, in.	Moment of inertia, in.[4] I	Area of metal, sq. in. A	Radius of gyration, in. k	Radius of gyration, sq. in. k^2	Wall thickness, in.	Moment of inertia, in.[4] I	Area of metal, sq in. A	Radius of gyration, in. k	Radius of gyration, sq in. k^2
1	0.133	0.0873	0.494	0.4205	0.1769	0.179	0.1056	0.639	0.4066	0.1653
1¼	0.140	0.1947	0.6685	0.5397	0.2913	0.191	0.2418	0.8815	0.5237	0.2743
1½	0.145	0.3099	0.800	0.6226	0.3876	0.200	0.3912	1.068	0.6052	0.3663
2	0.154	0.6657	1.075	0.7871	0.6196	0.218	0.8679	1.477	0.7665	0.5875
2½	0.203	1.530	1.704	0.9474	0.8976	0.276	1.924	2.254	0.9241	0.8539
3	0.216	3.017	2.228	1.164	1.354	0.300	3.892	3.016	1.136	1.291
4	0.237	7.23	3.17	1.510	2.279	0.337	9.61	4.41	1.477	2.181
6	0.280	28.14	5.58	2.24	5.04	0.432	40.49	8.40	2.19	4.82
8	0.322	72.5	8.40	2.94	8.63	0.500	105.7	12.76	2.88	8.28
10	0.365	160.7	11.91	3.67	13.50	0.593	244.9	18.91	3.60	12.94
12	0.406	300.3	15.74	4.37	19.07	0.687	475.2	26.03	4.27	18.25
14	0.437	429.1	18.64	4.80	23.02	0.750	687.5	31.22	4.69	22.02
16	0.500	731.9	24.35	5.48	30.06	0.843	1,156.6	40.14	5.37	28.81
18	0.562	1,177.0	30.85	6.17	38.10	0.937	1,833.9	50.23	6.04	36.50
20	0.593	1,704.0	36.15	6.86	47.12	1.031	2,771.0	61.44	6.72	45.11
24	0.687	3,422.0	50.30	8.25	68.00	1.218	5,673.0	87.17	8.07	65.06

DEFINITIONS:
I = moment of inertia, in.[4], see also p. 858.
A = cross-sectional area of pipe metal sq in.
k = radius of gyration = $\sqrt{I/A}$.

ELEMENTS FOR TAKING UP EXPANSION

General.—There are four general types of elements used to take up the elongation (or contraction) due to thermal expansion:

1. Packless expansion joints.
2. Slip joints.
3. Swivel joints.
4. Inherent flexibility of the pipe itself, utilized through pipe bends, loops, right-angle turns, offsets in the line, etc.

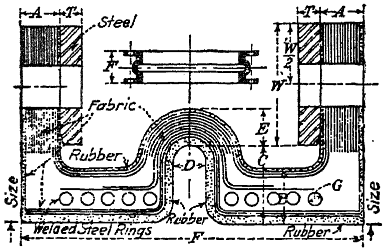


FIG. 1.—Reinforced rubber expansion joint.

Pipe size	Reinforced Rubber Expansion Joints																	
	A	B	C	D	E	F	G	W	T	Size	A	B	C	D	E	F	G	W	T
6	9/16	3/4	1 1/4	1/2	1/2	Face to Face 8" to 12"	1/4	1 1/2	1/4	26	1	1 1/2	2	1	3/4	Face to Face 8" to 12"	1/2	2 1/2	3/8
8	9/16	3/4	1 1/4	1/2	1/2		1/4	1 3/4	1/4	28	1	1 1/2	2	1	3/4		1/2	2 1/2	3/8
10	5/8	7/8	1 1/4	1/2	1/2		1/4	1 3/4	1/4	30	1	1 1/2	2	1	3/4		1/2	2 3/4	3/8
12	1 1/16	1	1 1/2	3/4	9/16		1/4	2	1/4	36	1 1/16	1 1/2	2	1	3/4		1/2	3 1/4	1/2
14	7/8	1 1/8	1 1/2	3/4	9/16		3/8	2 1/4	5/16	42	1 1/16	1 1/2	2 1/4	1	7/8		1/2	3 1/2	1/2
16	7/8	1 1/8	1 1/2	3/4	5/8		3/8	2 1/4	5/16	48	1 1/8	1 1/2	2 1/4	1	7/8		1/2	3 1/2	1/2
18	7/8	1 1/4	1 1/2	3/4	5/8		3/8	2 1/4	5/16	54	1 1/8	1 1/2	2 1/4	1	1		1/2	3 1/2	1/2
20	7/8	1 1/4	1 1/2	3/4	5/8		3/8	2 1/2	5/16	60	1 1/8	1 1/2	2 1/4	1	1		1/2	3 3/4	5/8
22	7/8	1 1/4	1 3/4	7/8	5/8		3/8	2 1/4	3/8	66	1 1/8	1 1/2	2 1/4	1	1		1/2	4	5/8
24	7/8	1 1/4	1 3/4	7/8	5/8		3/8	2 1/2	3/8	72	1 1/8	1 1/2	2 1/4	1	1		1/2	4	5/8

For use in vacuum and low internal pressure lines, free from oil, temp. under 180 F.

Several varieties of each of these types are in common use. The applicability of each type and variety depends on the amount of elongation expected and on the pressure and temperature conditions obtaining in the pipe. The more common varieties of these four general types of appliances are discussed in succeeding paragraphs and the particular field of each pointed out.

Packless Expansion Joints.—Under this classification come rubber expansion joints, corrugated copper, nickel steel, stainless steel, and other types of bellows expansion joints, etc.

Rubber expansion joints, such as that illustrated in Fig. 1, are sometimes used in vacuum and low-pressure lines where the temperature does not exceed 180 F. Such joints, made of ordinary rubber, should never be used where oil is present, because oil will rot the rubber. The traverse is usually from ½ to 1 in. Typical dimensions of such joints for various pipe sizes are given in connection with Fig. 1.

Copper expansion joints are made in a variety of forms, depending on working pressure, but should not be used in superheated steam lines.

Copper expansion joints for low vacuum or low pressure are commonly made in designs similar to those illustrated in Fig. 2, which is accompanied by a schedule of typical dimensions. The traverse of such joints ranges from ¼ to ½ in., or more, depending on the weight of copper and number of corrugations. Such joints may be used for saturated steam, water, air, or gases which do not attack copper.

Reinforced copper expansion joints are frequently used in moderate-pressure saturated-steam and water lines, and sometimes for air or gases which do not attack copper. A reinforced copper joint is shown in Fig. 3.

Expansion joints employing nickel steel or stainless-steel flexible elements are available in several forms for superheated steam service in bleeder lines from turbines, and similar high-temperature services. One type resembles the corrugated copper element of Fig. 3 while another is composed of a series of diaphragms welded at the inner and outer edges as shown in Fig. 4.

Slip Joints.—Slip joints are used very extensively in water and saturated-steam lines for pressures below 125 lb and occasionally for pressures up to or even above 250 lb. They are also used to a considerable extent in air, gas, and oil lines. Slip joints are made in a wide variety of styles to fit the structural requirements of almost any application for whose pressure and temperature requirements they are suited. For instance, a standard article can be obtained of either the single or the double type, flanged or screwed, with or without stops, with or without base or anchor pedestals, with or without service or drip connections, guided or unguided, etc. The principal disadvantage of a slip joint is that it depends

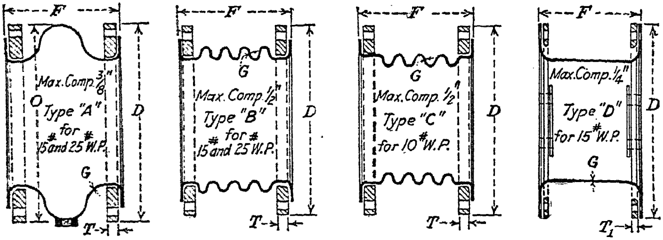


FIG. 2.—Coppper expansion joints for low pressure. Types *A*, *B*, and *C* flanges swivel; Type *D* flanges do not swivel.

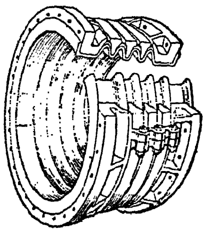


FIG. 3.—Reinforced copper expansion joint for moderate pressures.

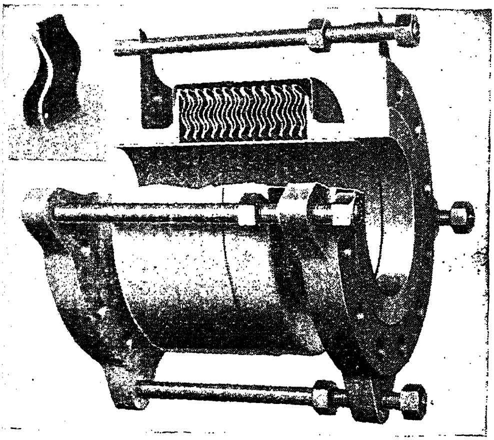


FIG. 4.—Steel expansion joint for high-temperature services.

TYPICAL DIMENSIONS OF COPPER EXPANSION JOINTS SHOWN IN FIG. 2

Pipe size	Flange							O. S. Diam.	Gage "G" Copper						Drain diam.	*Drilling			Pipe size
	Diam.	Thickness			Face—Face				Type							†Bolt 1/8" smaller			
					Type				A	A	B	B	C	D		Bolt			
		C. I.	Steel		A & B	C	D		Normal Working Pres.										
	D	T	T	T_1	F	F	F	O	15	25	15	25	10	15		No.	†Hole	Circ.	
6	11	1	1	3/8	9	8 5/8	6	11	16	15	16	16	16	14	1/2	8	7/8	9 1/2	6
8	13 1/2	1 1/8	1 1/8	1/2	10	8 5/8	6	13 1/2	16	15	16	16	16	13	3/4	8	7/8	11 3/4	8
10	16	1 3/16	1 1/8	1/2	11	8 5/8	6	16	16	14	16	15	16	13	3/4	12	1	14 1/4	10
12	19	1 1/4	1 1/4	1/2	11	8 5/8	6	19	15	13	16	15	16	13	3/4	12	1	17	12
14	21	1 3/8	1 3/8	1/2	12	10 1/2	6	21	15	13	16	15	16	12	3/4	12	1 1/8	18 3/4	14
16	23 1/2	1 7/16	1 3/8	1/2	12	10 1/2	6	23 1/2	15	12	15	14	16	11	1	16	1 1/8	21 1/4	16
18	25	1 9/16	1 1/2	5/8	13	12 3/4	6 1/2	25	14	12	15	13	16	11	1	16	1 1/4	22 3/4	18
20	27 1/2	1 11/16	1 1/2	5/8	13	12 3/4	6 1/2	27 1/2	14	12	15	13	16	11	1 1/4	20	1	25	20
24	32	1 7/8	1 1/2	3/4	14	12 3/4	7	32	14	11	14	12	16	10	1 1/4	20	1	29 1/2	24
30	38 3/4	2 1/8	1 3/4	7/8	16	17	8 1/2	38 3/4	13	8	13	10	16	9	1 1/2	28	1 1/8	36	30
36	46	2 3/8	1 3/4	1	17	17	8 1/2	45 3/4	11	7	12	8	15	8	1 1/2	32	1 1/8	42 3/4	36
42	53	2 5/8	1 7/8	1 1/8	19	17	9	52 3/4	10	6	11	7	14	7	2	36	1 1/4	49 1/2	42
48	59 1/2	2 3/4	1 7/8	1 1/4	20	17	9	59 1/4	9	6	10	6	13	6	2	44	1 1/4	56	48
54	66 1/4	3	2 1/8	1 3/8	22	17	9 1/2	66	8	5	9	6	12	5	2	44	1 1/4	62 3/4	54
60	73	3 1/8	2 1/8	1 1/2	23	17	10 1/2	72 3/4	7	4	8	5	12	4	2 1/2	52	1 3/8	69 1/4	60
72	86 1/2	3 1/2	2 1/8	1 3/4	26	17	11	86 1/4	6	2	7	3	10	2	3	60	1 3/8	82 1/2	72

* Flange drilling 20 in. and larger are shown drilled to the 25-lb low-pressure standard.
† Bolt holes are drilled 1/8 in. oversize.

on packing and stuffing boxes for fluid tightness. Consequently, it is subject to more or less maintenance and to leak on occasion. For this reason they should always be placed where readily accessible for inspection and maintenance, and, in the case of underground piping, manholes should be provided for this purpose. The necessity for packing imposes a limitation on their use with superheated steam to a temperature which the packing can stand for a reasonable length of time. Improved designs of slip joints are available for moderate superheats in which the packing is partially protected by an air-cooled hollow sleeve. The bodies of slip joints are commonly made of brass or cast iron, while the sleeve which traverses the packing is of necessity made of brass, chromium-plated steel, or other noncorrosive metal which will retain a smooth

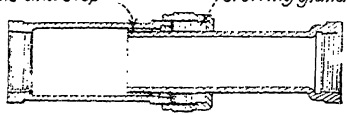


FIG. 5.—All-brass slip joint.

surface. When slip joints are used in oil lines, the sleeve is frequently made of steel. Several types of slip joints are illustrated in Figs. 5 and 6.

Slip joints possess certain inherent advantages for taking up expansion in that (1) the amount of elongation for which they are suitable is entirely definite and ascertainable without recourse to extensive calculation; (2) they require little space and no change in direction of the pipe line as is required with swivel joints, bends, offsets, etc.

The amount of elongation which can be taken up by a single slip joint is termed its "traverse." In other words, the traverse is the free movement of the sleeve from its extreme "out" to its extreme "in" position. The usual traverse of slip joints varies from about 2 to 12 in. per end, depending on the nominal pipe size and the design of the joint. The selection of a slip joint having suitable traverse should be made by reference to a manufacturer's catalogue. Where the expected elongation exceeds the traverse of a single joint, a double joint may be employed, or two or more single joints located at appropriate points. Where several joints are used, the pipe should be anchored midway between joints, unless the joint itself has an anchor base, in which case no other

anchors are required. In the case of a single-expansion joint, it should be placed at one end of the line if it has an anchor base (or is independently anchored); and, if unanchored, it should be placed in the middle of the line and the line should be anchored at both ends.

Where the temperature changes are small, as in buried gas, oil, and water lines, bell joints, and compression sleeve couplings perform the function of slip joints (see Couplings, page 1199).

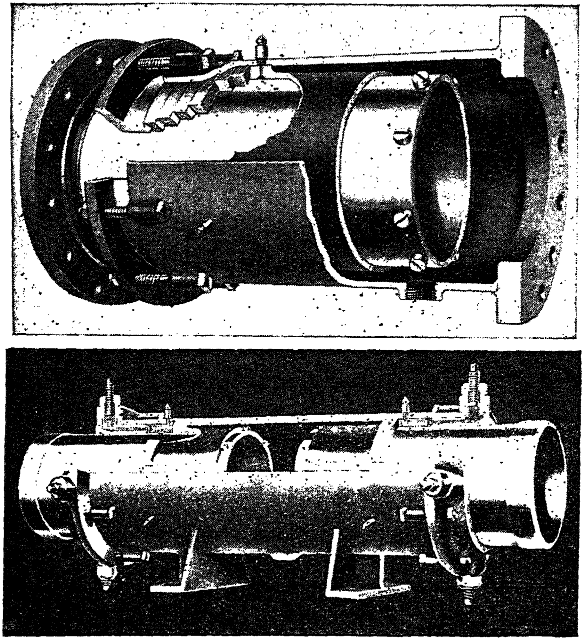


FIG. 6.—Typical slip expansion joints.

Swivel Joints.—Swivel joints are used extensively in steam and hot-water heating systems and, to some degree, in other applications. In very long horizontal heating mains, in which the movement would be too great to be absorbed by the branch connections, the elongation may be taken up by anchoring the pipe at two or more points and providing a swivel joint of the form shown in

Fig. 7*a*. In this arrangement, the crosswise piece of pipe is able to swing in a horizontal plane through swivel action in the threads of the short vertical pieces. In many cases the straight pipe in such offsets is sufficiently flexible to take up the expansion without developing enough thrust to produce swiveling in the threaded joints. An incidental disadvantage in the use of swivel joints is that they almost invariably introduce additional pressure drop through the extra ells required in their construction. On the other

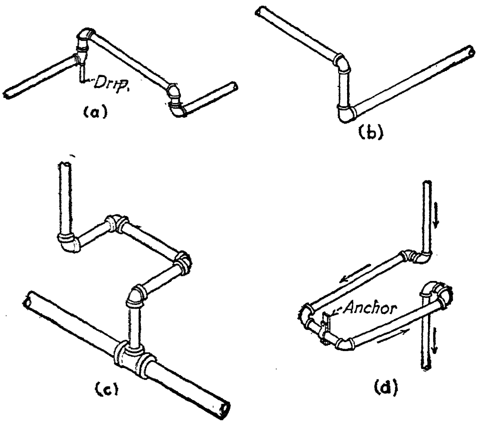


Fig. 7.—(*a*) Expansion swivel; (*b*) swivel turn; (*c*) flexible connection for riser; (*d*) expansion loop for riser.

hand, they are inexpensive and can be made up readily on the job from standard fittings and pipe.

Another scheme which is sometimes used where a main makes a 90-deg turn is shown in Fig. 7*b*. With this arrangement the expansion is largely absorbed by the spring of the members, although stress is relieved to a considerable extent by swivel action of the short vertical piece.

In heating risers where the expansive movement is too great to be taken up by the spring of branch connections, the types of swivel joints shown in Figs. 7*c* and 7*d* are sometimes used. A swivel loop, such as that shown in Fig. 7*d*, is easily capable of handling a length of riser of at least four stories in either direction and gives perfect flexibility. Space is required between joists to conceal the loop.

The swivel joints described above, as well as a large variety of similar types, are applied to many other services besides heating systems when space permits or the extra pressure drop involved is not objectionable. The amount of elongation that a swivel joint can take up is controlled by the length of swing piece employed and the lateral displacement which is permissible in the long pipe runs. These factors can be determined readily in each instance by the designer.

Inherent Flexibility of the Pipe.—In high-pressure, high-temperature piping it is the general practice to take care of thermal expansion through the elastic properties of the steel pipe itself. This can be accomplished frequently by planning the piping layout to take advantage of directional changes which, in themselves, will provide the necessary flexibility. Properly proportioned runs of pipe at right angles to one another in either a horizontal or vertical plane or in both are the usual means of accomplishing this end. Such runs of straight pipe may be joined together with either elbows or 90-deg pipe bends, as best suits the designer.

A good example of the use of directional changes to obtain required flexibility is found frequently in the main steam piping from boilers to turbines where two or more long runs of straight pipe are installed at right angles to one another without other elements for absorbing expansion.

In instances where it is impracticable to install the piping to obtain the required flexibility by directional changes, as described above, it may be advantageous to insert expansion bends in the line. Such expansion bends may be (1) a complete unit, such as the expansion U bend, the double-offset expansion U bend, or the circle bend (Fig. 8) applied as illustrated in Figs. 41 and 42; (2) a built-up bend composed of several of the smaller curved pieces illustrated in Fig. 8; (3) a combination of straight pieces with such curved pieces or cast ells; (4) a so-called "square" bend made up entirely of straight pieces and 90-deg. elbows (Fig. 40). Where short branch connections are taken off at frequent intervals from a long high-pressure line subject to considerable expansion, it may be necessary to install some type of expansion bend at intermediate points between the branch connections and suitably anchor the line to prevent undue strain on the branches.

In other instances, the layout of the line may be such that the use of expansion bends is preferable to directional changes of the entire line. For one reason or another, the use of various types

of expansion bends is quite general, and the practice governing their manufacture is well established. Unfortunately, a determination of the amount of elongation which can be absorbed safely by any particular bend involves rather difficult calculation. Methods of determining stresses and reactions due to thermal expansion are given in the latter part of this chapter. The elongation which can be absorbed by two common types of expansion bends, *viz.*, an expansion U bend and a double-offset expansion U bend when placed in a straight run of pipe with fixed anchors at the ends may be found from Tables VI to IX or Figs. 41 and 42.

The stresses and reactions obtaining with a "square" bend (made up with 90-deg arcs and tangents) when inserted in a run of straight pipe of varying length may be found from Fig. 40, page 843. Similar information for a 90-deg bend with tangents may be found from Fig. 43, page 847.

Pipe Bends.—The following descriptions of methods used to bend pipe indicate that bends can be made in a number of ways. The choice of method depends to a considerable extent on the facilities available. Practical methods for bending pipe and tubes of copper, brass, and related alloys are covered in the "Pipe and Tube Bending Handbook."[1] Much of the information given therein is applicable to steel pipe as well.

Cold Bending.—Pipe up to approximately 6 in. in size can be bent cold in a press or hydraulic bending machine. Smaller sizes can be bent on a roll bender of the sort used for bending boiler tubes. A method of cold-bending line pipe in the field by means of a series of forming shoes and a tractor also has been developed for cross-country gas and oil lines.

Bending by Local Heating.—Pipe may be bent by heating successive sections of the pipe with large torches or gas burners. In one such procedure the pipe is filled with sand and mounted on a bending slab preparatory to heating. The free end is pulled around a few degrees after each heating. The size and wall thickness of pipe that can be bent in this manner are limited by the capacity of the burners and the pulling equipment. Eight-inch, Schedule 80, Grade *B*, pipe has been bent successfully in this manner. A variation of the above procedure has been used in which the pipe is progressively heated by internal burners. In this case, of course, no sand is used.

[1] Published by the Copper and Brass Research Association, 420 Lexington Ave., New York 17, N.Y.

Furnace Bending.—In usual bending procedure the pipe is completely filled with sand which is packed tight by jarring with sledges or a mechanical vibrating device. The portion of pipe to be bent is placed in a gas- or oil-fired forging furnace and heated to a temperature of 1700 to 2000 F. It is then transferred to a bending table or slab and bent as quickly as possible to follow the contour of a template rod. During this bending operation water usually is applied where needed to make bending take place in the exact section desired. Because of the heat stored in the sand, the bend normally regains sufficient temperature to overcome the effect of the local water quenching. Subsequent annealing or normalizing operations sometimes are considered desirable, especially with alloy pipe. A variation of this procedure in which pipe is heated only to around 1400 F has been developed to avoid the need for subsequent normalizing or annealing.

Crease Bending.—In crease bending the pipe is filled with sand, but not packed as hard as when making smooth bends. One end of the pipe is clamped firmly to a bending table and a heating unit, consisting of a gas burner with numerous jets, is used to heat a short section on the inner side of the proposed bend to about 1800 F, the length heated depending on the size and wall thickness of the pipe. When properly heated, force is applied to the loose end of the pipe to pull it around the desired number of degrees, which is determined by dividing the total number of degrees in the bend by the proposed number of creases. The heated section bulges out to form the desired crease which is then allowed to cool while moving the heating element along the pipe to the center of the next crease. Water is used to restrict bending to the exact location desired. Subsequent annealing or normalizing operations customarily are performed on creased bends.

Wrinkle Bending.—A variation of the crease bending procedure adapted to bending line pipe in the field is known as "wrinkle" bending. No sand is used in this operation and heating is done with large gas- or oil-fired torches. The wrinkles are not so pronounced as the creases and the radius of a wrinkle bend usually is greater than in the case of a creased bend.

Corrugating Pipe.—In one procedure the pipe is slowly rotated in a horizontal press resembling a lathe while the section to have the first convolution is heated to about 1950 F. No sand is used in corrugating pipe. A hydraulic ram applied to what corresponds to the tailstock of the lathe is advanced a predetermined distance

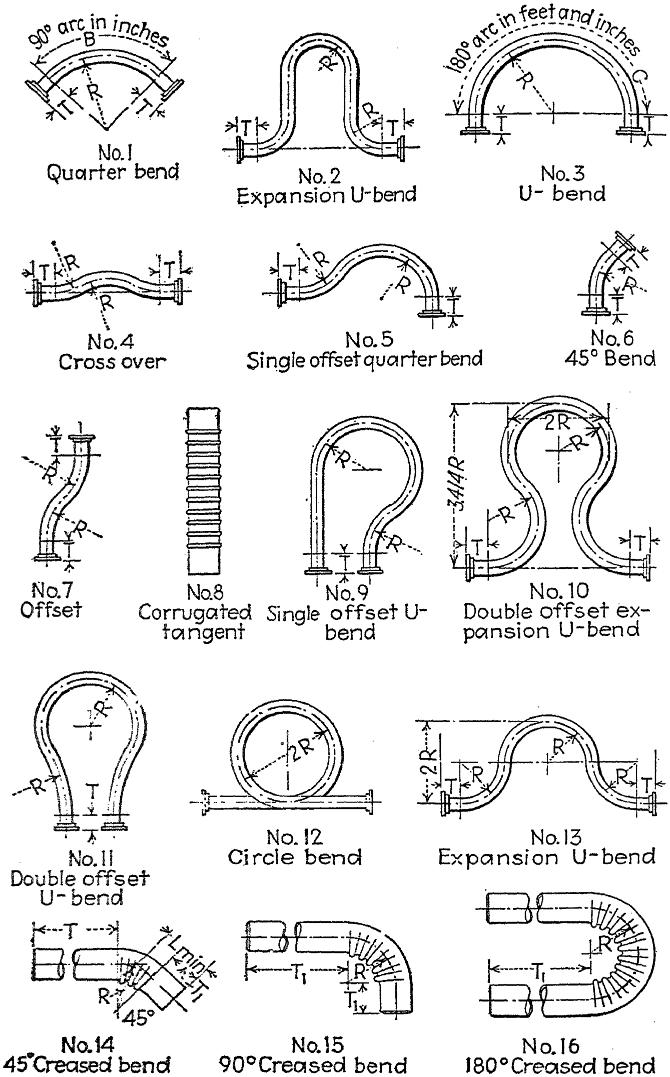


FIG. 8.—Common types of pipe bends and special corrugated tangent. (For dimensions, see pages 775–776.)

Dimensions of Pipe Bends Shown in Fig. 8

Pipe size	Minimum tangent ends			Radii					
				Recommended		Minimum		* Extreme minimum	
	Sc'd	Weld	Lap	Rad.	Arc.	Rad.	Arc.	Rad.	Arc.
	T	T	T	R	90	R	90	R	90
1	2¼	2¼		7	0′11″	2¼	0′ 3 9/16	2″	0′ 3⅛
1¼	2½	2½		8	1′ 0 9/16	2¾	0′ 4⅜	2½	0′ 3 15/16
1½	2¾	2¾		10	1′ 3 11/16	3¼	0′ 5⅛	3	0′ 4¾
2	3	3		1′0	1′ 6⅞	4½	0′ 7 1/16	4″	0′ 6¼
2½	4	4		1′2	1′10″	6	0′ 9 7/16	5″	0′ 7⅞
3	4	5	6	1′6	2′ 4¼	7	0′11	6	0′ 9 7/16
3½	5	5	6	1′8	2′ 7⅝	9	1′ 2⅛	8	1′ 0 9/16
4	5	5	6	2′0	3′ 1 11/16	1′0	1′ 6⅞	10	1′ 3 11/16
5	6	5	7	2′6	3′11⅛	1′6	2′ 4¼	1′3	1′11 9/16
6	7	6	7	3′0	4′ 8½	2′0	3′ 1 11/16	1′6	2′ 4¼
8	9	6	8	4′6	7′ 0 13/16	2′8	4′ 2¼	2′0	3′ 1 11/16
10	12	7	10	5′6	8′ 7 11/16	4′0	6′ 3⅜	2′6	3′11⅛
12	14	7	10	7′0	11′ 0	5′0	7′10¼	3′0	4′ 8½
14	16	7	14	8′0	12′ 6⅞	6′0	9′ 5⅛	4′0	6′ 3⅜
16	18	8	16	10′0	15′ 8½	7′0	11′ 0	5′6	8′ 7 11/16
18	18	8	18	11′0	17′ 3⅜	8′0	12′ 6⅞	6′6	10′ 2½
20	18	8	18	12′0	18′10¼	9′0	15′ 1⅝	8′0	12′ 6¾
24	18	9	20	15′0	23′ 6¾				

Full dimension sketch or blueprint should accompany all inquiries or orders for bends.
Drawings submitted should include dimensions R and T and center-to-end dimensions.
For [illegible] No. 7 [illegible] see page 2.
* When "Extreme minimum" radius bends are required, pipe should be Schedule 80 (extra strong) or heavier. "Recommended" and "Minimum" radii bends can be made from Schedule 40 (standard weight) pipe.

Dimensions of Creased Bends in Fig. 8

Nominal size, inches	Radius, inches, *R*	Minimum center to end, inches 90 deg	Minimum tangent, inches, *T*[1]	Minimum center to end, inches, 45 deg
2	3	5	2	3¼
2½	3¾	6¼	2½	4 1/16
3	4½	7½	3	4⅞
3½	5¼	8¾	3½	5 11/16
4	6	10	4	6½
5	7½	12½	5	8⅛
6	9	15	6	9¾
8	12	20	8	13
10	15	25	10	16 3/16
12	18	30	12	19 7/16
14 O.D.	21	35	14	22 11/16
16 O.D.	24	40	16	25 15/16

Note.—Creased bends can be made from pipe of all commercial pipe thicknesses.
[1] The maximum tangent is dependent upon length of pipe available.

Length of Arc for Pipe Bends Shown in Fig. 8

Degree and minute constants for arc lengths, radius being unity

Degree	Constant	Degree	Constant	Degree	Constant	Minute	Constant
1	.0175	41	.7156	81	1.4137	21	.00610
2	.0349	42	.7330	82	1.4312	22	.00639
3	.0524	43	.7505	83	1.4486	23	.00669
4	.0698	44	.7679	84	1.4661	24	.00698
5	.0873	45	.7854	85	1.4835	25	.00727
6	.1047	46	.8029	86	1.5010	26	.00756
7	.1222	47	.8203	87	1.5184	27	.00785
8	.1396	48	.8378	88	1.5359	28	.00814
9	.1571	49	.8552	89	1.5533	29	.00843
10	.1745	50	.8727	90	1.5708	30	.00872
11	.1920	51	.8901			31	.00901
12	.2094	52	.9076			32	.00930
13	.2269	53	.9250	Constants for each Minute		33	.00959
14	.2443	54	.9425			34	.00989
15	.2618	55	.9599			35	.01018
16	.2793	56	.9774			36	.01047
17	.2967	57	.9948			37	.01076
18	.3142	58	1.0123			38	.01105
19	.3316	59	1.0297	**Min.**	**Constant**	39	.01134
20	.3491	60	1.0472			40	.01163
21	.3665	61	1.0647	1	.00029	41	.01192
22	.3840	62	1.0821	2	.00058	42	.01221
23	.4014	63	1.0996	3	.00087	43	.01250
24	.4189	64	1.1170	4	.00116	44	.01279
25	.4363	65	1.1345	5	.00145	45	.01309
26	.4538	66	1.1519	6	.00174	46	.01338
27	.4712	67	1.1694	7	.00203	47	.01367
28	.4887	68	1.1868	8	.00233	48	.01396
29	.5061	69	1.2043	9	.00261	49	.01425
30	.5236	70	1.2217	10	.00290	50	.01454
31	.5411	71	1.2392	11	.00319	51	.01483
32	.5585	72	1.2566	12	.00349	52	.01512
33	.5760	73	1.2741	13	.00378	53	.01541
34	.5934	74	1.2915	14	.00407	54	.01570
35	.6109	75	1.3090	15	.00436	55	.01599
36	.6283	76	1.3265	16	.00465	56	.01628
37	.6458	77	1.3439	17	.00494	57	.01658
38	.6632	78	1.3614	18	.00523	58	.01687
39	.6807	79	1.3788	19	.00552	59	.01716
40	.6981	80	1.3963	20	.00581	60	.01754

Example:
To obtain length of arc—32° 47′, radius—6′ 4½″

Solution:
Constant—32° = 0.5585
47′ = 0.01367
0.57217 × 6.375 = 3.64758 = 3′ 7¾″

thus forcing the heated section out to make one convolution. A water curtain is used to control the heating to the section desired. The procedure is repeated until the desired number of convolutions are obtained. Corrugated fillers and bends made therefrom customarily are given a final anneal or normalize to remove the

effects of the local heating and quenching operations. Corrugated bends are bent in a separate operation on the bending slab using straight pipe which has been corrugated as described above.

A number of the common forms of curved pipe which are used in building up expansion bends are illustrated in Fig. 8, along with certain bends which are complete units in themselves. A schedule of the minimum dimensions to which such bends can be fabricated, along with tabulation of the arc lengths corresponding to the included angle for a condition of unity radius, is given in connection with the figure. The latter is of value in estimating the length of pipe required to make any particular bend. Although the schedule of minimum dimensions is not standard and varies somewhat with different piping fabricators, it agrees very closely with general American practice. Instructions are given in the figure and table as to what dimensions should be furnished in ordering bends. A method for determining the proportions of offset bends, such as No. 7 in Fig. 8, is given in connection with Fig. 4 on page 2.

It should be noted that bends can be formed to a smaller radius without buckling when thick-walled pipe is used. For this reason, especially with large pipe, it is necessary to use pipe at least as heavy as Schedule 80 if the bend is to be formed to a radius of much less than 5 pipe diameters. Schedule 40 pipe can be bent to the "recommended" and "minimum" radii given in Fig. 8.

Where bends of shorter radius are desired, resort may be made to welding elbows or to creased bends. Dimensions of creased 45-, 90-, and 180-deg bends are given in connection with Fig. 8. In some cases the use of corrugated tangents, shown in Fig. 8, permits great reductions in end thrusts and bending moments. For information as to the most effective use of creased bends and corrugated tangents, reference should be made to the "Pittsburgh Piping Design Manual," by E. A. Wert, S. Smith, and others.[1]

The above discussion applies to steel and wrought-iron pipe. In cases where temperatures and pressures are not too severe, sections of copper pipe are sometimes used to relieve expansion strains, in the form of either bends or straight pieces placed at right angles to the direction of expansion. Although the use of copper pipe for this purpose is common in marine work, it is not so general in land practice and the matter of commercial shapes is not so well established as in the case of steel pipe.

[1] Published by Pittsburgh Piping and Equipment Company, Pittsburgh, Pa.

Cold Spring.—It is common practice to compute the expansive movement from the cold to the hot operating condition and make all runs of piping somewhat shorter than the measured cold dimensions. Cutting the pipe short by an amount equal to approximately one-half the calculated expansion and springing the pipe into place are found to facilitate assembly and give an additional expansion precaution.

The Code for Pressure Piping permits a maximum credit of one-third the computed expansion where the pipe is cut short by 50 per cent or more of the computed expansion and sprung into position. The full amount of the cold spring must, of course,

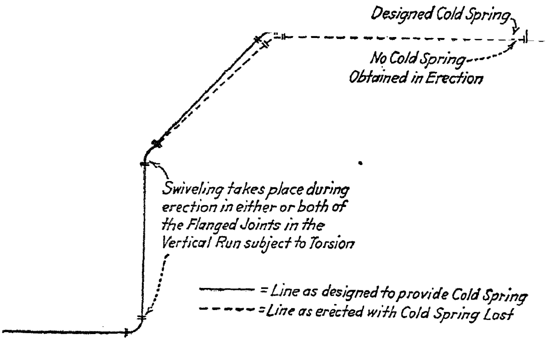


FIG. 9.—Loss of designed cold spring in erection, isometric view.

be taken into account in considering forces, moments, and stresses acting in the cold condition. The chief reason full credit in the hot condition is not allowed for cold spring is because some or all of it may be lost in the course of erection. One of the ways by which designed cold spring may be lost in erection is shown in Fig. 9.

RADIAL OR CIRCUMFERENTIAL EXPANSION OF PIPES

Coefficient of Expansion.—The increase (or decrease) in diameter or circumference of a pipe with change in temperature is computed in the same manner as explained on page 754 for elongation along the pipe axis. The same coefficient of linear expansion is used, but in this case the length over which expansion is computed is the mean diameter or mean circumference of the pipe, *i.e.*, mean value between inside and outside dimensions. In most cases the

increases in diameter and circumference are too small to be of any practical significance and they are seldom considered.

Stresses.—If the inside and outside of a pipe are not at approximately the same temperature, as in the case of a boiler tube or an uncovered steam line in open air, there will be unequal expansion in the inner and outer layers of the pipe, and appreciable fiber stresses will result.

Formulas for computing stresses induced in water tubes of modern boilers are given in a paper by George A. Orrok, on "High-pressure Steam Boilers," *Trans. ASME*, SP-1928. The computation of temperature stresses under conditions such that creep or plastic flow is an important factor was described by R. W. Bailey in an article on "The Design of Plant for High-temperature Service," *Engineering* (London), July 8, 1927, p. 44.

ELASTIC PROPERTIES OF STRAIGHT PIPE AND BENDS[1]

By Sabin Crocker and Arthur McCutchan

BASIS OF SOLUTION

General.—The stresses and reactions resulting from the thermal expansion of piping have engaged the attention of many engineers and mathematicians in recent years, with the result that certain rather definite principles and conclusions have been established.[2] These principles and conclusions may be summarized as follows:

1. *Unit for Study.*—The proper unit to consider in calculating the flexibility of a pipe line is a section of piping between two fixed points or anchors.

[1] The section on "Elastic Properties of Straight Pipe and Bends" was originally prepared by the engineering staff of The Detroit Edison Company for the company's own use, and is published here with the company's express permission. Figures 31 to 37, inclusive, and part of Fig. 35 are reproduced from the paper by A. M. Wahl[2c].

[2] The following references have been selected as furnishing the best statement of the generally accepted analyses of stresses and reactions due to thermal expansion in piping. Reference (*a*) contains a bibliography of the more important articles written prior to its publication, and references (*c*) and (*e*) refer to a number of other articles dealing with flattening of the circular cross section of a pipe during flexure.

(*a*). "The Elasticity of Pipe Bends," by Sabin Crocker and S. S. Sanford, *Trans. ASME*, 1922, p. 547.

(*b*). "The Elastic Deformation of Pipe Bends," by William Hovgaard, *J. Math. Phys., Mass. Inst. Tech.*, Vol. 6, No. 2, 1926; and Vol. 7, Nos. 3 and 4, 1928; Vol. 8, No. 4, 1929. See also "Stresses in Three Dimensional Pipe Bends,"

2. *End Conditions.*—The ends of a section of piping should be assumed as rigidly fixed at the anchor points, *i.e.*, the ends are entirely prevented from either angular rotation or linear displacement. This assumption most closely approximates the end conditions of an actual pipe line and is on the side of safe design.

3. *Shape.*—The true shape of the piping should be used in determining its flexibility as far as practicable. Piping composed of arcs of circles should be considered as such rather than assumed as represented by straight lengths and square bends. Where the curvature of the arcs is such that the "rigidity multiplication factor" K is approximately 0.6 (see page 800), the arcs may be treated as square bends without introducing serious error.

4. *Flattening of the Cross Section.*—Flattening of the circular cross section of the curved portions of a pipe line during bending, caused by a moment acting in the plane of the bend, has a marked effect upon the reactions and stresses in the line and should be taken into account.

5. *Reduction of Modulus of Elasticity with Increase in Temperature.*—The modulus of linear elasticity E and shearing elasticity G decrease with increase in temperature. The value of the modulus of elasticity for the piping material at its working tem-

Trans. ASME, Vol. 57, 1935, FSP 57-12; and "Further Studies of Three Dimensional Pipe Bends," *Trans. ASME*, Vol. 59, 1937, FSP 59-13.

(*c*). "Stresses and Reactions in Expansion Pipe Bends," by A. M. Wahl, *Trans. ASME*, Vol. 50, No. 15, p. 241, 1928, FSP 50-49.

(*d*). "The Flexibility of Plain Pipes," by J. R. Finniecome, *The Engineer* (London), Aug. 17, 1928, and following issues.

(*e*). "Design of Steam Piping to Care for Expansion," by W. H. Shipman, *Trans. ASME*, Vol. 51, No. 22, p. 415, 1929, FSP 51-52.

(*f*). "Frictional Resistance and Flexibility of Seamless-Tube Fittings Used in Pipe Welding," by Sabin Crocker and Arthur McCutchan, *Trans. ASME*, Vol. 53, 1931, FSP 53-17.

(*g*). "Load-Deflection Relations for Large, Plain, Corrugated, and Creased Pipe Bends," by E. T. Cope and E. A. Wert, *Trans. ASME*, Vol. 54, 1932, FSP 54-12.

(*h*). "End Reactions and Stresses in Three Dimensional Pipe Lines," by G. B. Karelitz and J. H. Marchant, *Trans. ASME*, June, 1937, A68-A64.

(*i*). "Elastic Properties of Curved Tubes," by Irwin Vigness, *Trans. ASME*, Vol. 65, No. 2, p. 105, 1943.

(*j*). "Bending of Curved Thin Tubes," by Leon Beskin, *Trans. ASME*, *Paper* No. 44-A8, *J. Applied Mechanics*, Vol. 12, No. 1, pp. A1-A7, March, 1945.

(*k*). "Moment Distribution Analysis for Three Dimensional Pipe Structures," by R. C. DeHart, *Trans. ASME*, *J. Applied Mechanics*, Vol. 11, No. 4, p. A240, 1944.

perature should be used in determining the reactions due to thermal expansion (see Fig. 39 on page 830).

The solution of symmetrical shapes lying in one plane can be accomplished readily, as only elementary calculus and the theory of flexure of cantilever beams are required. The application of the principles of analytic solution to simple bends, such as those illustrated in Fig. 8, made up of circular arcs and tangents, or to straight pipe and ells lying in one plane, offers no serious difficulty and has been generally employed by those studying the elastic properties of piping. The principles of *superposition* described on page 782 have been used by several investigators as a means of combining the flexural properties of the various component parts of a line to obtain the elastic properties of an entire section between two anchors. It is unfortunate that some of the published articles have not been sufficiently explicit regarding superposition to enable the uninitiated to make full use of it. The present authors have endeavored to remedy this deficiency and at the same time to approach the whole problem of the elastic properties of piping by a method which is not limited to lines in one plane but can be applied successfully to those in space. Owing to the complex nature of the problem when applied to lines in space, the authors prefer to use a grapho-analytical method based on area-moment diagrams rather than a straight analytical solution. The grapho-analytical method has been used consistently throughout for simple piping in one plane as well as for lines in space to avoid confusion resulting from introducing more than one method.

Before taking up the principles of superposition and grapho-analytical methods of studying the flexibility of piping, it is necessary to call attention to the fact that a curved pipe behaves differently during flexure than does a straight pipe or a solid curved bar.[1] A curved pipe having a comparatively thin wall and formed to a short radius tends to flatten during flexure instead of having its cross section remain circular, as is the case with a straight pipe or a solid curved bar. The amount of such flattening in a curved pipe is determined by the diameter of the pipe, its wall thickness, and the radius to which it is bent, as explained on page 816. Flattening of the circular cross section is attended by a reduction in the resistance to flexure, equivalent to a reduction in the moment of inertia of the pipe, and by a change in stress distribution in the pipe wall. This reduction in the resistance that a curved pipe

[1] See footnote 2(*b*), 2(*c*), and 2(*i*) listed on p. 779.

offers to flexure is taken into account by multiplying its moment of inertia by a factor K, less than unity, which is termed the "rigidity multiplication factor." By introducing this factor K in the proper places, it is possible to compute the flexibility of piping made up of curved pieces, or combinations of curved and straight pieces, and corrugated straight pieces with the same exactitude as is obtained where the pipe is all straight. The application of K to problems of this sort is accomplished readily in the grapho-analytical method described in succeeding pages.

Superposition.—Piping-flexibility problems can be simplified by restricting the length of line dealt with at one time. The first step in this process is to limit the section under analysis to the length between two anchor points. The next step is to divide this section between anchors into its component parts, such as straight lengths, quarter bends, etc. The component parts are then analyzed as separate cantilevers, *i.e.*, one end is considered as fixed with certain forces and moments applied to the other through the structure of adjacent piping. Straight lengths are identical with the simple cantilevers encountered in ordinary beam analysis. Curved lengths, arcs of circles, etc., are handled as special types of cantilevers, as explained in connection with the elementary cases given on pages 787 to 793, inclusive.

The idea of superposition can be condensed into a few words, as follows: Superposition is a means of determining the elastic properties of a section of piping by first transposing the thrusts and bending moments existing at an anchor point so as to apply them to each component part of the section of piping and then making an algebraic summation of (1) the deflections of the component parts produced by such transposed forces and moments and (2) the deflections of the component parts due to the cumulative rotations of the elements between each component part and the anchored end of the section of piping.

The manner in which forces and moments are transposed and deflections combined is explained most readily through an example: Consider a section of piping arranged as in Fig. 10a with a transverse force F acting at the free end. The component parts of this section of piping are shown in Fig. 10b with the transposed forces and moments acting upon them. The hypothetical members, to which the component parts l_2 and l_3 are attached (see Fig. 10b), are shown rotated through angles ψ_b and ψ_c, respectively. These rotations are equivalent to the total rotation of the elements

between each component part and the anchored end of the section of piping. The rotations are shown greatly exaggerated. The transposed forces are shown rotated through angles equal to the rotation of the component parts. These rotations apparently give components at right angles to the original direction of the force at this end but are actually so small that any components at right angles to the original direction of the force are negligible. The component part l_3 is a straight cantilever with a transverse force F at its free end. l_2 is a straight cantilever with a

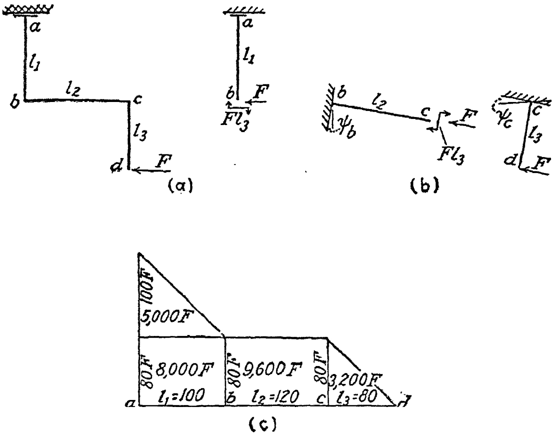


FIG. 10.—Section of piping acted on by a single force at the end in the plane of the section.

moment Fl_3 at its free end. The direct force F is simply transmitted through l_2 to length l_1. Part l_1 is a cantilever with a moment Fl_3 and a transverse force F at its free end. In addition to the deflections at the free ends of these cantilevers due to bending within themselves, rotations of the elements between the hypothetical fixed ends of the cantilevers and the anchored end of the section of piping represented by ψ_b and ψ_c give deflections at the ends of the cantilevers which are added algebraically to the individual deflections of the component parts.

Grapho-analytical Method.—The following grapho-analytical method[1] facilitates the combination of the deflections due to

[1] So far as the authors are aware, the first application of area moments to the solution of such problems was made by Paul E. Todd while employed by the Drafting and Surveying Bureau of The Detroit Edison Company.

bending within each part with those due to rotation of the elements between each part and the anchored end of the section of piping: A bending-moment diagram is first constructed (see Fig. 10*c*). The lengths of the component parts of the pipe line are laid off consecutively on a horizontal base line. (In this case the true lengths are used. Where torsion, changes in moments of inertia, or other modifying factors affect the flexibility of the section of piping, virtual lengths must be used as explained on page 793.) The bending moment acting on each component part, corresponding to the transposed forces and moments shown in Fig. 10*b*, is drawn approximately to scale. In order to avoid complications as to the sign of the bending-moment areas, some definite convention should be adopted. In this treatise, the bending-moment diagram for the component part on which the force at the free end of the section of piping first acts is consistently drawn above the horizontal base line. The direction of rotation of the resulting moments is then apparent from inspection of the free-body diagrams (see Fig. 10*b*).

The deflection at the free end of each component part is the algebraic sum of two deflections, each of which bears a definite relation to the bending-moment areas. (1) The first portion of the deflection of the free end of any component part results from the rotations of all the preceding parts between the hypothetically fixed end and the pipe anchorage. This portion of the deflection is equal to the length of the part in question multiplied by the algebraic sum of all the areas of the bending-moment diagrams preceding the fixed end of the part, divided by EI. (2) The second portion of the deflection of the free end of the component part is due to the forces and moments which act on the part itself. This part of the deflection is caused by the successive rotations of the elementary lengths comprising the component part. The deflection is given by the summation of the products of the angular rotation of each element and its distance to the free end. This rotation of each element is found by dividing the area of the diagram for the bending moment which acts on the element by EI. The distance between each element and the free end is expressed in terms of the length of the component part if it is straight, or in terms of the radius if a quarter bend. These deflections have been determined for straight cantilevers and quarter bends and are expressed in terms of the rotations in Cases I and X on pages 787 to 793, inclusive.

The symbols used in this section are

E = modulus of linear elasticity of the pipe material at the operating temperature, psi (Fig. 39, page 830).

G = modulus of shearing elasticity at the operating temperature = $E/2(1 + \lambda)$, psi (Fig. 39, page 830).

λ = Poisson's ratio or coefficient of lateral contraction (pages 32–40, 296, 344, and 830).

I = moment of inertia of pipe cross section in.^{4} (Table X, page 858).

J = polar moment of inertia of pipe cross section = $2I$ in.^{4}

Δ = deflection in direction indicated, in. Δ is numerically equal to change in length between anchor points caused by temperature increase or decrease (page 756).

Δ_X = deflection along the X axis, in.

Δ_Y = deflection along the Y axis, in.

Δ_Z = deflection along the Z axis, in.

F = generalized force acting as shown on sketches, lb.

F_X = reacting force acting along the X axis, lb.

F_Y = reacting force acting along the Y axis, lb.

F_Z = reacting force acting along the Z axis, lb.

H = height of bend, in.

l = length of cantilever or section of pipe, in.

R = mean radius of curvature of pipe bend, in.

r = mean radius of pipe cross section, in.

r_0 = outside radius of pipe cross section, in.

r_1 = inside radius of pipe cross section, in.

D = outside diameter of pipe, in. (pages 357 to 361).

d = inside diameter of pipe, in. (pages 357 to 368).

t = thickness of pipe wall, in. (pages 357 to 368).

h = tR/r^2 = pipe-bend ratio.

K = rigidity multiplication factor for curved pipe (Fig. 32, page 818).

β = longitudinal stress multiplication factor for curved pipe (Fig. 34, page 820).

γ = transverse stress multiplication factor for curved pipe (Fig. 37, page 823).

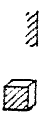
S = combined longitudinal bending and longitudinal pressure stress, psi.

S_l = longitudinal stress due to bending moment, psi.

S_s = shear stress due to torsion, psi.

S_t = transverse stress in curved pipe due to bending moment, psi.
S_1 = transverse pressure stress, psi.
S_2 = longitudinal pressure stress, psi.
M = bending moment at section indicated, in.-lb.
M_0 = restraining moment at end of pipe structure, in.-lb.
p = internal pressure, psi.
ψ_M = rotation due to a moment at the end of a cantilever.
= area of rectangular bending-moment diagram, divided by EI.
ψ_F = rotation due to a transverse force at the end of a cantilever.
= area of triangular bending-moment diagram, divided by EI.
$\psi_{l,c,etc.}$ = algebraic sum of the rotations of the elements between the point designated by the subscript and the fixed end of the section of piping.
= algebraic sum of the areas of the bending-moment diagrams to the left of the point designated by the subscript, divided by EI.

Actual Anchor Points in Pipe Line



Hypothetical Fixed Points for purposes of analysis.

Numerical values for the lengths l_1, l_2, and l_3 are introduced at this point to overcome the difficulty of expressing areas of the moment diagrams in general terms. Let l_1 = 100 in., l_2 = 120 in., and l_3 = 80 in. The resulting areas of the bending-moment diagrams of Fig. 10c are 5,000F, 8,000F, 9,600F and 3,200F (lb-in.^{2}), respectively, as indicated thereon.

The deflections at the free end (d) of the section of piping illustrated in Fig. 10a may now be obtained from the deflections

of the component parts as follows: (see Cases I, II, and III on page 789):

Case II.	$EI\Delta_{X_b} = \Sigma\psi_F \tfrac{2}{3} l_1 = 5{,}000F \times \tfrac{2}{3}\ 100 = 333{,}300F$	left.
Case III.	$\psi_M \tfrac{1}{2} l_1 = 8{,}000F \times \tfrac{1}{2}\ 100 = 400{,}000F$	left.
Case I.	$EI\Delta_{Y_c} = \Sigma\psi_b l_2 = 13{,}000F \times 120 = 1{,}560{,}000F$	down.
Case III.	$\psi_M \tfrac{1}{2} l_2 = 9{,}600F \times \tfrac{1}{2}\ 120 = 576{,}000F$	down.
Case I.	$EI\Delta_{X_d} = \Sigma\psi_c l_3 = 22{,}600F \times 80 = 1{,}810{,}000F$	left.
Case II.	$\psi_F \tfrac{2}{3} l_3 = 3{,}200F \times \tfrac{2}{3}\ 80 = 170{,}700F$	left.

The summations of the above increment deflections along the X axis and along the Y axis give

$$\Delta_X = \frac{2{,}714{,}000F}{EI} \text{ left}$$

and

$$\Delta_Y = \frac{2{,}136{,}000F}{EI} \text{ down.}$$

The deflections caused by any given force acting at the free end of any particular size and weight of pipe, arranged as in Fig. 10a, at any given working temperature may be found by substituting proper values for F, E, and I in the above equations. These equations are, of course, only true for stresses below the proportional limit for the material. This condition, in which only a single force acts at the end of a section of piping, was assumed for simplicity in explanation and is not of particular application to the problem of determining flexibility of piping subjected to thermal expansion.

The above illustration should not be confused with the usual problem of determining the flexibility of a section of piping subjected to temperature change, as in such cases the ends would be considered as anchored and the resulting vertical force and restraining moment at the hypothetical free end would have to be taken into account as is done in the example on page 796. The vertical force and restraining moment were omitted in order to simplify the initial explanation of this grapho-analytical method of determining the relations of forces and deflections in a pipe line.

Elementary Cases.—The majority of problems having to do with flexibility of pipe lines can be reduced to the consideration of two simple elements, a cantilever and a quarter bend. The rotations and deflections of these elements are tabulated below,

The area of the bending-moment diagram[1] divided by EI gives the total rotation of the element due to the action of the force or moment indicated in each case. These rotations are designated by ψ_F, ψ_M, etc., the subscript in each case referring to the particular force or moment acting at the free end of the element. The deflections are given in terms of the rotations and lengths or radii of the element in each case.

The following special symbols are used in these elementary cases:

Δ_F = deflection in direction of force, in.
$\Delta \perp_F$ = deflection perpendicular to direction of force, in.
Δ_A = deflection along moment arm, in.
$\Delta \perp_A$ = deflection perpendicular to moment arm, in.
θ = angle through which bend is formed, radians.

Ten cases are required to determine completely the relations existing between the possible forces and moments which may act on these elements and the resulting rotations and deflections.

As an example of the manner in which these relations are determined, consider a straight cantilever acted on by a transverse force F at the free end, Case II (Fig. 12).

The bending-moment diagram for a straight cantilever with a force on the free end is a triangle of base l and altitude Fl. The total rotation of the cantilever is

$$\psi_F = \frac{\text{area of bending-moment diagram}}{EI} = \frac{Fl^2}{2EI}.$$

The deflection of the free end from the tangent at the fixed end is the sum of the products of the rotations of the successive elements and their distances from the free end. This is equivalent to the area of the bending-moment diagram multiplied by the distance of its center of gravity from the free end.

$$\Delta_F = \psi_F \frac{2}{3} l = \frac{Fl^3}{3EI}.$$

The following tabulation of cases is given for use in the solution of problems dealing with flexibility of piping. The manner in which these cases are combined in complicated runs of piping will be demonstrated by a series of examples (see pages 796 to 815, inclusive).

[1] See "Strength of Materials," by J. E. Boyd, 4th ed., McGraw-Hill Book Company, Inc., New York, 1938, p. 160.

Case I. Fixed end of a cantilever rotated through an angle ψ (Fig. 11).

$\Delta_b = \psi_a l$ = deflection of the free end of a cantilever due to rotation of the fixed end.

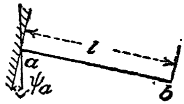


FIG. 11.—Case I.

Case II. Straight cantilever acted on by a transverse force F at the free end (Fig. 12).

$$\psi_F = \frac{Fl^2}{2EI}.$$

$\Delta_F = \psi_F \tfrac{2}{3} l$ = deflection in direction of force.

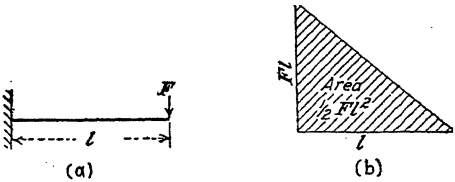


FIG. 12.—Case II.

Case III. Straight cantilever acted on by a moment M at the free end (Fig. 13).

$$\psi_M = \frac{Ml}{EI}.$$

$\Delta_A = \psi_M \tfrac{1}{2} l$ = deflection along moment arm as shown.

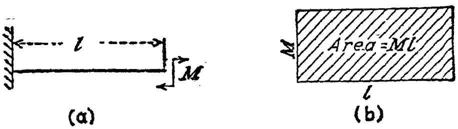


FIG. 13.—Case III.

Case IV. Straight cantilever acted on by a torsional moment M_t (Fig. 14).

$$\psi_{M_t} = 1.30\,\frac{M_t l}{EI}.$$

See page 793 for explanation of use of length 1.30l as the base of the bending-moment diagram.

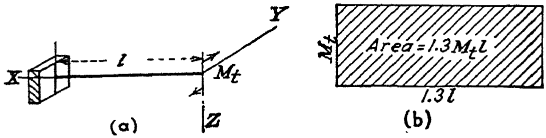


FIG. 14.—Case IV.

Case V. Quarter bend fixed at one end and acted on by a radial force F_r in plane of bend (Fig. 15).

$$\psi_{F_r} = \frac{F_r R^2}{KEI}$$

in which K is the rigidity multiplication factor (see page 818).
(a) $\Delta_{F_r} = \psi_{F_r} 0.7854R$ = deflection in direction of force F_r.
(b) $\Delta_{\perp F_r} = \psi_{F_r} 0.50R$ = deflection perpendicular to direction of force F_r.

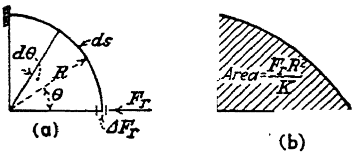


FIG. 15.—Case V.

Case VI. Quarter bend fixed at one end and acted on by an axial force F_a in plane of bend (Fig. 16).

$$\psi_{F_a} = \frac{0.5708F_a R^2}{KEI}$$

(a) $\Delta_{F_a} = \psi_{F_a} 0.624R$ = deflection in direction of force F_a.
(b) $\Delta \perp_{F_a} = \psi_F 0.876R$ = deflection perpendicular to direction of force F_a.

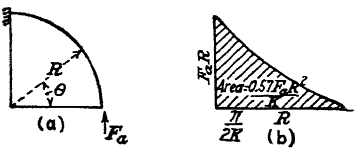


FIG. 16.—Case VI.

Case VII. Quarter bend fixed at one end and acted on by a bending moment M in plane of bend (Fig. 17).

$$\psi_M = \frac{1.5708MR}{KEI}.$$

(a) $\Delta_{A_M} = \psi_M 0.637R$ = deflection along moment arm as shown.
(b) $\Delta_{\perp A_M} = \psi_M 0.363R$ = deflection perpendicular to moment arm as shown.

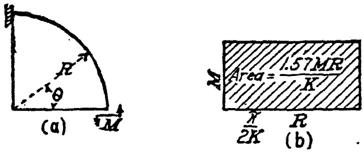


FIG. 17.—Case VII.

Case VIII. Quarter bend fixed at one end and acted on by a force F perpendicular to plane of bend (Fig. 18).

(a) $\psi_{XY} = \dfrac{0.5064FR^2}{EI}$; Δ_{FXY} = deflection in direction of force (proportional to rotation in plane perpendicular to free end of bend).

$\psi_{ZY} =$ deflection in direction of force (proportional to rotation in plane tangent to free end of bend).

$\Delta_{FY} = \Delta_{FXY} + \Delta_{FYZ} = \dfrac{1.2485FR^3}{EI}$ = total deflection in direction of force.

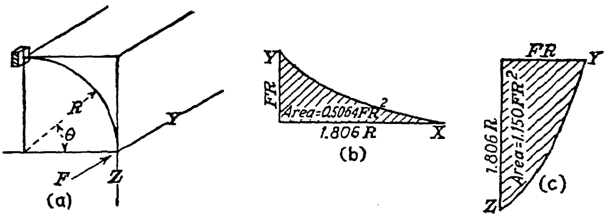


FIG. 18.—Case VIII.

Case IX. Quarter bend fixed at one end and acted on by a moment M in plane perpendicular to plane of bend (Fig. 19).

(a) $\frac{0.150MR}{EI}$; $\Delta_{MXY} = \psi_{MXY}0.695R$ = deflection to left along moment arm (proportional to rotation in plane perpendicular to free end of bend).

(b) $\frac{1.806MR}{EI}$; $\Delta_{MYZ} = \psi_{MYZ}0.695R$ = deflection to right along moment arm (proportional to rotation in plane tangent to free end of bend).

$$\Delta_{MY} = \Delta_{MYZ} - \Delta_{MXY} = \frac{1.150MR^2}{EI} \text{ resultant deflection.}$$

(c) $\Delta_{MY} \quad \psi_{MYZ}0.637R = \frac{1.150MR^2}{EI}$ = resultant deflection in terms of larger rotation.

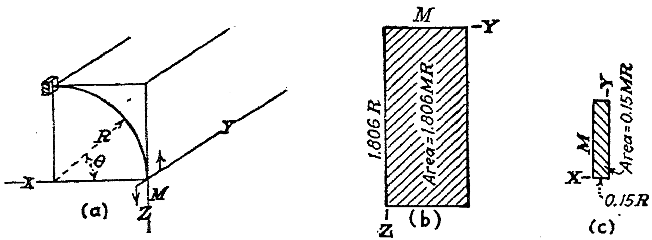


FIG. 19.—Case IX.

Case X. Quarter bend fixed at one end and acted on by a torque moment M_t in plane perpendicular to free end of bend (Fig. 20).

(a) $\psi_{XY} = \frac{1.806M_tR}{EI}$; $\Delta_{M_tXY} = \psi_{XY}0.306R$ = deflection to right perpendicular to plane of bend (proportional to rotation in plane perpendicular to free end of bend).

(b) $\psi_{YZ} = \frac{0.150M_tR}{EI}$; $\Delta_{M_tYZ} = \psi_{YZ}0.306R$ = deflection to left perpendicular to plane of bend (proportional to rotation in plane tangent to free end of bend).

$$\Delta_{M_tY} = \Delta_{M_tXY} - \Delta_{M_tYZ} = \frac{0.5064M_tR^2}{EI} = \text{resultant deflection.}$$

(c) $\Delta_{M_tY} = \psi_{XY}0.280R = \frac{0.5064M_tR^2}{EI}$ = resultant deflection in terms of larger rotation.

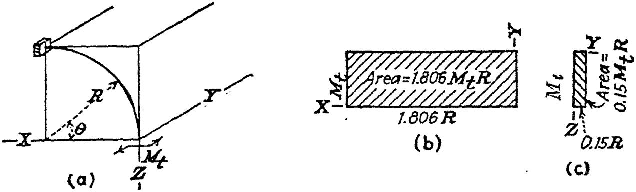


FIG. 20.—Case X.

Virtual Lengths.—Variations in the flexural rigidity of different component parts of a section of piping due to changes in size or weight of piping, flattening of the cross section of a curved pipe due to flexure in plane of the bend,[1] effect of torsion, etc., may be, and in the case of sections of piping in space must be, taken into account by the use of virtual lengths of the abscissas of the bending-moment diagrams.

In Case IV, the abscissa of the bending-moment diagram representing torsional moment is 1.30 times the actual length of the cantilever. The factor 1.30 is necessary to convert the expression for torsional rigidity into linear equivalents. It is determined by replacing the terms J and G in the equation for torsion, $\psi = M_tl/JG$, by their equivalents in terms of I and E. The polar moment of inertia $J = 2I$. The modulus of shearing rigidity G in terms of E is determined from the relation[2] $G = E/2(1 + \lambda)$ in which λ is Poisson's ratio. λ for steel pipe may be assumed as 0.30. Values of λ for other materials are given on page 344. Combining the above relations, the expression for torsion in terms of linear rigidity is obtained

$$\psi = \frac{M_tl}{JG} = \frac{M_tl}{\dfrac{2EI}{2(1 + 0.30)}} \quad \frac{1.30M_tl}{EI}.$$

Wherever torsion occurs in a section of piping under investigation, whether caused by reacting forces at the end or by resisting moments, the abscissas of the bending-moment diagrams for the straight component parts subjected to torsion, Case IV, are obtained by multiplying the true lengths of the component parts by the factor 1.30.

[1] See footnote 2(*c*), and 2(*i*), p. 780.
[2] See "Strength of Materials," *op. cit.*, p. 52.

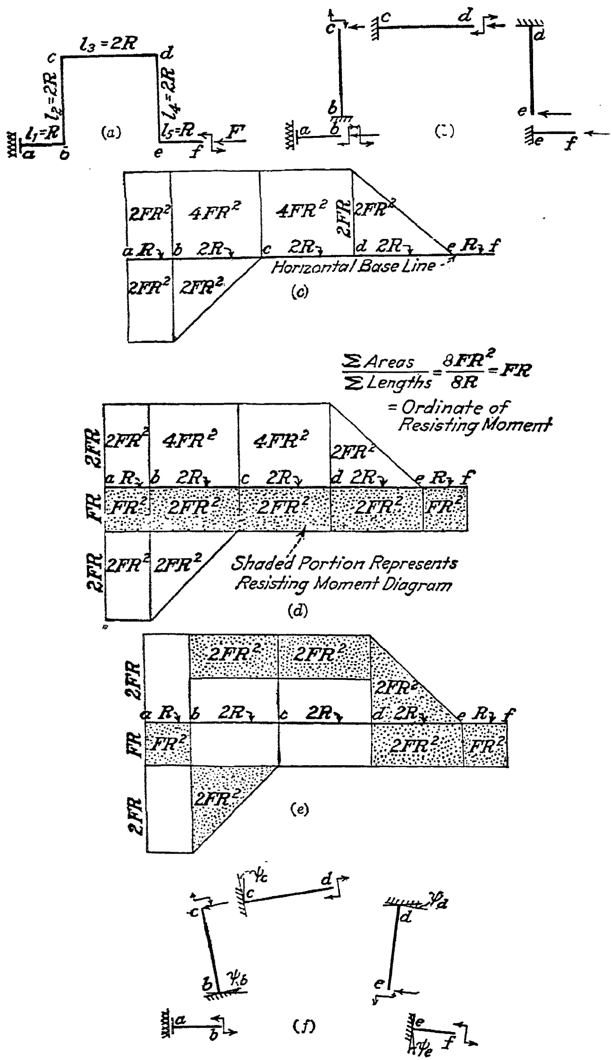


FIG. 21.—Square bend equivalent to expansion U bend anchored at both ends.

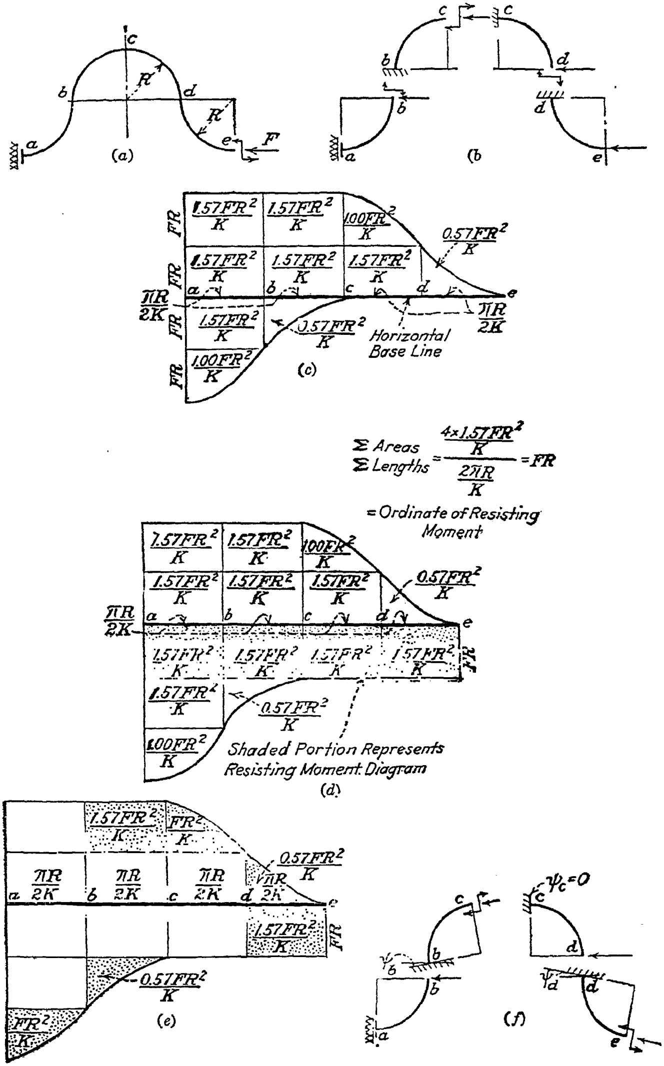


FIG. 22.—Expansion U bend anchored at both ends.

In Cases V, VI, and VII, the abscissas of the bending-moment diagrams are the developed lengths of the bends modified by the rigidity multiplication factor K (see page 818).

In Cases VIII, IX, and X, the abscissas of the bending-moment diagrams are 1.8 times the mean radii of the bends. The factor 1.8 is found from the combined effects of torsion and bending which occur in each of these cases.[1]

APPLICATION OF ELEMENTARY CASES

Symmetrical Bends in One Plane.—A square bend and an expansion U bend which have the same over-all dimensions are illustrated in Figs. 21 and 22. Both ends of each bend are actually anchored so that the linear expansion between the ends, caused by change in temperature, is compressed back into the bend. The magnitude of this linear expansion, or hypothetical displacement, designated as Δx is determined from the elongation of the piping due to temperature change with respect to the distance between anchors measured along the X axis. The flexural rigidity of the bend resisting this compression causes horizontal forces at the anchors and moments that tend to rotate the ends of the bend about the anchors in the plane of the bend. In order, however, to determine the relations existing between the horizontal forces and the expansion which is compressed back into the bend, the left end is considered as fixed and the anchor at the right end is replaced by its equivalents, *viz.*, a horizontal force and a moment which prevent lateral displacement and angular rotation of the end of the bend. Since the ends of the bend are prevented from rotation, the algebraic sum of the rotations of the elements of the bend from one end of the bend to the other is zero. This fact permits the direct determination of the ordinate of the resisting-moment diagram for all cases in which the ends of a section of piping under investigation are anchored.

In Fig. 21*b* the component parts of the square bend are shown with the transposed forces and moments arising from the horizontal force F acting on each part. The bending-moment diagram corresponding to these forces and moments is given in Fig. 21*c*.

[1] The effect of flattening of the cross section under sidewise loading in a plane at 45 deg to the plane of the bend reported by Vigness (see reference 2 (*i*), p. 780) has not been taken into account in these cases since its effect in the usual three-dimensional problem is not significant. In trying to reconcile theory with test results for sidewise loading of individual bends, the modifications suggested by Vigness should be taken into account.

If there were no restraint at the ends of the bend, *i.e.*, if the ends were supported on knife-edges, the relation between the reacting force and the deflections would be determined directly from the diagram of Fig. 21*c*. But by the conditions of the problem, the ends are actually anchored. Hence, there is a moment at the hypothetical free end which prevents rotation of the end. The magnitude of this restraining moment may be found from the condition described above, that the algebraic sum of the rotations of the elements from one end of the bend to the other must be zero. Since these rotations are represented by the areas of the bending-moment diagram, the algebraic sum of the area divided by the sum of the lengths of the component parts, either virtual or true lengths as the case may be, gives the ordinate of the resisting-moment diagram, as shown in Fig. 21*d*. Since the algebraic sum of the areas of the bending-moment diagram of Fig. 21*d* is zero, the condition that the algebraic sum of the rotation of the elements from one end of the bend to the other should be zero is met.

In Fig. 21*e*, the resultant bending-moment areas for each component parts are shown shaded. This diagram is identical with that of Fig. 21*d* save that congruent areas[1] directly above and below the horizontal base line for each component part have been canceled, which has been done to eliminate unnecessary work in the solution.

In Fig. 21*f* are shown the component parts with the resulting rotations, forces, and moments corresponding to the resultant bending-moment areas of Fig. 21*e*.

The deflections at the hypothetical free end of the bend are obtained from Fig. 21*e* as follows: The lengths of the component parts are given in terms of the radius R of the expansion U bend of Fig. 22 in order to afford a direct comparison between the flexibility of a square bend and an expansion U bend having the same over-all dimensions. The deflections of the component parts are found in terms of the rotations of the parts and their lengths (see tabulation of cases, pages 789-793, inclusive).

Case III. $EI\Delta_{Y_b} =$		$= FR^2 \times \frac{1}{2}R$	$= \frac{1}{2}FR^3$ up.
Case I. $EI\Delta_{X_c} =$	$\psi_b(2R)$	$= FR^2 \times 2R$	$= 2FR^3$ left.
Case II.	$\psi_F \frac{2}{3}(2R)$	$= 2FR^2 \times \frac{2}{3}(2R)$	$= \frac{8}{3}FR^3$ left.
Case III.	$\psi_M \frac{1}{2}(2R)$	$= 2FR^2 \times \frac{1}{2}(2R)$	$= 2FR^3$ right.
Case I. $EI\Delta_{Y_d} =$		$= FR^2 \times (2R)$	$= 2FR^3$ up.

[1] Congruent areas are identical, *i.e.*, have exact coincidence throughout.

Case III. $\quad \psi_M \frac{1}{2}(2R) = 2FR^2 \times \frac{1}{2}(2R) = 2FR^3$ down.
Case I. $EI\Delta_{X_e} = \Sigma\psi_d(2R) = FR^2 \times (2R) = 2FR^3$ left.
Case II. $\quad \psi_F \frac{2}{3}(2R) = 2FR^2 \times \frac{2}{3}(2R) = \frac{8}{3}FR^3$ left.
Case III. $\quad \psi_M \frac{1}{2}(2R) = 2FR^2 \times \frac{1}{2}(2R) = 2FR^3$ right.
Case I. $EI\Delta_{Y_j} = \Sigma\psi_e(R) = FR^2 \times R = FR^3$ down.
Case III. $\quad \psi_M \frac{1}{2}R = FR^2 \times \frac{1}{2}R = \frac{1}{2}FR^3$ up.

The summation of the above deflections to the left, minus those to the right, gives the resultant deflection of the free end of the bend in the X direction. In like manner, the summation of the deflections up minus those down gives the resultant deflection of the free end of the bend in the Y direction.

$$\Delta_X = \frac{16}{3}\frac{FR^3}{EI} = 5.33\frac{FR^3}{EI} \text{ left.}$$
$$\Delta_Y = 0.$$

The expansion U bend (Fig. 22) is handled in a manner identical with that followed in finding the relation of forces and deflections for its square-bend counterpart. The rigidity multiplication factor K is constant throughout the bend and, as there are no straight lengths to take into account, it is merely carried through as a constant.

The deflections at the free end of the expansion U bend are found from the deflections of its component parts given graphically in Fig. 22f as follows (see tabulation of cases, pages 789 to 793, inclusive):

Case V(a). $\quad EI\Delta_{X_b} = \psi_F 0.7854R = \frac{FR^2}{K} \times 0.7854R$

$$0.7854 \quad \text{left.}$$

Case V(b). $\quad EI\Delta_{Y_b} = \psi_F 0.50R = \frac{FR^2}{K} \times 0.50R = 0.50\frac{FR^3}{K}$ up.

Case I. $\quad EI\Delta_{X_e} = \Sigma\psi_b R =$

Case VI(a). $\quad \psi_F 0.624R = \frac{0.5708FR^2}{K} \times 0.624R$

$$= 0.3562\frac{FR^3}{K} \text{ left.}$$

Case VII(b). $\quad \psi_M 0.363R = \frac{1.5708FR^2}{K} \times 0.363R$

$$= 0.5708\frac{FR^3}{K} \text{ right.}$$

Case I. $EI\Delta_{Y_c} = \Sigma\psi_b R \quad = \frac{FR^2}{K} \times R = \frac{FR^3}{K}$ up.

Case VI(b). $\psi_F 0.876R = \frac{0.5708FR^2}{K} \times 0.876R$

$= 0.50\frac{FR^3}{K}$ up.

Case VII(a). $\psi_M 0.637R = \frac{1.5708FR^2}{K} \times 0.637R$

$= \frac{FR^3}{K}$ down.

Case I. $EI\Delta_{X_d} = \Sigma\psi_c R \quad = 0 \times R \quad = 0$

Case V(a). $\psi_F 0.7854R = \frac{FR^2}{K} \times 0.7854R$

$= 0.7854\frac{FR^3}{K}$ left.

Case I. $EI\Delta_{Y_d} = \Sigma\psi_c R \quad = 0 \times R \quad = 0$

Case V(b). $\psi_F 0.50R = \frac{FR^2}{K} \times 0.50R$

$= 0.50\frac{FR^3}{K}$ down.

Case I. $EI\Delta_{X_e} = \Sigma\psi_d R \quad = \frac{FR^2}{K} \times R = \frac{FR^3}{K}$ left.

Case VI(a). $\psi_F 0.624R = \frac{0.5708FR^2}{K} \times 0.624R$

$= 0.3562\frac{FR^3}{K}$ left.

Case VII(b). $\psi_M 0.363R = \frac{1.5708FR^2}{K} \times 0.363R$

$= 0.5708\frac{FR^3}{K}$ right.

Case I. $EI\Delta_{Y_e} = \Sigma\psi_d R \quad = \frac{FR^2}{K} \times R = \frac{FR^3}{K}$ down.

Case VI(b). $\psi_F 0.876R = \frac{0.5708FR^2}{K} \times 0.876R$

$= 0.50\frac{FR^3}{K}$ down.

Case VII(a). $\psi_M 0.637R = \frac{1.5708FR^2}{K} \times 0.637R$

$= \frac{FR^3}{K}$ up.

The summations of the above deflections give the resultant deflections of the free end of the bend.

$$\Delta_X \quad 3.1416 \frac{FR^3}{KEI} \text{ left.}$$

$$\Delta_Y \quad 0.$$

It is, of course, apparent from Figs. 21 and 22 that the deflections in the vertical or Y direction are zero, but these deflections were carried through since they make a very easy check on the method, *i.e.*, the summation of the vertical deflections must of necessity be zero where the bend is symmetrically loaded.

The results of the above determination show that the relative flexibility of the square bend and the expansion U bend having the same dimensions depends entirely upon the value of K. The equation for the square bend determined above, $\Delta_X = 5.33FR^3/EI$, and that for the expansion U bend, $\Delta_X = 3.1416(FR^3)/(KEI)$, incidentally show that the assumption that square turns can be considered as representing arcs of circles in determining flexibility of piping may be in error more than 40 per cent either too great or too small, depending upon the value of K. When $K = 0.59$, the expression for deflection of an expansion U bend is identical with that for its square-bend counterpart. Where K is constant for the entire section of piping under consideration, the section may be handled as made up of straight lengths and square turns and a correction factor based on the above relation applied. K depends upon the weight of pipe, the size of pipe, and the radius to which it is bent (see Fig. 32 and pages 816–818 inclusive, for values of K and full discussion of its significance).

The relations between deflections and reacting forces determined above for the square bend and the expansion U bend may be found directly from the curves of Figs. 40 and 41. For the square bend the ratio of the sum of the lengths of straight pipe on each end to twice the height is 0.5. The height is $2R$. The value 0.666 read from the curve $r = 1$ corresponding to $n = 0.5$ multiplied by $(2R)^3$ gives the relation $\Delta_X = 5.33(FR^3)/(EI)$. The corresponding relation for the expansion U bend is found from the curve $K = 1$ for $n = 0$ or $\Delta_X = 3.1416(FR^3)/(KEI)$ as indicated thereon.

Pipe Line in Space.[1]—A typical pipe line in space is illustrated in Fig. 23. Both ends of the section of piping are rigidly anchored,

[1] For the solution of "Intersecting Piping in Space with Three Anchorages,"

as in the previous examples. For purposes of analysis the anchor at the right end of the section is again represented by reacting forces and resisting moments which prevent displacement and rotation of the end as the section of piping expands due to rise in temperature.

The hypothetical displacements along the X, Y, and Z axes of the right end of the section of piping are determined from the elongation of the piping due to temperature change with respect to the distances between anchors measured along these axes.

The reacting forces which prevent displacement are designated as F_X, F_Y, and F_Z, corresponding to the axes along which they

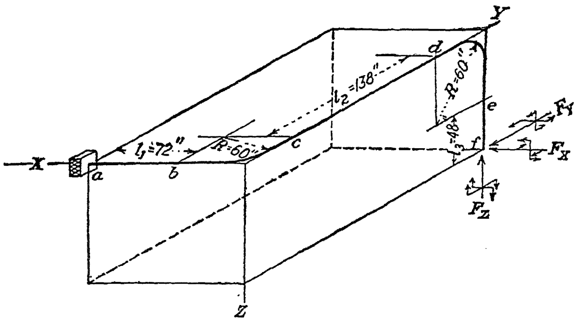


FIG. 23.—Typical section of pipe line in space anchored at both ends.

act. Each force causes rotation of the elements of the section of piping in three planes. An examination of Cases IX and X, however, shows that the rotations at the ends of a quarter bend acted upon by bending or torsional moments consist of a principal rotation in the plane in which the moment acts and a minor rotation in a perpendicular plane. The principal rotation is twelve times the secondary in both cases. In order to simplify the description and solution of this complex problem, these minor rotations are neglected in the following analysis. The effect of

by the grapho-analytical method, reference may be made to "Piping Flexibility and Stresses," by A. D. Vinieratos and D. R. Zeno, Cornell Maritime Press, 1941, p. 67. The three-anchor problem also is covered in a booklet on "Design of Piping Systems," published by the M. W. Kellogg Co., New York, 1941, and in a paper entitled "Simplified Method of Analysis of Reactions Developed by Expansion in a Three-anchor Piping System," by Boris Lochak, *Trans. ASME*, Vol. 66, No. 4, p. 311, 1944.

taking these secondary rotations into account is to increase slightly the stresses found.

The results obtained by applying the cases to the component parts of the line, when combined by the grapho-analytical method, give equations between the reacting force and the displacements occurring in each plane. As in this case there are three planes, the magnitude of the three reacting forces may be found by solving simultaneously the three equations. The basic equation relating displacement and reacting force is $EI\Delta = CF$. This equation is derived from the fundamental equation for deflection obtained by solution of the differential equation of the elastic curve,[1] $d^2y/dx^2 = M/EI$. Solving this equation for deflection gives

$$\Delta = \int \frac{Myds}{EI}.$$

In the case of a quarter bend as illustrated in Fig. 15 of elementary cases, $M = FR \sin \theta$, $y = R \sin \theta$, $ds = Rd\theta$, and the limits of integration are 0 and $\pi/2$. The deflection in the direction of the applied radial force F_r is given by

$$\begin{aligned}\Delta &= \frac{1}{KEI} \int_0^{\pi/2} F_r R^3 \sin^2 \theta d\theta \\ &= \frac{F_r R^3}{KEI} \left[_0^{\pi/2} - \frac{1}{2} \cos \theta \sin \theta + \frac{1}{2} \theta \right] \\ &= \frac{\pi}{4} \frac{F_r R^3}{KEI} \text{ (see Case V, page 790).}\end{aligned}$$

The quantity $(\pi/4)R^3$ is represented by the factor C in the general equation relating deflections and forces. The rigidity multiplication factor[2] K which appears in the above expression for flexure of a curved pipe in the plane of the bend is taken into account in the moment diagrams so that equation becomes $EI\Delta = CF$. The quantity C is found from the area-moment diagrams in each particular case, as shown on the figures referred to.

The numerical values and operating conditions assumed for illustration of this method of determining the relations of deflection and acting forces for a pipe line in space (Fig. 23) represent typical conditions encountered in high-pressure superheated-steam lines.

[1] See "Strength of Materials," by S. Timoshenko, D. Van Nostrand Company, Inc., 250 Fourth Ave., New York, N.Y., Part I, 2d ed., p. 135.

[2] See pp. 816 to 818, inclusive, for full discussion of K.

Assumed Data:

1. Pipe, 12 in. nominal diameter, steel, wall thickness = ½ in.
2. Temperature variation, 60 to 725 F.
3. Over-all dimensions: X = 11 ft 0 in.; Y = 21 ft 6 in., and Z = 9 ft 0 in.
4. Radius of bend: mean radius of bend equals five times nominal diameter of pipe, R = 60 in.
5. Straight lengths: l_1 = 72 in.; l_2 = 138 in.; and l_3 = 48 in.

Solution:

1. K, rigidity multiplication factor for curved pipe, applies to flexure in plane of the bend only. Its value is found conveniently from chart of Fig. 32. For 12-in. pipe bent to a radius of five times nominal diameter of pipe and having a wall thickness of ½ in.

$$K = 0.50.$$

2. E, modulus of elasticity, in tension or compression, is found from curves of Fig. 39 corresponding to a working temperature of 725 F.

$$E = 25{,}000{,}000 \text{ psi.}$$

3. I, moment of inertia of a 12-in. pipe having ½-in. wall thickness is found from formula given on page 860.

$$I = 361.5 \text{ in.}^4$$

4. Δ_X, Δ_Y, and Δ_Z, the hypothetical displacements of the free end of the section of piping, are found from the over-all dimensions given in item 3 of assumed data and the elongation of the pipe due to temperature change. The elongation of pipe corresponding to the temperature range of 60 to 725 F. (see item 2 of assumed data) is found from Table I as 5.88 in. per 100 ft of pipe.

$$\Delta_X = 11 \times 0.0588 = 0.647 \text{ in.}$$
$$\Delta_Y = 21.5 \times 0.0588 = 1.270 \text{ in.}$$
$$\Delta_Z = 9 \times 0.0588 = 0.530 \text{ in.}$$

5. C, the coefficient of the force, is determined in each plane as follows:

a. Consider the effect of the F_Y force in the YZ plane alone. The transposed forces and moments caused by the F_Y force in the YZ plane are shown acting on the component parts of the section

of piping in Fig. 24*a*. In constructing the bending-moment diagram of Fig. 24*b*, the abscissas of the moment diagram for each component part are laid off on a horizontal base line.

The abscissas of the moment diagrams are either the true length of the part or a virtual[1] length depending upon the particular forces or moments acting on the part. In Fig. 24*a*, the component part *ef* is a cantilever acted on by a force at the free end (see

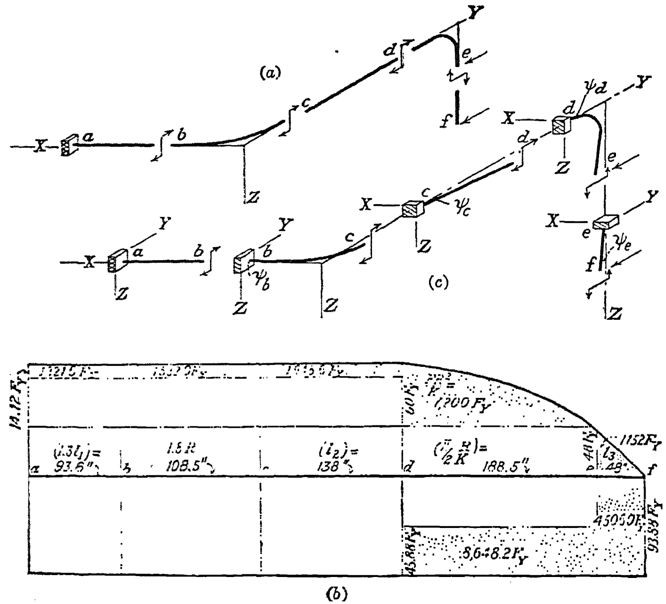


FIG. 24.—Determination of deflection caused by F_Y force in YZ plane.

Case II). The abscissa of the moment diagram is the true length of this part. The quarter bend *de* is acted on by a radial force and a moment, both in the plane of the bend. The abscissa of the moment diagrams for this loading is the developed length of this quarter bend divided by K (see Cases V and VII). Part *cd* is a cantilever with a moment at its free end. It is represented by its true length (see Case III). The quarter bend *bc* is acted on by a moment in the ZY plane that is perpendicular

[1] See p. 793 for discussion of virtual lengths.

to the plane of the bend. The abscissa of the moment diagram for the quarter bend with this loading has a length 1.8 times the mean radius[1] of the bend (see Case IX). Part *ab* is acted on by a torsional moment caused by force F_Y in the YZ plane. Hence the abscissa of the moment diagram for *ab* is 1.3 times its true length.

The bending-moment diagram for the entire section of piping may now be drawn. The moment diagram for part *ef* is a triangle, zero at point f and $F_Y l_3$ at e. The moment $F_Y l_3$ carries all the way through the section of piping, hence a horizontal line is drawn to end a. The quarter bend *de* is subjected to an additional bending moment due to the transposed radial force. This diagram is a parabolic half segment, zero at e and $F_Y R$ at d. This moment $F_Y R$ also carries through the section to end a. The bending-moment diagram thus obtained corresponds to the action of the transposed forces and moments shown in Fig. 24*a*.

The ordinate of the resisting-moment diagram is found by dividing the sum of the areas just determined by the sum of the lengths of the abscissas of the bending-moment diagrams. The algebraic sum of the areas of the acting-moment diagrams and of the resisting-moment diagrams is consequently zero for the entire section of piping.

The resultant bending moment acting on each part is shown shaded in Fig. 24*b*. Congruent areas have been canceled as in the previous example. The forces and moments which correspond to these shaded areas are shown applied to the component parts in Fig. 24*c*. The deflections caused by these forces and moments and those due to rotation of the hypothetical fixed ends of the component parts are found from the elementary cases to which they correspond. These deflections are summarized and the relations between displacements and the force F_Y in the YZ plane are obtained as indicated on the figure.

FIG. 24, F_Y FORCE IN YZ PLANE

Case I. $EI\Delta_{Z_e} = \Sigma\, \psi_b R = 1{,}321.6F_Y \times 60 = 79{,}296F_Y$ down.

Case IX(c). $\psi_{MYZ} 0.637R = 1{,}532F_Y \times 38.22$

$= 58{,}553F_Y$ down.

Case I. $EI\Delta_{Z_d} = \Sigma\, \psi_c l = 2{,}854F_Y \times 138 = 393{,}797F_Y$ down.

Case III. $\psi_M \tfrac{1}{2} l = 1{,}949F_Y \times 69$

$= 134{,}453F_Y$ down.

[1] See p. 796 for derivation.

Case I. $EI\Delta_{Z_e} = \Sigma\,\psi_d R = 4{,}802F_Y \times 60 = 288{,}132F_Y$ down.
Case V(*b*). $\psi_{F_r}0.50R = 7{,}200F_Y \times 30$
$= 216{,}000F_Y$ down.
Case VII(*b*). $\psi_M 0.363R = 8{,}648F_Y \times 21.78$
$= 188{,}358F_Y$ up.
Case I. $EI\Delta_{Y_e} = \Sigma\,\psi_d R = 4{,}802F_Y \times 60 = 288{,}132F_Y$ left.
Case V(*a*). $\psi_{F_r}0.7854R = 7{,}200F_Y \times 47.12$
$= 339{,}264F_Y$ left.
Case VII(*a*). $\psi_M 0.637R = 8{,}648F_Y \times 38.22$
$= 330{,}534F_Y$ right.
Case I. $EI\Delta_{Y_f} = \Sigma\,\psi_e l = 3{,}354F_Y \times 48 = 160{,}992F_Y$ left.
Case II. $\psi\tfrac{2}{3}l = 1{,}152F_Y \times 32 = 36{,}864F_Y$ left.
Case III. $\psi_M\tfrac{1}{2}l = 4{,}506F_Y \times 24 = 108{,}144F_Y$ right.
Summation: $EI\Delta_Z = 981{,}873F_Y$ down; $EI\Delta_Y = 386{,}574F_Y$ left.

b. The F_Y force also causes bending in the XY plane. The transposed forces and moment which act in the XY plane are shown in Fig. 25*a*. In this case, although it is evident that there is no moment in the XY plane due to the action of the force F_Y on the component part *ef* at the right end of the section of piping, it must be remembered that the resisting moment does act on this part. Owing to the action of the resisting moment, part *ef* is in torsion, hence the length which represents part *ef* in the bending-moment diagram is 1.3 times the true length of *ef*. Quarter bend *de* is also acted on by the resisting moment, hence has a torsional moment at the end. It is represented by the length $1.8R$, as in Case X. Part *cd* is in bending, due to the resisting moment, so is represented by its true length. Quarter bend *bc* is acted on by an axial force in the XY plane, therefore is represented by the developed length of the bend divided by K (see Case VI). Part *ab* is in bending due to a force and a bending moment, so is represented by its true length. The sum of the areas of the moment diagrams for the acting force divided by the sum of the lengths of the abscissas of the moment diagrams gives the ordinate of the resisting-moment diagram.

The resultant bending-moment areas and the forces and moments which they represent are shown in Figs. 25*b* and 25*c*. The deflections are found from the elementary cases to which the shaded areas correspond. The summations of these deflections give the relations between displacements and the acting force F_Y in the XY plane.

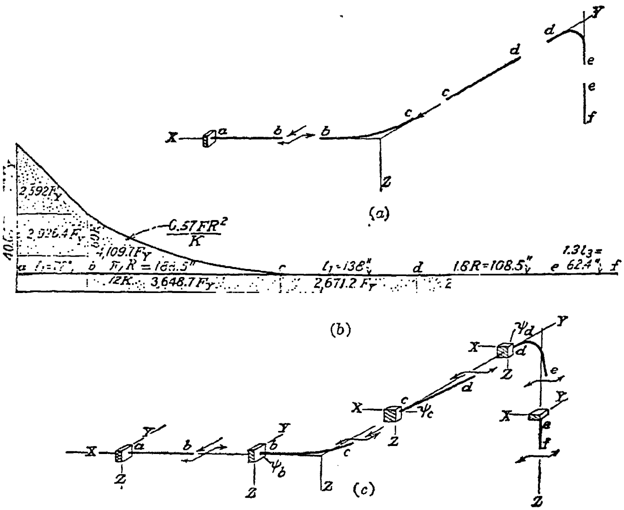


Fig. 25.—Determination of deflections caused by F_Y force in XY plane.

Fig. 25, F_Y Force in XY Plane

Case II. $EI\Delta_{Y_b} = \Sigma\, \psi_F \tfrac{2}{3} l = 2{,}592F_Y \times 48 = 124{,}416F_Y$ left.
Case III. $\psi_M \tfrac{1}{2} l = 2{,}926.4F_Y \times 36$
$= 105{,}350F_Y$ left.
Case I. $EI\Delta_{Y_c} = \Sigma\, \psi_b R = 5{,}518.4F_Y \times 60 = 331{,}104F_Y$ left.
Case VI(a). $\psi_{F_a} 0.62R = 4{,}109.7F_Y \times 37.44$
$= 153{,}867F_Y$ left.
Case VII(b). $\psi_M 0.363R = 3{,}648.7F_Y \times 21.78$
$= 79{,}469F_Y$ right.
Case I. $EI\Delta_{X_c} = \Sigma\, \psi_b R = 5{,}518.4F_Y \times 60 = 331{,}104F_Y$ right.
Case VI(b). $\psi_{F_a} 0.876R = 4{,}109.7F_Y \times 52.56$
$= 216{,}005F_Y$ right.
Case VII(a). $\psi_M 0.637R = 3{,}648.7F_Y \times 38.22$
$= 139{,}453F_Y$ left.
Case I. $EI\Delta_{X_d} = \Sigma\, \psi_c l = 5{,}979.4F_Y \times 138 = 825{,}157F_Y$ right.
Case III. $2{,}671.2F_Y \times 69$
$= 184{,}313F_Y$ left.
Case I. $EI\Delta_{X_e} = \Sigma\, \psi_d R = 3{,}308F_Y \times 60 = 198{,}492F_Y$ right.
Case X(c). $\psi_{M_t} 0.280R = 2{,}100F_Y \times 16.80$
$= 35{,}283F_Y$ left.

Summation: $EI\Delta_Y = 635{,}269F_Y$ left; $EI\Delta_X = 1{,}211{,}710F_Y$ right

c. The deflections caused by the F_Z force acting in the YZ plane are found by means of the grapho-analytical method from Figs. 26*a*, *b*, and *c*. It is evident from inspection of Fig. 26*a* that the part *ef* is in bending due to the resisting moment. It is represented by its true length in Fig. 26*b*. Quarter bend *de* is subjected to an axial force; therefore, the length of the abscissa of the moment diagram is given by the developed length of the

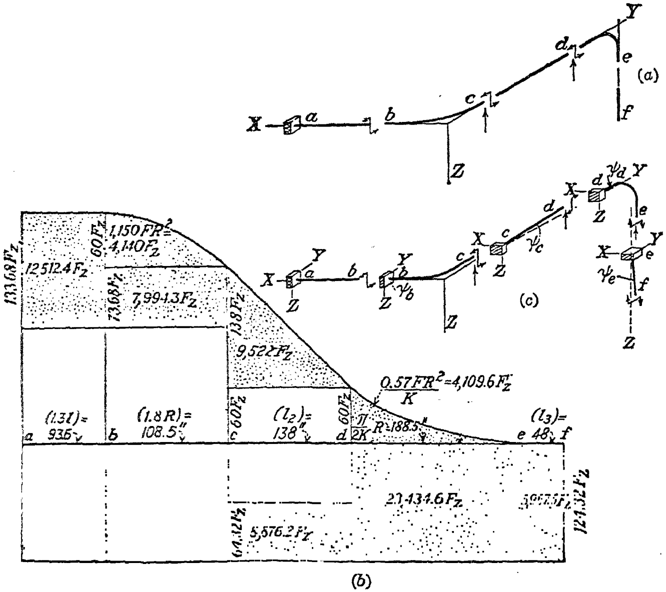


FIG. 26.—Determination of deflections caused by F_Z force in YZ plane.

bend divided by K. Part *cd* is represented by its true length. Quarter bend *bc* is acted on by a force and a moment perpendicular to plane of the bend hence is represented by the length $1.8R$ (see Cases VIII and IX). Part *ab* is in torsion due to the action of force F_Z in the YZ plane. It is, consequently, represented by a length $1.3l$.

The bending-moment diagram is drawn for the entire section, as before. Resultant areas are found and identified with the elementary cases to which they correspond. The summation of the deflections found from the elementary cases gives the relation

between the displacements along the Z and Y axes, respectively, and the force FZ.

FIG. 26, F_Z FORCE IN YZ PLANE

Case I. $EI\Delta_{Z_c} = \Sigma\, \psi_b R = 12{,}512.4F_Z \times 60 = 750{,}744F_Z$ up.
Case VIII(b). $\psi_{FYZ} 0.754R = 4{,}140F_Z \times 45.24$
$= 187{,}294F_Z$ up.
Case IX(c). $= 7{,}994F_Z \times 38.22$
$= 305{,}542F_Z$ up.
Case I. $EI\Delta_{Z_d} = \quad = 24{,}647F_Z \times 138$
$= 3{,}041{,}245F_Z$ up.
Case II. $\times 92$
$= 876{,}024F_Z$ up.
Case III. [illegible] $\times 69$
$= 612{,}458F_Z$ down.
Case I. $EI\Delta_{Z_e} = \Sigma\, \psi_d R = 25{,}293F_Z \times 60 = 1{,}517{,}550F_Z$ up.
Case VI(a). $= 4{,}109.6F_Z \times 37.44$
$= 153{,}863F_Z$ up.
Case VII(b). $= 23{,}534F_Z \times 21.78$
$= 510{,}406F_Z$ down.
Case I. $EI\Delta_{Y_e} = \Sigma\, \psi_d R = 25{,}293F_Z \times 60$
$= 1{,}517{,}550F_Z$ right.
Case VI(b). $4{,}109.6F_Z \times 52.56$
$= 216{,}000F_Z$ right.
Case VII(a). $23{,}435F_Z \times 38.22$
$= 895{,}670F_Z$ left.
Case I. $EI\Delta_{Y_f} = \Sigma\, \psi_e l = 5{,}968 \times 48 = 286{,}440F_Z$ right.
Case III. $\psi_M \tfrac{1}{2} l = 5{,}968 \times 24 = 143{,}220F_Z$ left.
Summation: $EI\Delta_Z = 6{,}069{,}398F_Z$ up; $EI\Delta_Y = 981{,}105F_Z$ right.

d. The F_Z force also causes bending in the XZ plane which is determined from Figs. 27*a*, *b*, and *c*. Rotations in the XZ plane are resisted by bending in part *ef*, a bending moment at the end of the quarter bend *de*, which is perpendicular to the plane of the bend, torsion in part *cd*, torsional moment at the end of quarter bend *bc*, and bending in part *ab*. The abscissas of the bending-moment diagram of Fig. 27*b* correspond to the virtual lengths representing the above conditions.

The deflections found from the cases which correspond to the resultant areas are summarized and the relations found between displacements along the X and Z axes and the F_Z force.

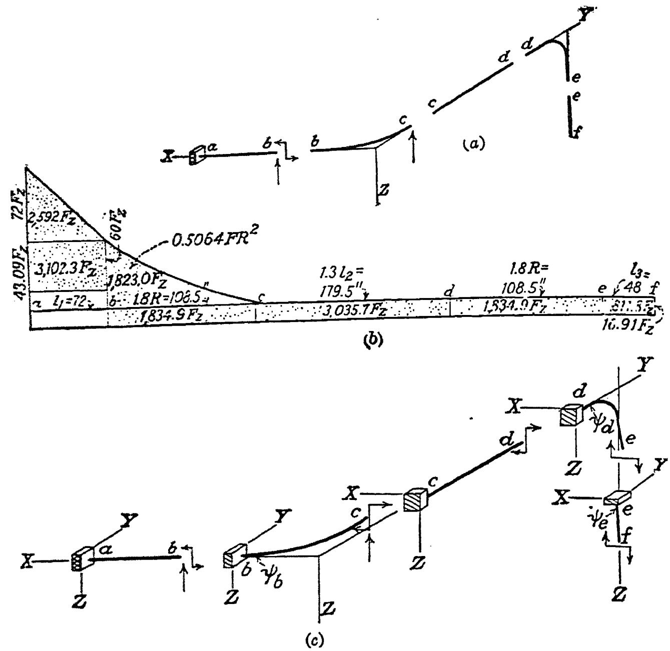


FIG. 27.—Determination of deflections caused by F_Z force in XZ plane.

FIG. 27, F_Z FORCE IN XZ PLANE

Case II.	$EI\Delta_{Z_b} = \Sigma\,\psi_F\tfrac{2}{3}l = 2{,}592F_Z \times 48 = 124{,}416F_Z$ up.	
Case III.	$\psi_M\tfrac{1}{2}l = 3{,}102F_Z \times 36 = 111{,}683F_Z$ up.	
Case I.	$EI\Delta_{Z_c} = \Sigma\,\psi_bR = 5{,}694F_Z \times 60 = 341{,}658F_Z$ up.	
Case VIII(a).	$\psi_{XY}0.754R = 1{,}823F_Z \times 45.24$	$= 82{,}472F_Z$ up.
Case X(c).	$\psi_{M_t}0.280R = 1{,}835F_Z \times 16.80$	$= 30{,}826F_Z$ down.
Case I.	$EI\Delta_{X_e} = \Sigma\,\psi_dR = 2{,}646F_Z \times 60 = 158{,}802F_Z$ right.	
Case IX(c).	$\psi_{MYZ}0.637R = 1{,}835F_Z \times 38.22$	$= 70{,}130F_Z$ left.
Case I.	$EI\Delta_{X_f} = \Sigma\,\psi_el = 812F_Z \times 48 = 38{,}966F_Z$ right.	
Case III.	$\psi_M\tfrac{1}{2}l = 812F_Z \times 24 = 19{,}483F_Z$ left.	

Summation: $EI\Delta_Z = 629{,}403F_Z$ up; $EI\Delta_X = 108{,}155F_Z$ right.

e. The relations between displacements and the F_X force in the XY plane are found from Figs. 28*a*, *b*, *c* and the elementary cases to which the resultant areas of Fig. 28*b* correspond. The

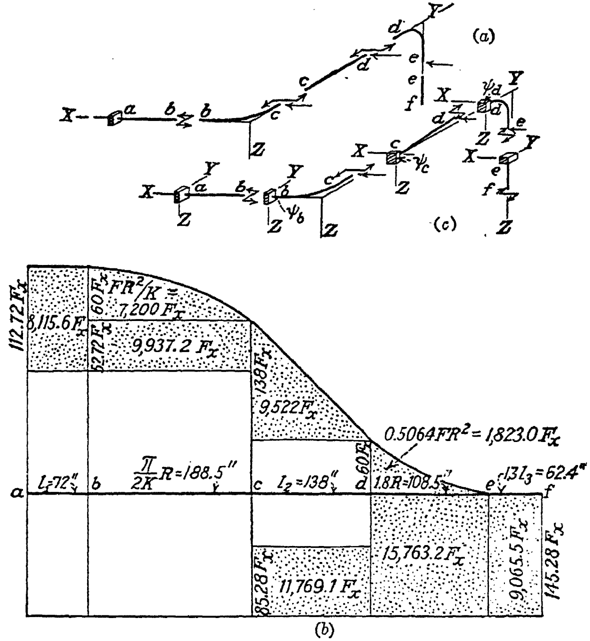


FIG. 28.—Determination of deflections caused by F_X force in XY plane.

abscissas of the moment diagram for the F_X force in the XY plane are identical with those given in Fig. 25*b* for the F_Y force in the XY plane.

FIG. 28, F_Z FORCE IN XY PLANE

Case III. $EI\Delta_{Y_b} = \psi_M \frac{1}{2} l = 8{,}115.6F_X \times 36$
$= 292{,}162F_X$ right.

Case I. $EI\Delta_{Y_c} = \Sigma\, \psi_b R = 8{,}115.6F_X \times 60$
$= 486{,}936F_X$ right.

Case V(*b*). $\psi_{F_r} 0.50R = 7{,}200F_X \times 30$
$= 216{,}000F_X$ right.

Case VII(*b*). $\psi_M 0.363R = 9{,}937.2F_X \times 21.78$

Case I. $EI\Delta_{X_c} = \Sigma\ \psi_b R = 8{,}115.6F_X \times 60$
$= 486{,}936F_X$ left.

Case V(*a*). $= 7{,}200F_X \times 47.12$
$= 339{,}264F_X$ left.

Case VII(*a*). $9{,}937.2F_X \times 38.22$
$= 379{,}800F_X$ left.

Case I. $EI\Delta_{X_d} = \Sigma$ $25{,}253F_X \times 138$
$= 3{,}484{,}886F_X$ left.

Case II. $= 9{,}522F_X \times 92$
$= 876{,}024F_X$ left.

Case III. $11{,}769F_X \times 69$
$= 812{,}068F_X$ right.

Case I. $EI\Delta_{X_e} = \Sigma$ $= 23{,}006F_X \times 60$
$= 1{,}380{,}342F_X$ left.

Case VIII(*a*). $= 1{,}823F_X \times 45.24$
$= 82{,}472F_X$ left.

Case X(*c*). $\psi_{M_t} 0.280R = 15{,}763.2F_X \times 16.80$
$= 264{,}822F_X$ right.

Summation: $EI\Delta_Y = 1{,}211{,}530F_X$ right; $EI\Delta_X$
$= 5{,}952{,}835F_X$ left.

f. The F_X force in the XZ plane is considered in relation to the displacements of the free end of the section of piping in the XZ plane in Figs. 29*a*, *b*, and *c*. The abscissas of the moment diagram are of the same lengths as those found for the F_Z force in the XZ plane.

FIG. 29, F_X FORCE IN XZ PLANE

Case II. $EI\Delta_{Z_b} = \psi_M \tfrac{1}{2} l = 892.4F_X \times 36 = 32{,}126F_X$ down.
Case I. $EI\Delta_{Z_c} = \Sigma\ \psi_b R = 892.4F_X \times 60 = 53{,}544F_X$ down.
Case X(*c*). $\psi_{M_t} 0.280R = 1{,}344.9F_X \times 16.80$
$= 22{,}594F_X$ down.

Case I. $EI\Delta_{X_e} = \Sigma\ \psi_d R = 4{,}462F_X \times 60 = 267{,}732F_X$ left.
Case VIII(*b*). $\psi_{FYZ} 0.754R = 4{,}140F_X \times 45.24$
$= 187{,}294F_X$ left.

Case IX(*c*). $\psi_{MYZ} 0.637R = 5{,}165F_X \times 38.22$
$= 197{,}414F_X$ right.

Case I. $EI\Delta_{X_f} = \Sigma\ \psi_e l = 3{,}437.0F_X \times 48 = 164{,}976F_X$ left.
Case II. $\psi_F \tfrac{2}{3} l = 1{,}152F_X \times 32 = 36{,}864F_X$ left.
Case III. $\psi_M \tfrac{1}{2} l = 4{,}589F_X \times 24$
$= 110{,}136F_X$ right.

Summation: $EI\Delta_Z = 108{,}265F_X$ down; $EI\Delta_X = 349{,}316F_X$ left.

6. The final step in the determination of the magnitude of the reacting forces at the anchors of the typical pipe line in space of Fig. 23 is the collection of terms and determination of the forces from the resulting simultaneous equations. These equations are given below. There is a certain symmetry existing in the relations

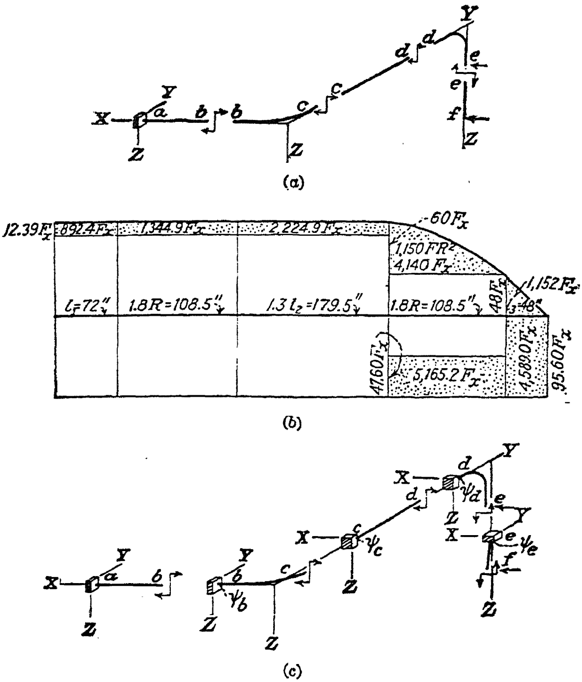


FIG. 29.—Determination of deflections caused by F_X force in XZ plane.

of the three forces which affords a check on the accuracy of the computations. The coefficient determined for the F_Y force in the expression for displacement along the X axis and the coefficient of the F_X force in the expression for displacement along the Y axis are numerically equal. The same equality exists between the coefficient of the F_Z force along the X axis and of the F_X force along the Z axis. The third pair of equivalent coefficients are

for the F_Z force along the Y axis and the F_Y force along the Z axis. These relations are indicated in the equations by underlining.

$$(1)\ EI\Delta_X = \underline{1,211,710F_Y} \text{ right } \underline{\underline{108,155F_Z}} \text{ right } \begin{array}{r} 349,316 \\ \underline{5,952,834} \\ 6,302,150F_X \text{ left.} \end{array}$$

$$(2)\ EI\Delta_Y = \begin{array}{r} 386,574 \\ \underline{635,269} \\ 1,021,843F_Y \text{ left} \end{array} \quad \underline{\underline{981,105F_Z}} \text{ right } \underline{1,211,530F_X} \text{ right.}$$

$$(3)\ EI\Delta_Z = \underline{\underline{981,873F_Y}} \text{ down } \begin{array}{r} 6,069,398 \\ \underline{629,403} \\ 6,698,801F_Z \text{ up.} \end{array} \quad \underline{108,265F_X} \text{ down.}$$

Substituting values of E, Δ, and I from Fig. 39 and Tables I and X and letting deflections to the left and up be designated as positive, the following simultaneous equations are obtained:

$$\begin{array}{llrrrr}
E = 25,000,000,\ I = 361.5 & & & \Delta_X = 0.647,\ EI\Delta_X & = & 5,847,262,500 \\
 & & & \Delta_Y = 1.270,\ EI\Delta_Y & = & 11,477,625,000 \\
 & & & \Delta_Z = 0.530,\ EI\Delta_Z & = & 4,789,875,000 \\
(1) & -1,211,710F_Y & -\ 10 & +6,302,150F_X & = & 5,847,262,500 \\
(2) & +1,021,843F_Y & -\ 98 & & & 11,477,625,000 \\
(3) & -\ 981,873F_Y & +6,698,801F_Z & -\ 108,265F_X & = & 4,789,875,000 \\
(1) & -1,211,710F_Y & -\ 108,155F_Z & +6,302,150F_X & = & 5,847,262,500 \\
(2) & +1,211,710F_Y & -1,163,398F_Z & -1,436,644F_X & = & 13,610,252,000 \\
(A) & & -1,271,553F_Z & +4,865,506F_X & = & 19,457,514,500 \\
(1) & -1,211,710F_Y & -\ 108,155F_Z & +6,302,150F_X & = & 5,847,262,500 \\
(3) & \pm 1,211,710F_Y & \mp 8,266,855F_Z & \pm\ 133,607F_X & = & \mp 5,911,082,769 \\
\hline
(B) & & -8,375,010F_Z & +6,435,757F_X & = & -63,820,269
\end{array}$$

$$\begin{array}{lrrcr}
(A) & -8,375,010F_Z & +32,046,472F_X & = & 128,155,850,871 \\
(B) & \pm 8,375,010F_Z & \mp\ 6,435,757F_X & = & \pm 63,820,269 \\
\hline
 & & +25,610,715F_X & = & 128,219,671,140
\end{array}$$

$$F_X = 5,010; \quad F_Y = 20,870; \quad F_Z = 3,855$$

The resisting moments which act at the free end in each plane may be determined by substituting values of the above forces in the ordinates of the bending-moment diagrams of Figs. 24 to 29, inclusive. These resisting moments at the free end need not be evaluated, however, if it is evident from the diagrams that the point of maximum bending moment does not occur at the free

end, since the moments can be determined at any section directly from the moment diagrams. In this case, the maximum bending moment occurs at the left or fixed end of the section of piping. The manner in which these moments are combined is indicated on Fig. 30. The resisting moments at the free end are also shown in order to afford a comparison between the relative magnitudes of the moments acting at each end of the section of piping. For symmetrically loaded bends, such as shown in Figs. 21 and 22, the bending moments at the ends are equal.

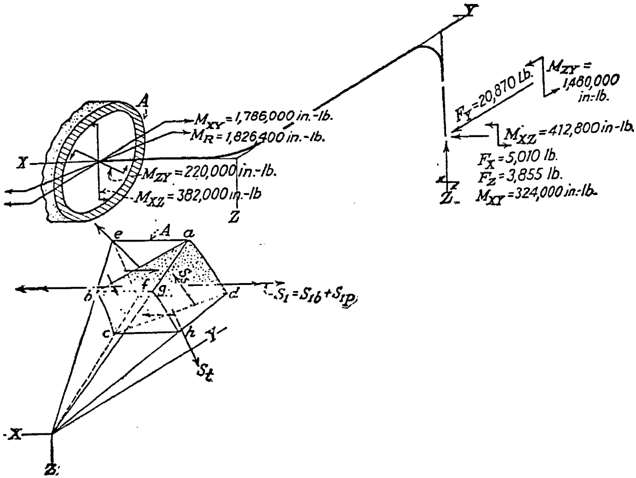


FIG. 30.—Combination of moments, pipe line in space.

If required, the deflections and rotations at the end of each component part in each plane may be found by substituting values of the reacting forces, determined above, in the elementary equations given on Figs. 24 to 29, inclusive (see example page 852).

Stresses Due to Expansion.—The bending moment corresponding to each force may be found at any section, and the stresses due to thermal expansion of the piping determined as illustrated on page 829. The stresses due to thermal expansion which may occur in any section of straight pipe are (1) longitudinal tension and compression and (2) shear stress due to torsional moment. The longitudinal tension is found from the ordinary relation $S_l = My/I = Mr_0/I$, in which M is the bending moment at the sec-

tion, y is the distance from the neutral axis, and I the moment of inertia of the section. The maximum value of y is r_0, the outer radius of the pipe. The torsional shear stress is found from the equation $S_s = 2M_t r_0/\pi(r_0^4 - r_1^4)$, in which M_t is the torsional moment acting at the section, r_0 is the outer radius, and r_1 the inner radius of the pipe. Where the maximum bending moment occurs in a curved section of pipe, the longitudinal and transverse stress-modifying factors found from curves of Figs. 34 and 37 must be taken into account. The large transverse stress induced by flattening of the cross section will, in some cases, be the maximum stress but may be ignored for reasons stated below.

The stresses due to thermal expansion and those due to internal pressure, etc., may be combined as shown on page 824 and the tensile stress equivalent to the total combined stress obtained. The equivalent tensile stress so determined is the strength criterion for which the piping should be designed.

FLATTENING OF THE CIRCULAR CROSS SECTION DURING FLEXURE

Theory.—The theory of flattening of the circular cross section of a curved pipe during flexure has been developed by a number

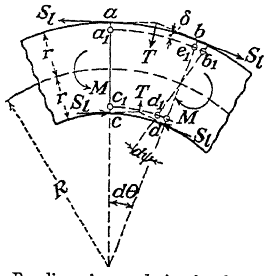


FIG. 31.—Bending of curved pipe in plane of bend.

of writers[1] in explanation of the lack of agreement between ordinary curved-bar formulas and test results on curved pipes. This theory is explained by reference to Fig. 31 and the following

[1] See discussion of Crocker and Sanford paper [footnote 2(a), p. 779] by Dr. S. Timoshenko, with reference to previous article by Th. v. Kármán. Also see papers by William Hovgaard and by A. M. Wahl cited in footnote 2(b) p. 779. In explaining this theory, the present authors have drawn extensively on the analysis of these references.

mathematical analysis. For list of symbols see page 785. Special symbols used in this section are given below:

$d\theta$ = small angle included between two adjacent cross sections ac and bd, before bending.

$d\psi$ = small angle through which section bd rotates during flexure.

M = bending moment at sections ac and bd, in.-lb.

S_l = longitudinal stress due to bending moment M, psi.

T = resultant force, lb, toward the neutral axis of the circular cross section of the pipe caused by the stress S_l acting on a strip of unit width and thickness t.

y = perpendicular distance of any element from the neutral axis, in.

δ = displacement of fiber ab to position a_1b_1, in.

e = extension of fiber ab per unit length of ab.

Assume that the bending moment M acts on the element, causing the cross section originally at bd to move to position b_1d_1 through an angle $d\psi$. The resultant forces T of the tensile stress in the fibers on the outer side of the bend and the compressive stress on the inner side cause these fibers to move toward the neutral axis a distance δ as shown, the fibers ab and cd taking the positions a_1b_1 and c_1d_1, respectively. The extensive e per unit of length of the outer fiber ab thus will be

$$e = \frac{a_1b_1 - ab}{ab} = \frac{a_1e_1 + e_1b_1 - ab}{ab}$$

and

$$\begin{aligned} a_1e_1 &= (R + r - \delta)d\theta. \\ e_1b_1 &= r d\psi \text{ (neglecting small quantities of the higher order).} \\ ab &= (R + r)d\theta. \end{aligned}$$

Substituting these values,

$$e \quad \frac{r d\psi}{(R + r)d\theta} - \frac{\delta}{R + r}.$$

The first term on the right-hand side of the equation is the elongation of the outermost fiber, according to the usual theory of curved bars. The second term represents the decrease in elongation due to flattening of the circular cross section during flexure.

Effect of Flattening on Flexibility.—Owing to the fact that the outermost fibers tend to relieve themselves of stress through flattening of the circular cross section during flexure, there is a

tion, y is the distance from the neutral axis, and I the moment of inertia of the section. The maximum value of y is r_0, the outer radius of the pipe. The torsional shear stress is found from the equation $S_s = 2M_t r_0/\pi(r_0^4 - r_1^4)$, in which M_t is the torsional moment acting at the section, r_0 is the outer radius, and r_1 the inner radius of the pipe. Where the maximum bending moment occurs in a curved section of pipe, the longitudinal and transverse stress-modifying factors found from curves of Figs. 34 and 37 must be taken into account. The large transverse stress induced by flattening of the cross section will, in some cases, be the maximum stress but may be ignored for reasons stated below.

The stresses due to thermal expansion and those due to internal pressure, etc., may be combined as shown on page 824 and the tensile stress equivalent to the total combined stress obtained. The equivalent tensile stress so determined is the strength criterion for which the piping should be designed.

FLATTENING OF THE CIRCULAR CROSS SECTION DURING FLEXURE

Theory.—The theory of flattening of the circular cross section of a curved pipe during flexure has been developed by a number

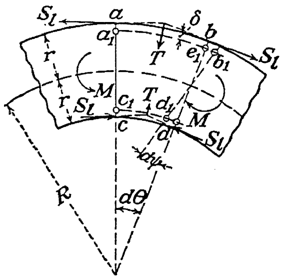


FIG. 31.—Bending of curved pipe in plane of bend.

of writers[1] in explanation of the lack of agreement between ordinary curved-bar formulas and test results on curved pipes. This theory is explained by reference to Fig. 31 and the following

[1] See discussion of Crocker and Sanford paper [footnote 2(*a*), p. 779] by Dr. S. Timoshenko, with reference to previous article by Th. v. Kármán. Also see papers by William Hovgaard and by A. M. Wahl cited in footnote 2(*b*) p. 779. In explaining this theory, the present authors have drawn extensively on the analysis of these references.

mathematical analysis. For list of symbols see page 785. Special symbols used in this section are given below:

$d\theta$ = small angle included between two adjacent cross sections ac and bd, before bending.

$d\psi$ = small angle through which section bd rotates during flexure.

M = bending moment at sections ac and bd, in.-lb.

S_l = longitudinal stress due to bending moment M, psi.

T = resultant force, lb, toward the neutral axis of the circular cross section of the pipe caused by the stress S_l acting on a strip of unit width and thickness t.

y = perpendicular distance of any element from the neutral axis, in.

δ = displacement of fiber ab to position a_1b_1, in.

e = extension of fiber ab per unit length of ab.

Assume that the bending moment M acts on the element, causing the cross section originally at bd to move to position b_1d_1 through an angle $d\psi$. The resultant forces T of the tensile stress in the fibers on the outer side of the bend and the compressive stress on the inner side cause these fibers to move toward the neutral axis a distance δ as shown, the fibers ab and cd taking the positions a_1b_1 and c_1d_1, respectively. The extensive e per unit of length of the outer fiber ab thus will be

$$e = \frac{a_1b_1 - ab}{ab} = \frac{a_1e_1 + e_1b_1 - ab}{ab}$$

and

$$a_1e_1 = (R + r - \delta)d\theta.$$
$$e_1b_1 = rd\psi \text{ (neglecting small quantities of the higher order).}$$
$$ab = (R + r)d\theta.$$

Substituting these values,

$$e = \frac{rd\psi}{(R + r)d\theta} - \frac{\delta}{R + r}.$$

The first term on the right-hand side of the equation is the elongation of the outermost fiber, according to the usual theory of curved bars. The second term represents the decrease in elongation due to flattening of the circular cross section during flexure.

Effect of Flattening on Flexibility.—Owing to the fact that the outermost fibers tend to relieve themselves of stress through flattening of the circular cross section during flexure, there is a

lessening in the resistance offered by the bend to flexure which is equivalent to a reduction in the moment of inertia I of the pipe section. Consequently, in dealing with a curved pipe, it is necessary to consider, not the apparent flexural rigidity EI, but a value of flexural rigidity KEI, where K is a factor, less than unity (which is called the rigidity multiplication factor), to be determined from the relation[1]

$$K = \frac{1 + 12h^2}{10 + 12h^2} \text{ where } h = \frac{tR}{r^2}.$$

Thus it is seen that the effect of flattening on flexural rigidity depends only on the magnitude of the quantity $h = tR/r^2$. Where the thickness of the pipe t or the radius of the curved pipe R is large, as compared with the diameter of the pipe, the value of the factor K approaches unity and ordinary bending formulas apply. Often, however, the factor K is much less than unity. For example, with 10-in. standard-weight pipe bent to a radius of six times the nominal pipe diameter, as is common practice, $h = 0.8$, and the factor K is approximately 0.5. In other words, the deflections caused by a force F, applied as in Cases V, VI, and VII on pages 790 and 791, will be twice that indicated by ordinary curved-bar formulas.

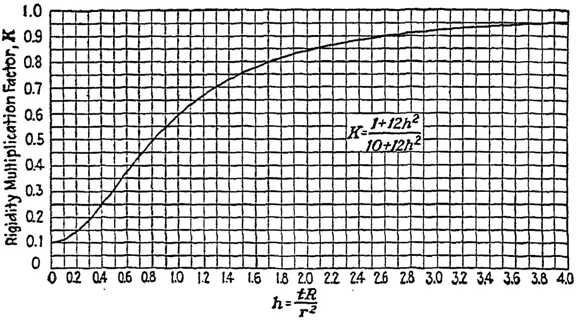


FIG. 32.—Curve for determining rigidity multiplication factor K.

Values of K plotted against corresponding values of h are given in Fig. 32 for convenience in solving problems dealing with curved pipe.

[1] For a derivation of this relation, see paper by Th. v. Kármán, *Zeit. V. D. I.* 1911, p. 1889, or William Hovgaard, in footnote 2(*b*), p. 779.

Longitudinal Stresses Due to Bending.—In computing stresses in curved pipes caused by moments acting in the plane in which the curved pipe lies, ordinary curved-bar formulas are at fault since they do not consider the effect which flattening of the circular cross section during flexure has on the distribution of longitudinal stress. Because of the tendency of the fibers farthest removed from the neutral axis to shirk taking their normal share of the bending stress, the distribution of longitudinal stress around the outer wall of the pipe cross section for the case in which the pipe-bend ratio $h = tR/r^2 = 0.8$ is somewhat as shown in Fig. 33. This case is typical of American commercial pipe-bend practice

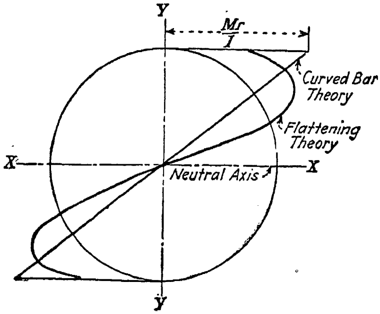


FIG. 33.—Longitudinal-stress distribution in outside wall of curved pipe, $h = tR/r^2 = 0.8$. Tensile stress shown to right of Y axis, compressive to left.

and corresponds to 10-in. standard-weight pipe bent to a radius of six pipe diameters. The straight-stress line of Fig. 33 represents longitudinal-stress distribution in accordance with the usual curved-bar theory. The longitudinal stress in the outer wall at any point y distant from the neutral axis is given by the equation[1]

$$S_l = \frac{My}{KI}\left[1 - \frac{6y^2}{r_0{}^2(5 + 6h^2)}\right]$$

in which y = distance of any element from the neutral axis, inches and S_l = longitudinal stress, pounds per square inch.

Stress at any intermediate point in the wall can be computed by substituting the corresponding radius for r_0.

Solution for the longitudinal stress at the point most distant from the neutral axis, which is less than at the maximum point

[1] *Ibid.*

shown in Fig. 33, is made by substituting r_0 for y and simplifying the above equation as follows:

$$S_l = \frac{Mr_0}{KI}\left[1 - \frac{6}{5 + 6h^2}\right].$$

According to Prof. Hovgaard, however, the most important bending stress due to expansion is the longitudinal stress in the mid-section of the pipe wall at the maximum point of the curve shown in Fig. 33. This stress is obtained from the formula

$$S_{l_{max}} = \beta M r$$

where

$$\beta = \frac{2}{3K}\sqrt{\frac{6h^2 + 5}{18}} \text{ if } h < 1.472$$

$$\beta = \frac{1}{K}\left(\frac{6h^2 - 1}{6h^2 + 5}\right) \text{ if } h > 1.472$$

Values of the maximum longitudinal-stress factor β, plotted against values of h, are given in Fig. 34 for convenience in calculat-

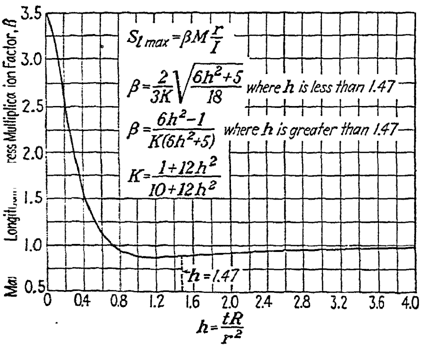


FIG. 34.—Stress multiplication factor β for maximum longitudinal stress in mid-fiber of wall, curved pipe.

ing this stress in curved pipe. As explained below, under "Total Combined Stress in Piping," where this maximum longitudinal bending stress as computed for the mid-point in the wall of a curved pipe exceeds the transverse bursting stress, it should be combined with any shearing stress present and with the longitudinal stress due to internal pressure, to obtain the total equivalent stress.

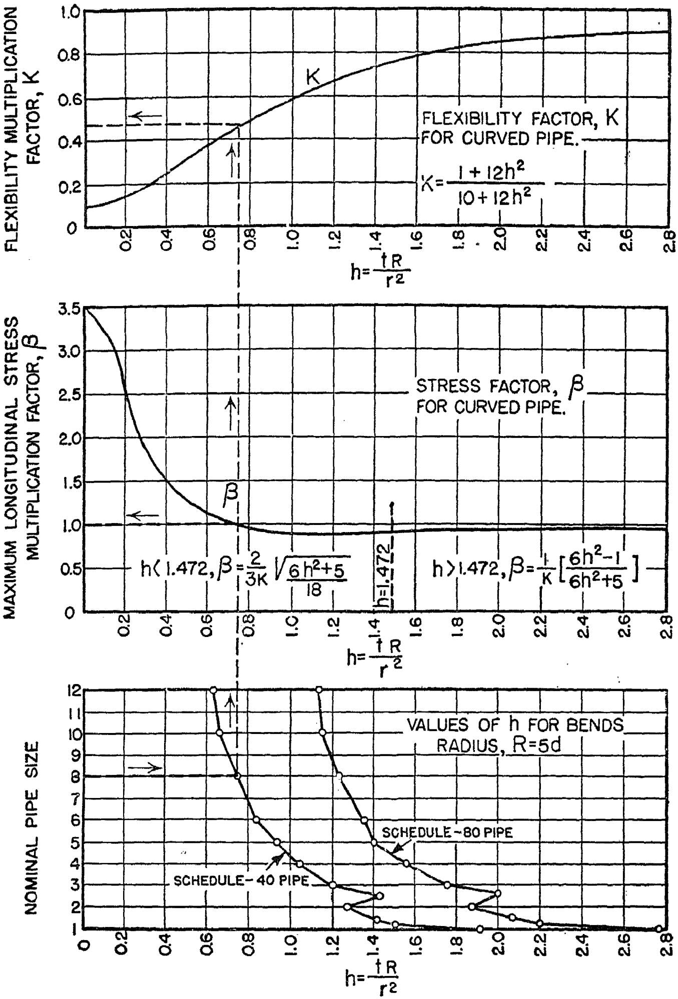


FIG. 35.—Chart for determining flexibility factor K and stress multiplication factor β for curved pipe. Direct solution given for Schedule 40 and Schedule 80 pipe formed to an R/d ratio of 5. Dashed guide line illustrates determination of β and K for 8-in. Schedule 40 pipe (see example on page 851).

Where bending stress due to dead weight between supports is of serious magnitude, it also should be included in the combination.

Values of K and β for Schedules 40 and 80 Pipe.—Numerical values of K and β for nominal sizes of Schedules 40 and 80 pipe bent to a radius of $R = 5d$ can be read directly from Fig. 35 without having to compute the value of h.

Transverse Stresses Due to Bending.—Flattening of the cross section of a curved pipe, caused by bending in the plane in which the curved pipe lies, induces transverse stresses in the pipe cross section. These stresses are compressive on the outside of the pipe wall at the point most distant from the neutral axis and tensile on the inside of the pipe wall at this point. The

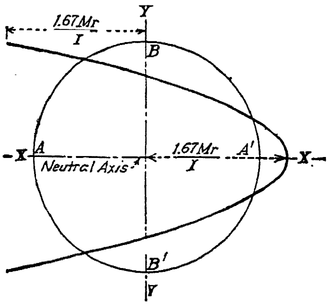


FIG. 36.—Transverse-stress distribution in outside wall of curved pipe, $h = tR/r^2 = 0.8$. Tensile stress shown to right of Y axis, compressive to left.

distribution of these stresses for the outside of the pipe wall is illustrated in Fig. 36 for a curved pipe where $h = 0.8$. The transverse stress on the outside of the pipe wall at any point y distant from the neutral axis of the cross section, induced by flattening of the cross section, is given by the formula[1]

$$S_t = \frac{18Mr_0h\left(1 - \frac{2y^2}{r^2}\right)}{I(1 + 12h^2)}.$$

Transverse bending stress on the inner wall can be computed by substituting r_1 for r_0 and noting that tensile and compressive stresses are reversed. This stress is zero at approximately the mid-point in the pipe wall.

[1] For derivation of this formula, see Appendix 3 of the paper by A. M. Wahl, footnote 2(c), p. 779.

The transverse stress becomes a maximum at the point most distant from the neutral axis and at the neutral axis where $y = r_0$ and $y = 0$, respectively. It is given by the equation

$$S_t = \gamma \frac{Mr_0}{I}$$

where $\gamma = \frac{18h}{1 + 12h^2}$.

Values of the transverse-stress multiplication factor γ, plotted against values of h, are given in Fig. 37 for convenience in calculat-

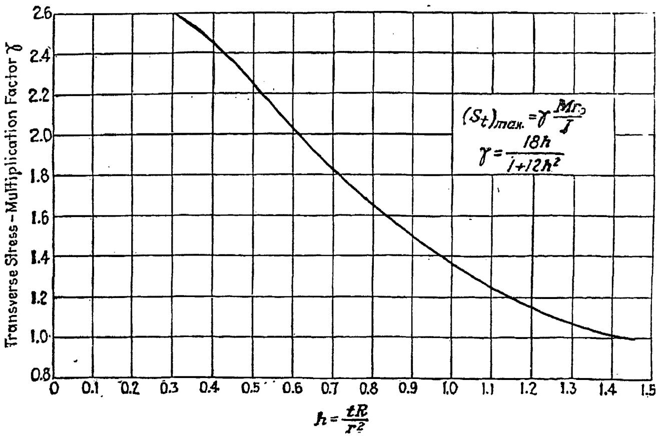


FIG. 37.—Curve for computing maximum transverse stress in curved pipe.

ing the maximum transverse stress on the outside of the pipe wall induced by flattening of the cross section of the curved pipe due to flexure in the plane of the bend.

For cast-iron pipes where the curvature is such that h is less than 1.5, the maximum transverse stress found from Fig. 37 is the determining stress for which the piping should be designed. Where h is greater than 1.5, the longitudinal stresses are numerically greater than the transverse stresses and approach the stresses found from ordinary curved-bar formulas. Where h is greater than 1.5, therefore, the determining stress for cast-iron pipes subjected to a bending moment is found from the usual equation

$$S = \frac{Mr_0}{I}.$$

In the case of pipe made of steel or other ductile metals, the transverse stress found from Fig. 37 frequently exceeds the yield point of the material. This condition is relieved by plastic flow under load and is not of serious consequence due to the very localized nature of the stress. For this reason the transverse stress due to bending can be neglected safely when combining stresses due to bending, shear, and internal pressure (see paper by William Hovgaard, "Test on High Pressure Pipe Bends," *J. Math. Phys., Mass. Inst. Tech.*, Vol. 8, No. 4, pp. 293–344, December, 1929). Methods are given below for combining all stresses involved for the purpose of determining the total equivalent stress at any point in a pipe line.

TOTAL COMBINED STRESS IN PIPING

Longitudinal Stress Due to Internal Pressure.[1]—According to the Code for Pressure Piping (Sec. 620g) the longitudinal stress due to internal pressure, S_2 (see also page 34), shall be determined by dividing the end force due to internal pressure by the cross-sectional area of the pipe wall, *viz.*,

$$S_2 = \frac{p\pi d^2/4}{\pi(D^2 - d^2)/4} = \frac{pd^2}{D^2 - d^2}$$

where p = internal pressure, psi.
D = nominal outside diameter of pipe, in.
d = D minus twice the nominal wall thickness, in.

Transverse Stress Due to Internal Pressure.—The transverse stress due to internal pressure is a hoop tension that is constant across the pipe wall thickness. This stress may be found from the relation $S_1 = pd/2t$ (see page 33), in which S_1 = transverse stress and other notations are as above.

Longitudinal Stress Due to Weight of Piping.—The longitudinal stress due to bending caused by dead weight of piping between supports may be found by incorporating the moments due to weight of piping directly in the diagrams used to determine the relations of deflection and reacting forces (see pages 794 to 816, inclusive), in which case the stress due to inherent weight of piping between supports is combined with that due to expansion. It is more convenient in most cases, however, to compute dead-

[1] See Chap. II, pp. 31 to 47 for complete discussion of stresses caused by internal pressure.

weight stresses independently and subsequently combine them with the stresses due to internal pressure and thermal expansion. Stresses due to weight of piping may be found by the approximations given in Fig. 13 and Table VI, Chap. VI (pages 747 to 753).

Except in the case of a single span, the maximum bending moment due to weight of piping given by these approximate equations occurs at a support and causes a tension in the upper side of the pipe. In most cases the stress due to weight of piping has a minor effect on the total stress, often tending to decrease rather than augment the total combined stress.

Longitudinal Stress Due to Direct Thrust.—The longitudinal stress due to direct thrust is a compression given by the relation $S = F/A$, in which S = stress, F = force along axis of pipe, and A is the cross-sectional area of pipe wall. This stress is of minor importance and usually can be ignored.

Shear Stress Due to Torsional Moment.—The shear stress at any section due to a torsional moment may be found from the relation[1] $S_s = 2M_t r_0/\pi(r_0^4 - r_1^4) = M_t r_0/2I$, in which S_s = shear stress, M_t = torsional moment, r_0 = the outer radius of the pipe cross section, and r_1 = inner radius. This shear stress may be combined conveniently with the longitudinal stresses due to bending and internal pressure, or with the transverse stress due to internal pressure by means of the chart of Fig. 38.

Tensile Stress Equivalent to Maximum Shear Stress.—In determining the strength of pipe made of steel, wrought iron, copper, brass, bronze, or other ductile material when subjected to the action of a combination of stresses, the maximum shear (Guest) theory[2] usually is employed which holds that, when a ductile material is subjected to the action of two or three stresses on mutually perpendicular planes, failure will occur only when the maximum tangential stress reaches a certain limiting value. The maximum value of the tangential stress is one-half the difference of the maximum and the minimum tensile stresses acting on the element under consideration (compressive stresses are considered as negative tensile stresses). The tensile stress equivalent to the maximum shear stress is equal to twice the shear stress or to the difference of the maximum and minimum stresses. The maximum equivalent tensile stress occurs on plane *abcd* of Fig. 38 for the particular case in which torsional stress acts in

[1] See "Strength of Materials," *op. cit.*, p. 166.

[2] See "Strength of Materials," *op. cit.*, p. 135.

combination with tensile stresses as shown. Its approximate value is given by the equation

$$S_{eq} = 2\sqrt{S_s^2 + \left(\frac{S_{max} - S_{min}}{2}\right)^2}$$

in which S_{eq} is the equivalent tensile stress corresponding to stress in simple tension, S_s = shearing stress due to torsional moment, S_{max} = the maximum tensile stress which acts on the element, and S_{min} = the minimum stress which exists on a plane perpendicular to the plane on which the maximum stress acts. The stress on the third plane which is perpendicular to the above planes has no effect on the maximum equivalent combined stress. The above equation is represented graphically in Fig. 38. It is evident from inspection of the infinitesimal element shown in Fig. 38 that the intermediate stress, which in the case of the problem illustrated in Fig. 23 is the transverse bursting stress, can have no effect on the stress in the plane along which maximum shear occurs. Where the stress due to bending is small, the transverse bursting stress due to internal pressure may become the maximum stress, and shear will occur along a plane parallel to the longitudinal stress and inclined at an angle of approximately 45 deg to the direction of the transverse stress.[1]

Actually, the particular plane along which the maximum shear occurs need not concern the piping designer, as all that is necessary is to determine: (1) S_{max} the maximum stress that acts at the particular point under investigation, which usually will be either the sum of the longitudinal stresses due to bending and to internal pressure or the transverse bursting stress; (2) S_{min} the minimum stress that acts on some plane perpendicular to the plane on which the maximum stress acts.

The minimum stress ordinarily occurs in a plane perpendicular to the radius of the pipe and has a value numerically equal to the internal fluid pressure at the inner surface of the pipe wall, decreasing to zero at the outer surface.[1] With low or moderate pressures, this stress is small in comparison with other stresses involved and can be neglected by placing $S_{min} = 0$ in the above equation. In exceptional cases the minimum stress will be transverse bending

[1] See "Strength of Materials," by S. Timoshenko, Part I, 2nd ed., p. 43. D. Van Nostrand Company Inc., 250 4th Ave., New York, N.Y.

stress in compression (negative tension in the formula), induced by flattening of the circular cross section of a curved pipe during flexure. The compressive nature of this stress reverses the negative sign preceding S_{min} in the equation and results in the

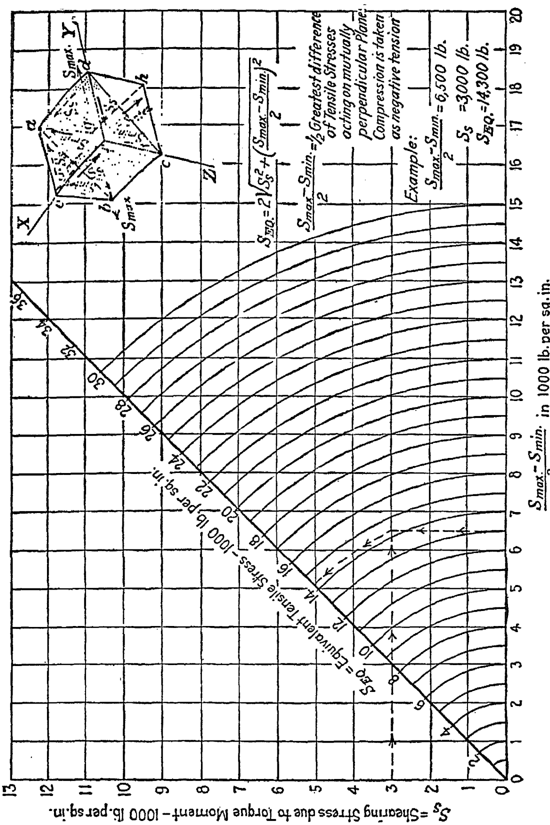


FIG. 38.—Chart for combining torsional shear with bending and bursting stresses in pipe wall to obtain the equivalent tensile stress.

arithmetical addition of S_{max} and S_{min}, which may give an excessively high value for S_{eq}. Owing to the very localized nature of the transverse bending stress existing in curved pipe, it tends to relieve itself by plastic flow and may be neglected as explained at the top of page 824.

The planes along which maximum shear occurs are discussed in detail above because of the tendency often evidenced in analysis of complex stresses to combine stresses regardless of the manner in which they act.

The maximum and minimum normal stresses and the torsional stress, which are determined as acting on an element of the pipe wall at a particular point, may be combined directly by means of Fig. 38 to obtain the equivalent combined tensile stress to which the material is subjected. The stress so determined should not exceed the allowable working stress in simple tension. A total equivalent stress of 15,000 psi is considered satisfactory for mild steel pipe.

NOTE.—In the absence of torsion, the combining of bending and pressure stresses to obtain the tensile stress equivalent to the maximum shear stress reduces to the simple procedure of adding together the longitudinal bending stress (tension) and longitudinal pressure stress as shown in the Solution of Sample Problems on pages 849 to 857.

Distinction between Bursting Stress and Bending Stress.—In considering values of allowable stress in piping design, a distinction should be made between bursting stress, which is continuous and, hence, results in a gradual reduction in wall thickness, and bending stress caused by constraint of thermal elongation of the line. In the latter case, if the stress is greater than can be supported, the line simply conforms to a shape giving lower stress. When the line is cooled down, reverse stresses will be found equal to the difference between the stress obtained in the initial design and the stress that could be supported by the material.[1] If a joint is unbolted after a period of high-temperature service, this reduction in bending stress through creep will manifest itself by an increase in the amount of cold spring. In other words, creep under high-bending stress tends, at elevated temperatures, to reduce the thrusts and bending moments caused by constraint of thermal elongation in exactly the same manner as cutting the pipe short and pulling it together during assembly.

The disappearance of cold spring sometimes observed when unbolting steam-line joints is attributable to method of erection and support, shifting of so-called "anchor points," swiveling of

[1] For a good discussion of the "Design Aspect of Creep," see paper by R. W. Bailey (*Journal of Applied Mechanics*), *Trans. ASME*, Vol. 3, p. A1, March, 1936.

loose flanges, etc., rather than to creep of the pipe. A comparison of calculated cold-spring allowances and the residual cold spring measured at a number of joints in the old main-steam piping at the Conners Creek Plant of The Detroit Edison Company, after some twenty years of service at 600 F, indicated that the effect of cutting the pipe short had survived the vicissitudes of erection and the years of service. Because of the low operating temperature, creep in this case could not have been expected to influence the amount of residual cold spring to any extent.

Change in Elasticity with Temperature.—The modulus of linear elasticity E is found by dividing unit tensile stress by unit strain $E = S/e$. The modulus of rigidity, or modulus of shearing elasticity G, is found by dividing unit shear stress by unit shear strain $G = S_s/e_s$. The two moduli are related by the expression $G = \frac{E}{2(1 + \lambda)}$, in which λ is the coefficient of lateral contraction or Poisson's ratio. Values for different materials are given in Table I on page 344.

The curves for E, G, and λ given on Fig. 39 for carbon steel are based on static tensile and torsional tests by Guy Versé.[1] Dynamic tests gave values of E at 850 F about 5 per cent greater than static tests and are theoretically more accurate. However, since some plastic strain will occur in any practical application at high temperatures, it seems preferable to use values based on static tests. The values for carbon steel are in reasonably good agreement with data by Orrok,[2] Tapsell,[3] and other investigators.

The curve for carbon molybdenum steel is based on a large number of short-time high-temperature tensile tests using recording strain gages. These tests were made in connection with an investigation by White and Crocker.[4]

Stress in Pipe Line in Space.—The maximum stress in the pipe line illustrated in Fig. 23 occurs at the fixed end a. It is composed of (1) longitudinal tensile stress due to bending, (2) shear stress due to torsional moment, and (3) longitudinal tensile stress

[1] "The Elastic Properties of Steel at High Temperatures," by Guy Versé, *Trans. ASME*, Vol. 57, pp. 1–4, 1935.

[2] "High Pressure Steam Boilers," by G. A. Orrok, *Trans. ASME*, Vol. 50, No. 15, p. 47.

[3] Creep of Metals," by H. J. Tapsell, Oxford University Press, New York, 1931, p. 28.

[4] "Effect of Grain Size and Structure on Carbon-molybdenum Steel Pipe," by A. E. White and Sabin Crocker, *Trans. ASME*, Vol. 63, p. 749, 1941.

due to internal pressure. The transverse tensile stress due to internal pressure, which also exists at this point, is less than the sum of the longitudinal stresses, but greater than the stress that exists on the plane perpendicular to the planes on which the longitudinal and transverse stresses due to internal pressure act, *i.e.*, $S_l > S_t > S_0$. These stresses are shown acting on an element of the pipe cut from the section at the fixed end of the pipe in Fig. 30.

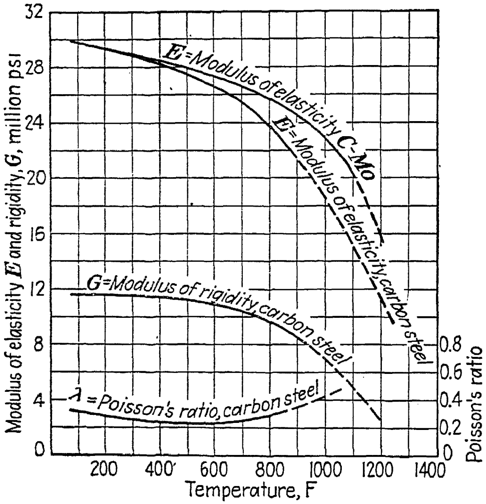


FIG. 39.—Change in modulus of elasticity, modulus of rigidity, and Poisson's ratio with temperature.

The longitudinal stress due to bending and the shear stress due to torsion, which exists at the fixed end of the pipe line illustrated in Fig. 23, are found from the moments obtained by substituting values of F_X, F_Y, and F_Z determined on page 814 in the ordinates of the moment diagrams of Figs. 24 to 29, inclusive. The magnitude and direction of the moments at a are shown in Fig. 30. The moments acting in each plane are combined, giving a bending moment of 1,786,000 in-lb in the XY plane, a bending moment of 382,000 in-lb in the XZ plane, and a torsional moment of 220,000 in-lb in the ZY plane. The bending moments which act in the XY and XZ planes are combined vectorially to

give a resulting bending moment of 1,826,400 in-lb in a plane which makes an angle α of approximately 12 deg with the XY plane. The magnitude of α is found from $\tan \alpha = 382/1{,}786 = 0.2140$.

$$\alpha = 12^\circ\, 08'.$$

The longitudinal tensile stress corresponding to the above bending moment is

$$S_l = \frac{Mr_0}{I} = \frac{1{,}826{,}400 \times 6.375}{361.5} = 32{,}200 \text{ psi.}$$

The shearing stress at the same point is found from the equation given on page 825 as

$$S_s = \frac{2M_t r_0}{\pi(r_0^4 - r_1^4)} = \frac{2 \times 220{,}000 \times 6.375}{\pi[(6.374)^4 - (5.875)^4]} = 1{,}940 \text{ psi.}$$

or where the moment of inertia I is known (see Table X, page 858)

$$S_s = \frac{M_t r_0}{2I} = \frac{220{,}000 \times 6.375}{2 \times 361.5} = 1{,}940 \text{ psi.}$$

The longitudinal stress due to internal pressure (see page 824) can be found conveniently by dividing the product of the internal pressure and the internal cross-sectional area of the 12-in., 0.50-in. wall pipe by the cross-sectional area of the metal. Both areas are given in Table II, page 358,

$$S_2 = \frac{400 \times 108.43}{19.24} = 2{,}254 \text{ psi.}$$

The transverse stress due to internal pressure is

$$S_1 = \frac{pd}{2t} = \frac{400 \times 11.75}{2 \times 0.50} = 4{,}700 \text{ psi.}$$

The total longitudinal stress from the above is 32,200 + 2,254 = 34,454 psi. The shear stress is 1,940 psi. and the transverse stress is a tension of 4,700 psi. The radial stress due to internal pressure is zero at the outside of the pipe wall and, hence, is the minimum stress acting on the element of the pipe wall at the end a.

$$\frac{S_{max} - S_{min}}{2} = \frac{34{,}454 - 0}{2} = 17{,}227 = \text{abscissa of Fig. 38.}$$

The ordinate is $S_s = 1{,}940$. The equivalent tensile stress is found from the diagonal scale of Fig. 38 as 34,670 psi.

The pipe material employed for 400-lb 725 F service corresponds to ASTM standard specifications for lap-welded and seamless-steel pipe for high-temperature service serial designation A106 for Grade *B* seamless pipe (see page 378). It has a minimum yield-point requirement of 35,000 psi.

The combined stresses due to bending and internal pressure for power piping is required by the Code for Pressure Piping, Par. 620 g, not to exceed three-fourths the sum of the *S* value for the piping material at 650 F and the *S* value at nominal operating temperature.

For Grade B, ASTM A106 pipe at an operating temperature of 725 F, the allowable combined stress $= \frac{3}{4}(S_{650F} + S_{725F}) = \frac{3}{4}(12{,}000 + 10{,}900) = 17{,}175$ psi. See Table V, page 43, for *S* values for Power Piping.

On the basis of the above considerations, the stress in the pipe line of Fig. 23 is double that allowable in conservative piping design. This illustration indicates the necessity for a rigorous analysis of the flexibility of piping subjected to temperature change.

SIMPLIFIED SOLUTIONS

With relatively simple structures it often is possible to obtain reasonably accurate answers through simplified solutions or the use of precalculated results read from tables or graphs.[1] Simplified solutions frequently are as accurate as circumstances justify, especially in view of the extensive labor required for exact solutions.

[1] The following publications give additional precalculated and simplified solutions:

(*a*). "A Manual for the Design of Piping for Flexibility by the Use of Graphs," by E. A. Wert, S. Smith, E. T. Cope, The Detroit Edison Company, Detroit, 1934.

(*b*). "Design Charts for Piping Flexibility," by E. A. Wert and S. Smith, *Catalogue* 34-1, Taylor Forge and Pipe Works, Chicago, 1934.

(*c*). "Pittsburgh Piping Design Manual," by E. A. Wert, S. Smith, G. Sinding Larsen and George W. Petrie, Jr., Pittsburgh Piping and Equipment Company, Pittsburgh, Pa., 1935.

(*d*). "Design of Piping for Flexibility with Flex-Anal Charts," by E. A. Wert and S. Smith, Power Piping Division, Blaw-Knox Co., Pittsburgh, Pa., 1941.

(*e*). "Piping Flexibility," by Arthur McCutchan, *Catalogue* 42, The Walworth Company, New York, 1942, pp. 434–443.

(*f*). "Frictional Resistance and Flexibility of Seamless Tube Fittings Used in Pipe Welding," by Sabin Crocker and Arthur McCutchan, *Trans. ASME*, 1931, pp. 215–245, FSP 53-17.

Tabular Solutions.—A convenient solution for *conventional expansion bends* used with straight runs of pipe is furnished in Tables VI to IX which give the over-all lengths of pipe cared for by various radii of bends and the corresponding reacting forces at different operating temperatures. Bends should be located midway between anchor points, or at least within the middle third of the span, to agree with the assumptions made in preparing these tables.

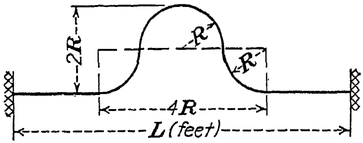
The over-all lengths given in Tables VI and VIII for expansion U bends and double-offset expansion U bends, respectively, are strictly accurate for Schedule 80 (extra-strong) pipe. But these tables may be used with reasonable accuracy for all pipe thickness schedules, since bending stress is independent of the wall thickness in straight pipe and is affected only to a minor degree in the case of long-radius bends by variations in the factors that correct for flattening of the circular cross section of curved pipe during flexure. For wall thicknesses less than Schedule 80, the lengths given in Tables VI and VIII are conservative, but not unduly so, because of the relatively long-radius bends considered in these tables. For thickness greater than Schedule 80, the lengths tabulated are slightly greater than found by a strict analysis, but not significantly greater.

The lengths given in Tables VI and VIII are based on an assumed maximum longitudinal bending stress of 12,000 psi and no credit for cold spring. If, for example, it is desired to permit a bending stress of 16,000 psi, it is only necessary to multiply the lengths given in the tables by $^{16}\!/_{12}$. Where pipe is cut short by 50 per cent or more of the computed expansion and sprung into position, the Code for Pressure Piping permits a maximum credit of one-third the computed expansion. The adjustment of lengths to take this maximum credit into account is simply effected by multiplying the tabulated lengths by 1.5. When adjusting for stresses above 12,000 psi, or when taking credit for cold spring, the resulting lengths are somewhat conservative, while for stresses below 12,000 psi the lengths thus found are slightly longer than given by a strict analysis. In neither case are the inaccuracies significant within the usual range of stress variation encountered.

Where pipe is designed in accordance with the Code for Pressure Piping, the permissible bending stress is obtained by subtracting the longitudinal pressure stress from the total allowable combined

TABLE VI.—EXPANSION U BENDS, ALL PIPE-THICKNESS SCHEDULES

(Over-all lengths of pipe in feet permissible with expansion U bends of radii and for operating temperatures shown)



Nominal pipe size, inches d_n	$\frac{R}{d_n}$	Mean radius of bend, inches R	L = over-all length of pipe in feet permissible with expansion U bends of radius indicated (Based on maximum longitudinal bending stress of 12,000 psi)						
			200 F	300 F	400 F	500 F	600 F	700 F	800 F
2	6	12	28.6	15.9	10.8	7.9			
	8	16	45.8	25.7	17.6	13.1			
	10	20	65.0	36.5	24.9	20.3			
2½	6	15	37.0	20.6	13.9	10.4			
	8	20	59.5	33.3	22.9	17.3			
	10	25	83.5	47.5	31.7	24.5			
3	6	18	46.7	26.1	17.5	13.2	10.7		
	8	24	72.0	40.3	28.1	20.8	16.7		
	10	30	102.0	58.5	40.0	31.0	25.0		
4	6	24	67.0	37.5	25.6	19.0	15.3		
	8	32	100.0	57.2	39.6	29.6	23.7		
	10	40	141.0	80.0	55.0	42.5	34.6		
6	6	36	100.7	60.0	41.2	30.4	24.1		
	8	48	158.3	90.5	61.7	46.7	37.1		
	10	60	207.0	118.0	81.0	63.0	51.0		
8	6	48	150.0	84.2	58.0	42.9	34.6		
	8	64	221.0	124.2	86.7	66.2	53.3	44.2	
	10	80	301.0	170.0	116.0	88.0	71.0	58.0	
10	6	60	200.0	112.1	72.1	57.9	45.8	38.7	
	8	80	297.0	167.2	115.7	86.7	69.2	59.2	51.7
	10	100	400.0	225.0	155.0	120.0	98.0	83.0	72.0
12	6	72	243.0	136.8	93.3	70.0	55.8	47.5	40.8
	8	96	357.0	201.0	139.2	104.2	84.2	71.7	63.0
	10	120	485.0	282.0	190.0	143.0	118.0	100.0	85.0
14 O.D.	6	84	290.0	168.2	115.2	86.3	69.2	57.9	51.7
	8	112	416.0	243.0	171.6	130.0	104.2	89.2	77.5
	10	140	615.0	345.0	234.0	178.0	144.0	127.0	105.0

TABLE VI.—(*Concluded*)

Nominal pipe size, inches d_n	$\frac{R}{d_n}$	Mean radius of bend, inches R	L = over-all length of pipe in feet permissible with expansion U bends of radius indicated (Based on maximum longitudinal bending stress of 12,000 psi)						
			200 F	300 F	400 F	500 F	600 F	700 F	800 F
16 O.D.	5	80	252.2	147.1	100.3	74.6	59.6	50.4	43.8
	6	96	327.0	191.0	132.0	98.8	79.7	67.5	58.7
	7	112	392.0	225.0	157.0	118.0	95.0	80.0	70.0
18 O.D.	5	90	289.0	168.4	115.5	85.8	68.4	57.6	50.4
	6	108	378.0	220.0	152.2	114.2	91.7	84.8	74.7
	7	126	460.0	258.0	175.0	133.0	106.0	90.0	79.0
20 O.D.	5	100	318.0	185.5	127.5	94.2	75.8	63.3	55.0
	6	120	414.0	241.5	169.2	126.3	101.6	85.8	75.4
	7	140	488.0	280.0	195.0	148.0	118.0	100.0	88.0
24 O.D.	5	120	375.0	218.5	157.5	115.5	92.5	78.0	68.0
	6	144	517.0	302.0	213.0	156.6	126.3	107.2	94.2
	7	168	610.0	357.0	252.0	195.0	126.0	136.2	120.0

Table is based on Fig. 41. Pipe is assumed to be installed at 60 F. No credit is taken for cold spring in establishing tabulated pipe lengths. See text for method of adjusting lengths for bending stresses above or below 12,000 psi and of taking into account partial or full credit for cold spring. Lengths are based on values of K and β for Schedule 80 pipe but are sufficiently accurate for all pipe thickness schedules.

Example.—What length of 16-in. O.D. pipe can be used under the following conditions with an expansion U bend having a radius of 6-pipe diameters if the line is installed at 60 F and the operating temperature is 400 F?

(*a*) For the basic conditions of Table VI?

(*b*) If the allowable bending stress is 16,000 psi instead of 12,000 psi?

(*c*) If credit is taken for cold springing the line?

(*d*) If advantage is taken of both (*b*) and (*c*)?

Solution.—(*a*) The length of line L read from Table VI is 132 ft.

(*b*) For a bending stress of 16,000 psi the length would be $132 \times {}^{16}\!/_{12} = 176$ ft.

(*c*) The maximum credit for cold spring allowed under the Code for Pressure Piping (see text page 778) would increase the basic length to $132 \times 1.5 = 198$ ft.

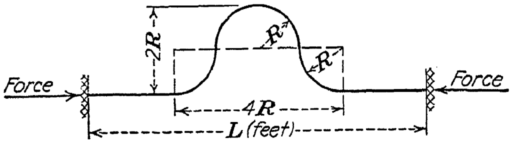
(*d*) If advantage is taken of both (*b*) and (*c*) the length could be increased to $132 \times 1.5 \times {}^{16}\!/_{12} = 264$ ft, or double the basic length.

stress determined in accordance with Par. 620g of that Code. See examples, pages 832, and 849 to 857.

The reacting forces on the anchors for Schedule 80 pipe bends corresponding to the over-all lengths determined from the tables described in the foregoing are given in Tables VII and IX for expansion U bends and double-offset expansion U bands, respectively. For pipe wall thicknesses greater or less than Schedule 80, the tabulated forces should be multiplied by the ratio of their

TABLE VII.—EXPANSION U BEND, SCHEDULE 80 PIPE

(Reacting forces in pounds for expansion U bends of radii and for operating temperatures shown when absorbing expansion of over-all lengths of pipe given in Table VI)



Nominal pipe size, inches d_n	$\frac{R}{d_n}$	Mean radius of bend, inches R	Reacting forces in pounds for Schedule 80 pipe expansion U bends of radius indicated when absorbing expansion of over-all lengths of pipe given in Table VI which are based on a bending stress of 12,000 psi and no credit for cold spring						
			200 F	300 F	400 F	500 F	600 F	700 F	800 F
$I = 0.8679$	6	12	458	504	550	615			
2	8	16	332	342	367	396			
	10	20	246	263	282	310			
$I = 1.924$	6	15	658	720	790	852			
2½	8	20	467	500	535	576			
	10	25	361	381	407	435			
$I = 3.894$	6	18	912	986	1,090	1,168	1,260		
3	8	24	652	700	747	804	863		
	10	30	502	532	570	597	632		
$I = 9.61$	6	24	1,335	1,435	1,560	1,700	1,850		
4	8	32	950	1,012	1,078	1,156	1,232		
	10	40	722	767	822	860	900		
$I = 40.49$	6	36	2,550	2,740	2,930	3,180	3,420		
6	8	48	1,800	1,920	2,050	2,170	2,330		
	10	60	1,325	1,410	1,516	1,580	1,670		
$I = 105.7$	6	48	3,810	4,080	4,400	4,770	5,060		
8	8	64	2,750	2,920	3,100	3,290	3,500	3,680	
	10	80	2,100	2,210	2,350	2,480	2,580	2,780	
$I = 244.8$	6	60	5,650	6,070	6,620	6,980	7,500	7,830	
10	8	80	4,120	4,360	4,620	4,910	5,210	5,330	5,680
	10	100	3,180	3,340	3,540	3,720	3,890	4,080	4,270
$I = 475.1$	6	72	7,680	8,220	8,860	9,470	10,100	10,660	11,080
12	8	96	5,590	5,920	6,320	6,680	7,050	7,410	7,680
	10	120	4,300	4,570	4,820	5,020	5,330	5,460	5,700
$I = 687.3$	6	84	9,100	9,670	10,350	11,060	11,700	12,000	12,900
14 O.D.	8	112	6,280	6,640	7,000	7,450	7,860	8,130	8,480
	10	140	4,870	5,080	5,470	5,640	6,120	6,130	6,340

TABLE VII.—(*Concluded*)

Nominal pipe size, inches d_n	$\frac{R}{d_n}$	Mean radius of bend, inches R	Reacting forces in pounds for Schedule 80 pipe expansion U bends of radius indicated when absorbing expansion of over-all lengths of pipe given in Table VI which are based on a bending stress of 12,000 psi and no credit for cold spring						
			200 F	300 F	400 F	500 F	600 F	700 F	800 F
$I = 1156$ 16 O.D.	5	80	13,600	14,800	16,000	17,100	18,250	19,300	20,100
	6	96	11,150	11,950	12,750	13,650	14,400	15,200	15,830
	7	112	9,520	10,050	10,750	11,450	12,050	12,650	13,200
$I = 1833$ 18 O.D.	5	90	16,960	18,400	19,860	21,400	22,700	23,700	25,000
	6	108	14,000	14,950	16,000	17,000	18,100	18,500	19,250
	7	126	11,830	12,600	13,500	14,350	15,150	15,830	16,500
$I = 2772$ 20 O.D.	5	100	20,900	22,600	24,400	26,200	27,900	29,300	30,600
	6	120	17,150	18,280	19,500	20,800	22,400	23,100	24,000
	7	140	14,560	15,400	16,400	17,400	18,400	19,500	20,300
$I = 5672$ 24 O.D.	5	120	29,700	32,100	34,300	36,900	39,300	41,200	43,100
	6	144	24,300	26,000	27,700	29,600	31,400	33,000	34,300
	7	168	20,600	21,900	23,300	24,500	25,600	26,900	27,900

Table is based on Fig. 41. Reacting forces correspond to Schedule 80 pipe of lengths given in Table VI. For reacting forces for Schedule 40 or other pipe thickness schedules, multiply tabulated forces by the ratio of their respective moments of inertia. See text for method of determining forces for lengths adjusted for stresses above or below 12,000 psi.

Example.—What will be the reacting force under the following conditions for a 16-in. O.D. Schedule 80 pipe line containing an expansion U bend having a radius of six pipe diameters if the line is installed at 60 F and the operating temperature is 400 F?

(*a*) For the basic conditions of Table VI?

(*b*) If the allowable bending stress is 16,000 psi instead of 12,000 psi?

(*c*) If credit is taken for cold springing the line?

(*d*) If advantage is taken of both (*b*) and (*c*)?

(*e*) If Schedule 40 pipe having a moment of inertia of $I = 732$ is used instead of Schedule 80 pipe having a moment of inertia of $I = 1{,}156$?

Solution.—(*a*) The reacting force for the basic conditions of Table VI and a length of line of 132 ft is read from Table VII as 12,750 lb (see also example under Table VI).

(*b*) For a bending stress of 16,000 psi and a corresponding length of 176 ft, the reacting force would be $12{,}750 \times {}^{16}\!/_{12} = 17{,}000$ lb.

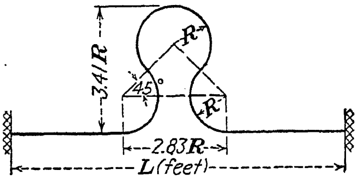
(*c*) No adjustment of forces is necessary if the line was cold sprung so as to increase its allowable length since the net amount of expansion which the bend has to absorb remains the same.

(*d*) The combination of (*b*) and (*c*) produces a reacting force of 17,000 lb.

(*e*) For Schedule 40 16-in. O.D. pipe the reacting forces would be reduced in proportion to the ratio of the moments of inertia, *viz.*, to $732/1{,}156 = 0.634$ of the forces existing with Schedule 80 pipe.

Table VIII.—Double-offset (540 Deg) Expansion U Bend, All Pipe-thickness Schedules

(Over-all lengths of pipe in feet permissible with double-offset expansion U bends of radii and for operating temperatures shown.)



Nominal pipe size, inches d_n	$\frac{R}{d_n}$	Mean radius of bend, inches R	L = over-all length of pipe in feet permissible with double-offset expansion U bend of radius indicated (Based on maximum longitudinal bending stress of 12,000 psi)							
			200 F	300 F	400 F	500 F	600 F	700 F	800 F	850 F
2	5	10	55.8	31.2	22.5	16.6	12.9	11.2	10.0	9.2
	6	12	72.0	40.8	29.1	21.7	17.5	15.0	13.3	12.5
	7	14	91.0	51.7	36.6	27.5	22.1	19.6	17.5	16.2
2½	5	12.5	72.5	40.8	28.3	20.8	17.1	15.0	13.3	12.1
	6	15	93.8	52.5	37.1	27.9	29.9	19.6	17.5	15.8
	7	17.5	117.5	66.6	46.7	35.0	29.1	25.0	22.1	20.8
3	5	15	91.3	51.3	35.9	27.5	22.1	18.3	16.7	15.4
	6	18	117.5	66.3	46.3	36.8	28.3	24.2	21.7	20.4
	7	21	142.5	82.5	57.5	44.2	35.9	30.8	27.5	25.8
4	5	20	131.2	73.3	50.8	38.3	31.3	26.7	23.3	22.5
	6	24	168.2	95.0	66.3	50.5	40.8	35.0	31.3	29.6
	7	28	203.0	118.4	82.5	62.5	50.8	44.2	39.2	37.5
6	5	30	208.5	117.8	81.2	62.5	50.4	42.8	38.3	36.2
	6	36	245.0	151.5	105.7	80.0	65.4	55.8	50.0	47.5
	7	42	320	186.6	132.5	99.2	81.3	69.7	62.5	59.2
8	5	40	285	166.6	114.2	86.7	70.8	60.4	53.3	50.8
	6	48	362	211.5	149.2	113.2	92.4	79.2	70.5	67.5
	7	56	443	262	185.0	141.5	115.8	99.2	88.3	84.2
10	5	50	372	216.5	152.5	115.8	85.0	80.8	70.8	67.5
	6	60	462	270	190.	149.0	122.5	105.0	93.8	89.3
	7	70	590	343	242	187.0	152.0	131.2	116.2	110.8
12	5	60	438	256	181.	139.5	114.2	97.5	86.7	82.5
	6	72	560	330	232	181.	149.2	128.6	114.6	109.2
	7	84	693	400	280	220	181.5	160.0	142.5	136.0
14 O.D.	5	70	542	316	172.6	159.2	142.	121.2	107.5	101.6
	6	84	687	400	283	219	179	159.2	141.6	133.2
	7	98	850	495	350	270	226.5	195.8	175.0	165.0

TABLE VIII.—(*Concluded*)

Nominal pipe size, inches d_n	$\frac{R}{d_n}$	Mean radius of bend, inches R	L = over-all length of pipe in feet permissible with double-offset expansion U bend of radius indicated (Based on maximum longtiudinal bending stress of 12,000 psi)							
			200 F	300 F	400 F	500 F	600 F	700 F	800 F	850 F
16 O.D.	5	80	624	365	258	196.	161.6	138.2	122.5	116.2
	6	96	782	456	322	249	205	179.2	160.8	153.3
	7	112	990	577	407	315	257	216	187	175
18 O.D.	5	90	697	407	287	222	180.	156.8	140.	133.5
	6	108	940	548	387	299	243	204	183.3	175.
	7	126	1,110	647	457	353	287	240	206	193.
20 O.D.	5	100	777	453	319	247	202	174	157.	148.
	6	120	1,010	588	405	305	248	209	181	170.
	7	140	1,230	717	507	392	320	267	228	214
24 O.D.	5	120	930	542	383	293	237	203.	186.	179.
	6	144	1,270	742	523	390	311	262	227	213
	7	168	1,580	922	642	495	403	340	291	273

Table is based on Fig. 42. Pipe is assumed to be installed at 60 F. No credit is taken for cold spring in establishing tabulated pipe lengths. See text for method of adjusting lengths for bending stresses above or below 12,000 psi and for taking into account partial or full credit for cold spring. Lengths are based on values of K and β for Schedule 80 pipe but are sufficiently accurate for all pipe thickness schedules.

Example.—What length of 16-in. O.D. pipe can be used under the following conditions with a double-offset expansion U bend having a radius of six pipe diameters if the line is installed at 60 F and the operating temperature is 400 F?

(*a*) For the basic conditions of Table VIII?

(*b*) If the allowable bending stress is 16,000 psi instead of 12,000 psi?

(*c*) If credit is taken for cold springing the line?

(*d*) If advantage is taken of both (*b*) and (*c*)?

Solution.—(*a*) The length of line L read from Table VIII is 322 ft.

(*b*) For a bending stress of 16,000 psi the length would be $322 \times {}^{16}\!/_{12} = 429$ ft.

(*c*) The maximum credit for cold spring allowed under the Code for Pressure Piping (see text page 778) would increase the basic length to $322 \times 1.5 = 483$ ft.

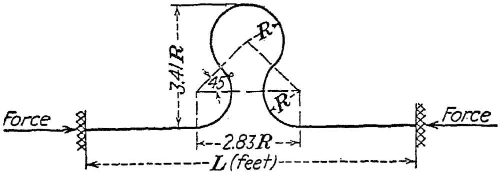
(*d*) If advantage is taken of both (*b*) and (*c*) the length could be increased to $322 \times 1.5 \times 16/12 = 644$ ft, or double the basic length.

respective moments of inertias. The moments of inertia for Schedule 80 pipe used in calculating the forces given on Tables VII and IX are reproduced on these tables. The moments of inertia for other schedules of pipe thicknesses may be found from Table X, page 858, or calculated by the formula given on page 860, if the pipe is of special thickness.

The tabulated forces are based on the lengths of pipe determined for a bending stress of 12,000 psi. The forces for the lengths

TABLE IX.—DOUBLE-OFFSET (540 DEG) EXPANSION U BEND, SCHEDULE 80 PIPE

(Reacting forces in pounds for double-offset expansion U bends of radii and for operating temperatures shown when absorbing expansion of over-all lengths of pipe given in Table VIII)



Nominal pipe size, inches d_n	$\frac{R}{d_n}$	Mean radius of bend, inches R	Reacting forces in pounds for Schedule 80 pipe double-offset expansion U bends of radius indicated when absorbing expansion of over-all lengths of pipe given in Table VIII which are based on a bending stress of 12,000 psi and no credit for cold spring							
			200 F	300 F	400 F	500 F	600 F	700 F	800 F	850 F
I = 0.8679	5	10	299	316	331	347	365	375	384	393
2	6	12	251	264	275	287	298	309	316	320
	7	14	211	220	229	239	247	253	259	263
I = 1.924	5	12.5	455	493	503	530	550	565	578	592
2½	6	15	367	385	402	418	432	448	457	467
	7	17.5	305	319	332	345	357	366	376	379
I = 3.894	5	15	631	663	695	727	757	784	802	817
3	6	18	508	532	556	577	602	621	635	642
	7	21	427	447	464	482	498	512	522	529
I = 9.61	5	20	903	952	1,000	1,048	1,086	1,122	1,152	1,162
4	6	24	745	778	815	847	873	900	922	932
	7	28	628	652	678	707	728	747	762	770
I = 40.49	5	30	1,720	1,810	1,910	1,990	2,070	2,110	2,160	2,190
6	6	36	1,420	1,480	1,540	1,600	1,660	1,710	1,740	1,760
	7	42	1,200	1,240	1,280	1,340	1,380	1,410	1,440	1,450
I = 105.7	5	40	2,590	2,720	2,850	2,970	3,090	3,180	3,260	3,290
8	6	48	2,130	2,230	2,310	2,410	2,490	2,530	2,600	2,630
	7	56	1,820	1,890	1,960	2,030	2,120	2,150	2,190	2,200
I = 244.8	5	50	3,850	4,050	4,230	4,420	4,640	4,720	4,840	4,870
10	6	60	3,190	3,330	3,460	3,590	3,710	3,810	3,870	3,910
	7	70	2,700	2,810	2,910	3,000	3,100	3,170	3,240	3,270
I = 475.1	5	60	5,260	5,520	5,770	6,000	6,210	6,400	6,550	6,600
12	6	72	4,330	4,530	4,720	4,870	5,020	5,170	5,270	5,320
	7	84	3,690	3,830	3,970	4,080	4,210	4,300	4,390	4,430
I = 687.3	5	70	5,920	6,220	6,720	6,820	6,950	7,170	7,300	7,380
14 O.D.	6	84	4,870	5,070	5,270	5,450	5,620	5,720	5,860	5,920
	7	98	4,060	4,200	4,350	4,480	4,590	4,720	4,790	4,850

TABLE IX.—(*Concluded*)

Nominal pipe size, inches d_n	$\frac{R}{d_n}$	Mean radius of bend, inches R	Reacting forces in pounds for Schedule 80 pipe double-offset expansion U bends of radius indicated when absorbing expansion of overall lengths of pipe given in Table VIII which are based on a bending stress of 12,000 psi and no credit for cold spring							
			200 F	300 F	400 F	500 F	600 F	700 F	800 F	850 F
$I = 1156$ 16 O.D.	5	80	7,630	8,000	8,350	8,670	8,950	9,220	9,440	9,570
	6	96	6,300	6,520	6,820	7,050	7,240	7,420	7,540	7,630
	7	112	5,330	5,530	5,720	5,880	6,060	6,230	6,390	6,470
$I = 1883$ 18 O.D.	5	90	9,520	10,000	10,420	10,820	11,230	11,480	11,740	11,840
	6	108	7,920	8,200	8,500	8,820	9,090	9,370	9,540	9,620
	7	126	6,690	6,960	7,180	7,420	7,620	7,830	8,070	8,170
$I = 2772$ 20 O.D.	5	100	11,700	12,040	12,780	13,320	13,750	14,120	14,440	14,650
	6	120	9,600	10,000	10,400	10,800	11,180	11,520	11,850	12,000
	7	140	8,180	8,480	8,780	9,040	9,320	9,580	9,840	9,940
$I = 5672$ 24 O.D.	5	120	16,600	17,420	18,140	18,960	19,600	20,200	20,500	20,650
	6	144	13,660	14,250	14,780	15,350	15,920	16,380	16,810	17,050
	7	168	11,620	12,040	12,480	12,880	13,260	13,640	14,000	14,160

Table is based on Fig. 42. Reacting forces correspond to Schedule 80 pipe of lengths given in Table VIII. For reacting forces for Schedule 40 or other pipe thickness schedules, multiply tabulated forces by the ratio of their respective moments of inertias. See text for method of determining forces for lengths adjusted for stresses above or below 12,000 psi.

Example.—What will be the reacting force under the following conditions for a 16-in. O.D. Schedule 80 pipe line containing a double-offset expansion U bend having a radius of six pipe diameters if the line is installed at 60 F and the operating temperature is 400 F?

(*a*) For the basic conditions of Table VIII?

(*b*) If the allowable bending stress is 16,000 psi instead of 12,000 psi?

(*c*) If credit is taken for cold springing the line?

(*d*) If advantage is taken of both (*b*) and (*c*)?

(*e*) If Schedule 40 pipe having a moment of inertia of $I = 732$ is used instead of Schedule 80 pipe having a moment of inertia of $I = 1{,}156$?

Solution.—(*a*) The reacting force for the basic conditions of Table VIII and a length of line of 322 ft is read from Table IX as 6,820 lb (see also example under Table VIII, page 839).

(*b*) For a bending stress of 16,000 psi and a corresponding length of 429 ft, the reacting force would be $6{,}820 \times {}^{16}\!/_{12} = 9{,}093$ lb.

(*c*) No adjustment of forces is necessary if the line was cold sprung so as to increase its allowable length since the net amount of expansion which the bend has to absorb remains the same.

(*d*) The combination of (*b*) and (*c*) produces a reacting force of 9,093 lb.

(*e*) For Schedule 40 16-in. O.D. pipe the reacting forces would be reduced in proportion to the ratio of the moments of inertia, *viz.*, to $732/1{,}156 = 0.634$ of the forces existing with Schedule 80 pipe.

obtained by adjustment for bending stress other than 12,000 psi are found by applying the ratio of the stresses in exactly the same manner as the adjustment for lengths.[1] It should be noted that no adjustment of forces is necessary because of taking credit for cold spring, since the net amount of expansion provided for remains constant. The slightly greater flexibility of the longer lengths of pipe between bend and anchor points when lengths are adjusted to take advantage of cold spring results in only a minor reduction in forces, which may be neglected.

Chart Solutions.—The charts of Figs. 40 to 42 together with the formulas given thereon enable the piping designer to determine the flexibility of any length of straight pipe in combination with either a square bend made up of 90-deg bends or elbows and straight tangents, an expansion U bend, or a double-offset expansion U bend inserted in its length. From these charts and formulas the reacting forces, bending moments, and longitudinal bending stresses can be determined for the respective types of bends for any given amount of expansion and for the assumed over-all line dimensions.

The chart for the square bend made up of 90-deg bends and straight lengths (Fig. 40) is particularly useful in estimating the flexibility of proposed piping layouts. The square-corner approximation gives results which closely agree with exact solutions if the flexibility[2] multiplication factor K for the curved portions is approximately 0.6, and if the bend is located midway between the anchor points. If the longer run between the bend and anchor is not more than four times the shorter run, the results are within the manufacturing tolerances permitted with pipe.

It should be noted that published data on the amount of elongation that can be taken care of by a given size and type of bend which does not take into account the position of the anchors and the lengths of straight pipe between the ends of the bend and the anchors is of little practical value. The relations between deflection and reacting forces and restraining moments at the ends of the section of piping are found from the curves of Figs. 40 to 42 by forming ratios between the lengths of straight pipe between the ends of the bend and the anchors and the principal dimensions

[1] For examples illustrating application of Tables VI to IX in the design of steam piping, see article on "Design of Steam Transmission Piping," by Arthur McCutchan, *Heating, Piping and Air Conditioning*, August and September, 1943.

[2] For derivation of this relation, see p. 800.

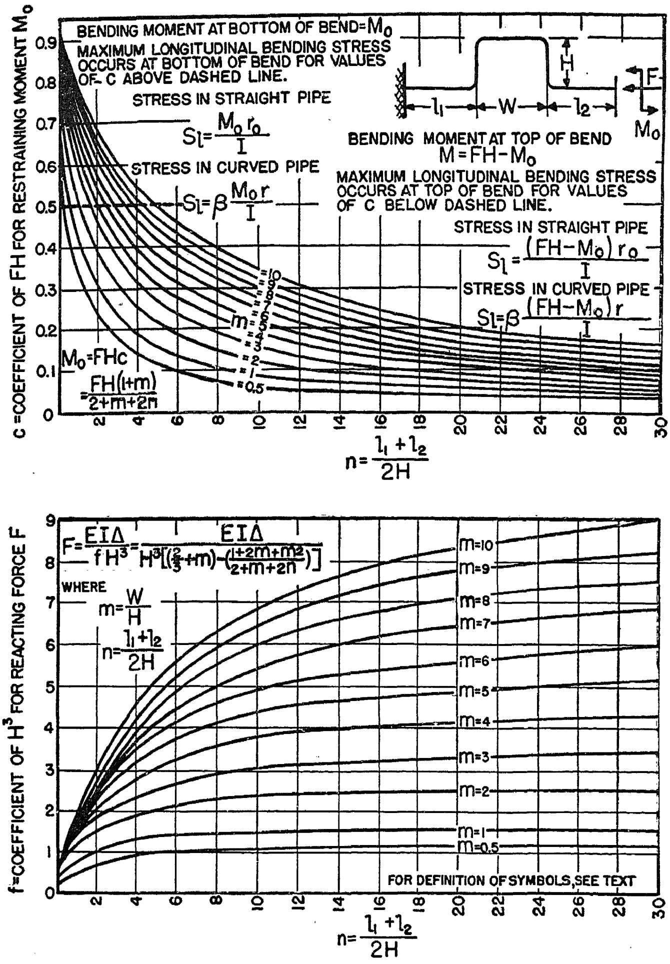


FIG. 40.—Chart for determining relation between deflection and reacting force for a *square bend* made up of 90-deg arcs and straight pipe inserted in a run of straight pipe having fixed anchors at the ends. Relations are accurately represented if the flexibility factor, K, for the curved pipe is 0.59 and if $l_1 = l_2$. This chart may be used with reasonable accuracy, however, for the pipe bend ratios usually employed if l_1 is not greater than four times l_2.

of the bends. When the lengths of straight pipe between the ends of the bend and the anchors are zero, *i.e.*, $n = 0$, the coeffi-

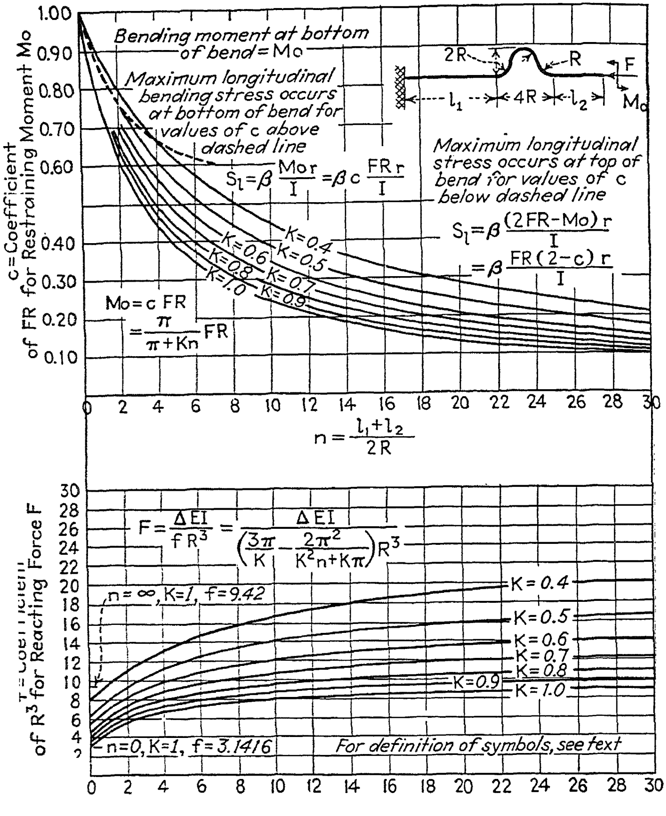


FIG. 41.—Chart for determining relation between deflection and reacting force for an *expansion U bend* inserted in a run of straight pipe having fixed anchors at the ends. Relations are strictly correct where $l_1 = l_2$. This chart may be used with reasonable accuracy, however, if l_1 is not greater than four times l_2.

cient relating deflection and reacting force corresponds to the values given for bends with fixed or fully restrained ends (see reference footnote 2(*c*), page 780). When the lengths of straight pipe

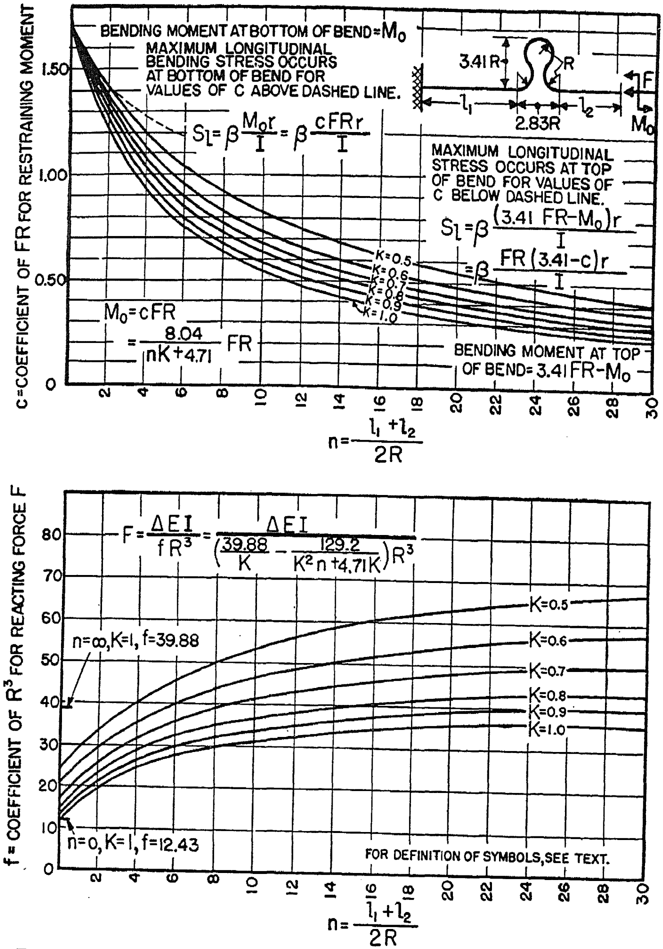
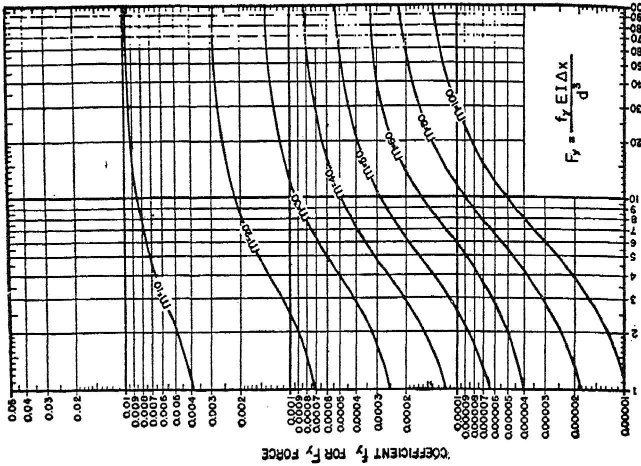
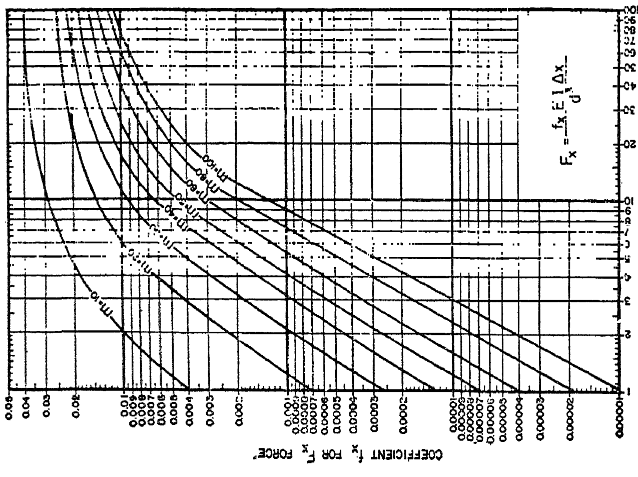


FIG. 42.—Chart for determining relation between deflection and reacting force for a *double-offset expansion U bend* inserted in a run of straight pipe having fixed anchors at the ends. Relations are strictly correct where $l_1 = l_2$. This chart may be used with reasonable accuracy, however, if l_1 is not greater than four times l_2.

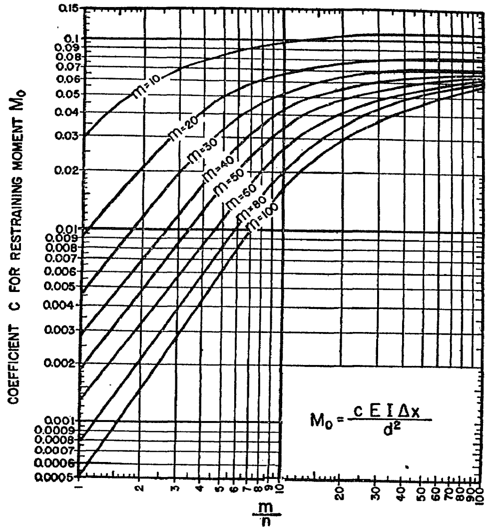


$F_y = \frac{f_y E I \Delta x}{d^3}$
$\frac{m}{n}$
COEFFICIENT f_y FOR F_y FORCE

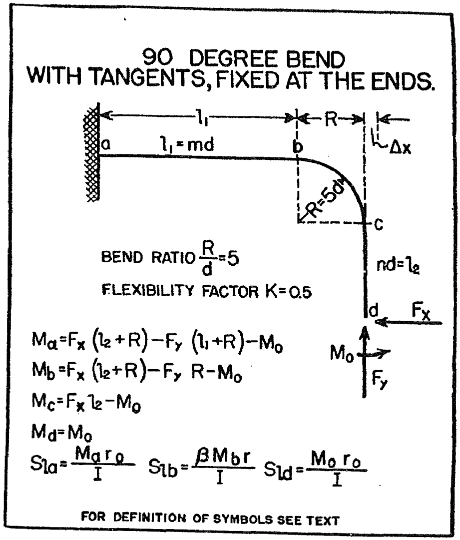


$F_x = \frac{f_x E I \Delta x}{d^3}$
$\frac{m}{n}$
COEFFICIENT f_x FOR F_x FORCE

(a)



(c)



(d)

FIG. 43.—Chart for determining reacting forces, bending moments, and stresses in a 90-*deg bend with tangents, fixed at the ends.* Relations are strictly correct for bends formed to a center-line radius five times the nominal pipe diameter and where the flexibility factor for curved pipe, $K = 0.5$.

Reference to the table of correction factors on page 848, will serve to indicate where it is necessary to correct for variations in the value of K above and below 0.5. For the usual proportions encountered in piping-design problems, the correction will not exceed 10 per cent.

between the ends of the bends and the anchors are large, *i.e.*, $n = \infty$, the values found from the curves approach those given in reference footnote 2(*a*), page 779, for free or hinged ends. These limiting conditions are noted on the curves of Figs. 40 to 42.

The charts of Fig. 43 for the 90-deg bend with tangents fixed at the ends afford a distinct advance in the ease with which this case can be solved. The forces, moments, and bending stresses are obtained directly from the charts with a minimum of computation. The results are strictly correct for an arc where the center-line radius is five times the nominal diameter of the pipe and where the flexibility multiplication factor K is 0.5.

The charts of Figs. 40 to 43, inclusive, in their present revised form were developed in cooperation with the Walworth Company for use in its *Catalogue* 89.

Reference to the following table will serve to indicate where it is necessary to correct for variations in the value of K above and below 0.5. For the usual proportions encountered in piping-design problems, the correction will not exceed 10 per cent.

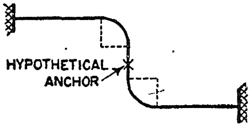
CORRECTION FACTORS FOR RESULTS OBTAINED FROM FIG. 43 FOR 90-DEG. BEND IF K IS NOT 0.5

DIRECTIONS: If K is not 0.5, multiply the results obtained from Fig. 43 for F_x, F_y, or M_o, as the case may be, by the correction factor shown below for the desired value of K, interpolating where necessary.

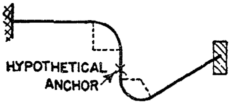
		$K = 0.4$			$K = 0.6$			$K = 0.8$		
		m/n 1	m/n 10	m/n 100	m/n 1	m/n 10	m/n 100	m/n 1	m/n 10	m/n 100
$m = 10$	F_x	0.915	0.873	0.820	1.070	1.114	1.141	1.200	1.300	1.440
	F_y	0.915	0.897	0.873	1.070	1.096	1.117	1.200	1.260	1.333
	M_o	0.955	0.873	0.801	1.030	1.111	1.148	1.100	1.290	1.485
$m = 50$	F_x	0.938	0.963	0.840	1.048	1.023	1.147	1.119	1.064	1.433
	F_y	0.938	0.952	0.917	1.048	1.036	1.061	1.119	1.094	1.151
	M_o	0.963	0.968	0.816	1.028	1.018	1.173	1.072	1.052	1.519
$m = 100$	F_x	0.962	0.983	0.880	1.033	1.007	1.107	1.075	1.021	1.311
	F_y	0.962	0.960	0.971	1.033	1.022	1.019	1.075	1.061	1.059
	M_o	0.977	0.989	0.866	1.022	1.004	1.121	1.048	1.012	1.359

The chart of Fig. 43, through the insertion of hypothetical anchor points as indicated in sketches A and B, makes possible an approximate solution of many problems encountered in piping

design. While the forces and stresses determined will be approximate, it can be shown that the line will possess adequate flexibility if the sections are themselves sufficiently flexible. In the three-dimensional pipe line shown in sketch B, the true forces will not



Sketch "A"



Sketch "B"

be found by this means although adequate flexibility can be assured by such an approximation.

SOLUTION OF SAMPLE PROBLEMS

The use of these charts, curves, and tables for simplified solutions is illustrated by the following series of examples. The symbols used in this section are defined on page 785.

Square Bend (Fig. 40).—Assume a 10-in. high-pressure steam-transmission line is to be installed overhead to convey saturated steam at 400 lb pressure. The length of the line is 1,000 ft, and available space is such that the most economical method of caring for expansion can be used. As was demonstrated in *Trans.*[1] *ASME* 1931, FSP 53-17, a series of square bends with widths equal to one-half the distance between anchor points will utilize the inherent flexibility of the line to best advantage, and the problem resolves into finding a height H for the offsets which will afford sufficient flexibility.

If anchors are installed every 200 ft, five "square" bend offsets will be required, each having a width of 100 ft and an assumed height of 10 ft.

Assumed Data:

1. Pipe, 10-in. Schedule 40 (standard weight) steel, 0.365-in. wall thickness (page 361).
2. Temperature variation, 60 to 448 F corresponding to 400 lb gauge saturated steam (steam tables, page 235).
3. Distance between anchor points, 200 ft.
4. The "square" bend offsets will be made with a series of 90-deg arcs and straight sections of pipe. But, in order to simplify the solution, these arcs are considered as replaced by square corners. The square-corner approximation

[1] Footnote 2(f), p. 780.

gives results which are exact if the flexibility multiplication factor K for the curved portions is 0.59 and if the bend is located midway between the anchor points. The error involved in assuming the arcs as replaced by square corners is of small consequence when, as in this example, the length of straight pipe subjected to bending is large as compared with the length of curved pipe. The stress modification in the curved pipe caused by flattening of the cross section is taken into account by applying the factor β in the same manner as in an exact solution.

5. Dimensions:

$$l_1 = l_2 = 50 \text{ ft}; l_1 + l_2 = 100 \text{ ft} = 1{,}200 \text{ in.}$$
$$w = \text{width of bend} = 100 \text{ ft} = 1{,}200 \text{ in.}$$
$$H = \text{height of bend} = 10 \text{ ft} = 120 \text{ in.}$$

Solution:

1. Δ, linear expansion to be absorbed when steel pipe is heated from 60 to 448 F is found from Table I (page 756), as 3.25 in. per 100 ft. Since the distance between anchors is 200 ft,

$$\Delta = 2 \times 3.25 = 6.50 \text{ in.}$$

2. E, modulus of elasticity, corresponding to 448 F, is found from Fig. 39.

$$E = 28{,}200{,}000 \text{ psi.}$$

3. I, moment of inertia of 10-in. Schedule 40 (standard weight) steel pipe from Table X (page 858) is

$$I = 160.7 \text{ in.}^4$$

4. n, ratio of lengths of pipe between offsets and anchors to height of offset is found as

$$n = \frac{l_1 + l_2}{2H} = \frac{1{,}200}{2 \times 120} = 5.$$

5. m = ratio of width of bend to height of offset

$$m = \frac{w}{H} = \frac{1{,}200}{120} = 10.$$

6. The value of the coefficient f is read from Fig. 40 corresponding to $n = 5$ and $m = 10$.

$$f = 5.1.$$

7. The reacting force F is found by substituting the values just determined in the equation

$$F = \frac{\Delta EI}{fH^3} = \frac{6.50 \times 28{,}200{,}000 \times 160.7}{5.1 \times (120)^3} = 3{,}340 \text{ lb}$$

8. The value[1] of the coefficient c is found from the upper group of curves of Fig. 40 corresponding to $n = 5$ and $m = 10$.

$$0.5.$$

[1] Footnote 1(f), p. 780.

9. Since the optimum shape of bend was chosen, the bending moments in the lengths of pipe between the offsets and the anchors is equal to the bending moment at the top of the bend.

$$M_o = FHc = 3{,}340 \times 120 \times 0.5 = 200{,}400 \text{ in-lb.}$$

10. The bending stress in the straight pipe is

$$S_l = \frac{M_o r_o}{I} = \frac{200{,}400 \times 5.375}{160.7} = 6{,}700 \text{ psi}$$

where r_o is the outside radius of pipe cross section, in.

11. The longitudinal bending stress in the curved portions is found from the following formula, which takes into account the stress multiplication factor β. This factor for Schedule 40 pipe having an R/d ratio of 5 from Fig. 35 is

$$\beta = 1.10.$$

12. The maximum longitudinal bending stress in the curved portions is

$$S_l = \frac{\beta M_o r_o}{I} = \frac{1.1 \times 200{,}400 \times 5.192}{160.7} = 7{,}120 \text{ psi}$$

where r is the mean radius of pipe wall.

13. The longitudinal pressure stress[1] is found by dividing the product of the internal pressure and the internal cross-sectional area of the pipe by the cross-sectional area of pipe metal. For 10-in. Schedule 40 pipe these areas from Table X, page 366, and Table XI, page 859, are 78.85 and 11.90 sq in., respectively.

$$S_2 = \frac{400 \times 78.85}{11.90} = 2{,}650 \text{ psi.}$$

14. The total combined stress in the curved portions is the sum of the bending stress and the pressure stress

$$S = 7{,}120 + 2{,}650 = 9{,}770 \text{ psi.}$$

Summary of Results:

1. The reacting force at the anchors, $F = 3{,}340$ lb.
2. The restraining moment at the anchors, $M_o = 200{,}400$ in-lb.
3. The maximum bending stress in the curved pipe, $S_l = 6{,}700$ psi.
4. The total combined longitudinal pressure plus longitudinal bending stress $S = 9{,}770$ psi.
5. The assumed height of 10 ft may be considered adequate for these service conditions.

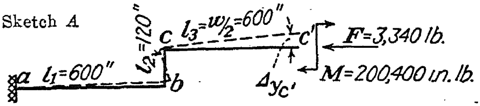
Movement of Top of Bend.—In addition to checking the reacting forces and the bending stress, the upward movement at the top of the bend caused by compressing the linear expansion of 6.5 in. into the bend should be determined. This can be done by applying the force and moment existing at the point in question and solving for the corresponding upward movement[2] (see page 784). A free body diagram of the left half of the square bend considered in this example is shown in Sketch *A*. Since the deflections are related to the cube of the linear

[1] See p. 824.

[2] "Strength of Materials," *op. cit.*, p. 147.

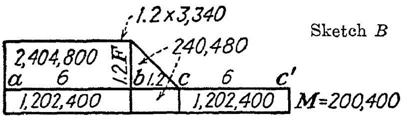
dimensions, it is possible to reduce the numerical task by dividing all lengths by 100 and dividing the product EI by the cube of 100 which is 1,000,000.

Sketch *A*



The bending moment diagram for this one-half square bend is given in Sketch *B*. As explained on page 784, the deflection at point c' is the algebraic sum of the deflection of the several parts between c' and the fixed end a. The point b moves upward an amount proportional to the product of the next area of the bending moment diagram and one-half the length l_1 (see Case III, page 789). The point c' moves upward this amount plus an amount determined by the rotations of all the parts between c' and a or the algebraic sum of the bending moment areas to the left of c times l_3 (see Case I), minus an amount proportional to the product of the area of the bending moment diagram which acts on the length c-c' times $\frac{1}{2}l_3$. The summation of these deflections up and down divided by the product of E and I for the 10-in. Schedule 40 pipe at 448 F of 28,200,000 and 160.7 in.^{4}, respectively, gives the net deflection in inches.

Sketch *B*



Case III. $\dfrac{EI\Delta y_b}{1{,}000{,}000} = \psi_M \tfrac{1}{2} l_1 = 1{,}202{,}400 \times 3 = 3{,}607{,}200$ up.

Case I. $\dfrac{EI\Delta y_{c'}}{1{,}000{,}000} = \psi_c l_3 = 1{,}202{,}400 \times 6 = 7{,}214{,}400$ up.

Case III. $\psi_M \tfrac{1}{2} l_3 = 1{,}202{,}400 \times 3 = \underline{3{,}607{,}200 \text{ down.}}$

$$EI\Delta y = 7{,}214{,}400 \text{ up.}$$

$$\Delta y_{c'} = \frac{7{,}214{,}400}{28.2 \times 160.7} = 15.92 \text{ in.}$$

Since the linear expansion compressed back into the bend was 6.5 in., the top of the bend is found to move 15.92/6.5 = 2.45 times as much as the amount of expansion cared for by the bend.

Expansion U Bend (Fig. 41).—The reacting forces and bending stresses in an expansion U bend may be determined in a similar manner from Fig. 41. An example of the solution for a double-offset expansion U bend is given below.

Double-offset Expansion U Bend (Fig. **42**).—Assume that an 8-in. heating header designed for 10-lb saturated steam is to be installed and that expansion is to be taken care of by means of a double-offset expansion U bend.

Required: to determine the reactions at the anchors, the bending moments at the ends, and the maximum bending stress in the line.

Assumed Data:

1. Pipe, 8-in., Schedule 40 (standard weight) steel, 0.322-in. wall thickness (page 366).

2. Temperature variation, 60 to 240 F corresponding to 10-lb gauge saturated steam (steam tables, page 234).

3. Distance between anchor points, 50 ft.

4. Radius of double-offset expansion U bend, $R = 5d = 40$ in.

5. Dimensions $l_1 + l_2$ = (distance between anchors) − (width of bend) = $(50 \times 12) - (2.83 \times 40) = 486.8$ in.

Solution:

1. Δ, linear expansion to be absorbed when pipe is heated from 60 to 240 F is found from Table I (page 756), as 1.43 in. per 100 ft. The distance between anchors in this case is 50 ft.

$$\Delta = \frac{50}{100} \times 1.43 = 0.715 \text{ in.}$$

2. E, modulus of elasticity, corresponding to a working temperatures of 240 F is found from Fig. 39.

$$E = 29{,}500{,}000 \text{ psi.}$$

3. I, moment of inertia of an 8-in. Schedule 40 (standard weight) pipe from Table X (page 858) is

$$I = 72.49 \text{ in.}^4$$

4. K, flexibility multiplication factor for an 8-in. Schedule 40 pipe formed to a radius of $5d$ or 40 in. from Fig. 35 is

$$K = 0.48.$$

5. n, ratio of lengths of tangents to radius of bend

$$n = \frac{l_1 + l_2}{2R} = \frac{486.8}{2 \times 40} = 6.08.$$

6. The value of the coefficient f is read from the lower group of curves of Fig. 42 corresponding to $n = 6.08$ and $K = 0.48$.

$$f = 47.$$

7. The reacting force F is found by substituting the values just determined in the equation

$$F = \frac{\Delta EI}{fR^3} = \frac{0.715 \times 29{,}500{,}000 \times 72.49}{47 \times (40)^3} = 508 \text{ lb.}$$

8. The value of the coefficient c is found from the upper group of curves of Fig. 42 corresponding to $n = 6.08$ and $K = 0.48$.

$$c = 1.06.$$

9. The restraining moment M_o at the right anchor is found as

$$M_o = cFR = 1.06 \times 508 \times 40 = 21{,}540 \text{ in-lb.}$$

10. Since the value of the coefficient c corresponding to $n = 6.08$ and $K = 0.48$ is below the dotted line on Fig. 42, the maximum bending stress occurs at the

top of the double-offset expansion U bend, the stress multiplication factor β for Schedule 40 pipe formed to an R/d ratio of 5 may be read from Fig. 35.

$$\beta = 1.00.$$

NOTE.—When using bends formed to an R/d ratio of approximately 5, β is found to be about unity. In the case of short-radius fittings β may be as high as 2.5.

11. The maximum longitudinal-bending stress at the top of the bend is then found as

$$S_l \quad \frac{- M_o)}{}.$$

$$= \frac{1.0(3.41 \times 508 \times 40 - 21{,}540)}{72.49} 4.15 = 2{,}730 \text{ psi}$$

where r = mean radius of pipe wall,

$$r = \frac{8.625 - 0.322}{2} \quad 4.15 \text{ in.}$$

12. The longitudinal stress due to the internal pressure of 10-lb gauge is found by dividing the product of the internal pressure and the internal cross-sectional area of the 8-in. Schedule 40 pipe from Table X, page 858, by the cross-sectional area of the pipe metal from Table XI, page 859.

$$S_2 = \frac{10 \times 50.03}{8.396} \quad 60.0 \text{ psi.}$$

13. The total combined stress is simply the sum of the longitudinal-bending and longitudinal-pressure stresses or

$$S = 2730 + 60 = 2{,}790 \text{ psi.}$$

Summary of Results:

1. The reaction at the anchors, $F = 508$ lb.
2. The restraining moment at the anchors, $M_o = 21{,}540$ in.-lb.
3. The maximum longitudinal-bending stress at the top of the bend, $S_e = 2{,}730$ psi.
4. The total combined longitudinal-bending plus longitudinal-pressure stress, $S = 2{,}790$ psi.

Ninety-degree Bend with Tangents Fixed at the Ends (Fig. 43).—Assume a 90-deg bend is located in a 12-in. Schedule 80 boiler lead in a 650-lb 850 F superheated steam system which has a horizontal run of 50 ft and a vertical run of 40 ft to fixed anchor points. Required: to find the horizontal and vertical reactions, the maximum bending moment, and the stress in the bend.

Assumed Data:

1. Pipe; 12-in., Schedule 80, 0.687-in. wall thickness (page 361).
2. Temperature variation, 70 to 850 F.
3. Radius of bend, $5d = 5 \times 12 = 60$ in.
4. Horizontal tangent, $l_1 = (50 \times 12) - 60 = 540$ in.
5. Vertical tangent, $l_2 = (40 \times 12) - 60 = 420$ in.

Solution:

1. Δx, linear expansion to be absorbed when pipe is heated from 70 to 850 F is found from Table I (page 756), as 7.10 in. per 100 ft. The longest run in this case is 50 ft. Therefore,

$$\Delta x = \frac{50}{100} \times 7.12 = 3.56 \text{ in.}$$

NOTE.—Only the expansion corresponding to the longer run need be determined when using the method given on Fig. 43.

2. E, modulus of linear elasticity for a working temperature of 850 F is found from Fig. 39.

$$E = 23{,}000{,}000 \text{ psi.}$$

3. I, moment of inertia of 12-in. Schedule 80 pipe from Table X (page 858) is

$$I = 475.1 \text{ in.}^4$$

4. m = ratio of length of longer tangent l_1 to nominal diameter of pipe.

$$\frac{l_1}{12} = \frac{540}{12} = 45.$$

5. n = ratio of length of shorter tangent l_2 to nominal diameter of pipe.

$$\frac{l_2}{12} = \frac{420}{12} = 35.$$

6. The quotient m/n is $^{45}\!/_{35} = 1.28$.

7. The value of the coefficient f_X for the F_X force is read from the upper left-hand group of curves of Fig. 43 corresponding to the quotient $m/n = 1.28$ and interpolating for $m = 45$ between the curves from $m = 40$ and $m = 50$.

$$f_X = 0.000138.$$

8. The horizontal reacting force F_X is found by inserting the values just found in the equation.

$$F_X = \frac{f_X E I \Delta x}{d^3}.$$

$$F_X = \frac{0.000138 \times 23{,}000{,}000 \times 475.1 \times 3.56}{(12)^3} = 3{,}100 \text{ lb.}$$

9. The value of the coefficient f_Y for the F_Y force is found from the upper right-hand group of curves of Fig. 43 for the same quotient $m/n = 1.28$ and interpolating for $m = 45$ between the curves for $m = 40$ and $m = 50$.

$$f_Y = 0.00009.$$

10. The vertical reacting force F_Y is found from the equation

$$F_Y = \frac{f_Y E I \Delta x}{d^3}.$$

Since the quantity $EI\Delta x/d^3$ is identical with that in the equation for the F_X force,

$$F_Y = \frac{f_Y}{f_X} F_X = \frac{0.00009}{0.000138} \times 3{,}100 = 2{,}020 \text{ lb}$$

TABLE X.—OUTSIDE-DIAMETER FUNCTIONS AND MOMENTS OF INERTIA FOR ASA B36.10 STEEL PIPE[1]

Nominal pipe size, inches	Outside diameter functions[2]				I = Moment[3] of inertia, in.[4]									
	D	D^2	D^3	D^4	Schedule 10	Schedule 20	Schedule 30[2]	Schedule 40[2]	Schedule 60[2]	Schedule 80[2]	Schedule 100	Schedule 120	Schedule 140	Schedule 160
⅛	0.405	0.164	0.0664	0.027	.....	.....	.....	**0.001064**	.....	**0.001216**				
¼	0.540	0.292	0.1575	0.085	.....	.....	.....	**0.003312**	.....	**0.003766**				
⅜	0.675	0.456	0.3075	0.208	.....	.....	.....	**0.007291**	.....	**0.008619**				
½	0.840	0.706	0.5927	0.498	.....	.....	.....	**0.01709**	.....	**0.02008**	.....	.....	.....	0.02212
¾	1.050	1.103	1.158	1.216	.....	.....	.....	**0.03704**	.....	**0.04479**	.....	.....	.....	0.05269
1	1.315	1.729	2.274	2.990	.....	.....	.....	**0.08734**	.....	**0.1056**	.....	.....	.....	0.1251
1¼	1.660	2.756	4.574	7.593	.....	.....	.....	**0.1947**	.....	**0.2418**	.....	.....	.....	0.2839
1½	1.900	3.610	6.859	13.03	.....	.....	.....	**0.3099**	.....	**0.3912**	.....	.....	.....	0.4824
2	2.375	5.641	13.40	31.82	.....	.....	.....	**0.6660**	.....	**0.8679**	.....	.....	.....	1.162
2½	2.875	8.266	23.76	68.32	.....	.....	.....	**1.530**	.....	**1.924**	.....	.....	.....	2.353
3	3.500	12.25	42.88	150.1	.....	.....	.....	**3.017**	.....	**3.894**	.....	.....	.....	5.032
3½	4.0	16.00	64.00	256.0	.....	.....	.....	**4.788**	.....	**6.280**				
4	4.5	20.25	91.13	410.1	.....	.....	.....	**7.233**	.....	**9.610**	.....	11.65	.....	13.27
5	5.563	30.95	172.2	957.7	.....	.....	.....	**15.16**	.....	**20.67**	.....	25.73	.....	30.03
6	6.625	43.89	290.8	1,926	.....	.....	.....	**28.14**	.....	**40.49**	.....	49.61	.....	58.97
8	8.625	74.39	641.6	5,534	.....	57.72	**63.35**	**72.49**	88.73	**105.7**	121.3	140.5	153.7	165.9
10	10.75	115.6	1,242	13,355	.....	113.7	**137.4**	**160.7**	**212.0**	244.8	286.1	324.2	367.8	399.3
12	12.75	162.6	2,073	26,426	.....	191.8	**284.4**	300.3	400.4	475.1	561.6	641.6	700.5	781.1
14 O.D.	14.0	196.0	2,744	38,416	255.3	314.4	372.8	429.1	562.3	687.3	824.4	900.3	1,027	1,117
16 O.D.	16.0	256.0	4,096	65,536	383.7	473.2	562.1	731.9	932.4	1,156	1,364	1,355	1,760	1,868
18 O.D.	18.0	324.0	5,832	104,976	549.1	678.2	930.3	1,172	1,458	1,833	2,180	2,453	2,749	2,983
20 O.D.	20.0	400	8,000	160,000	756.4	1,113	1,457	1,703	2,257	2,772	3,250	3,754	4,216	4,534
24 O.D.	24.0	576	13,824	331,776	1,315	1,942	2,843	3,424	4,521	5,672	6,740	7,617	8,625	9,367
30 O.D.	30.0	900	27,000	810,000	3,206	5,042	6,224							

[1] For dimensions and inside diameter functions see Tables IX to XII for ASA B36.10 pipe, pp. 365 to 368.
[2] Outside diameter functions are the same for all NPS pipe.
[3] Moments of inertia shown in boldface type in Schedules 30 and 40 are identical with moments of inertia for standard-weight steel pipe in former tables; those in Schedules 60 and 80 are identical with moments of inertia for extra-strong steel pipe.

Although the moments of inertia computed for nominal schedule thicknesses of wrought-iron pipe differ somewhat from those computed for steel, the differences are well within the manufacturing tolerance permitted with both kinds of pipe.

$3EI\Delta/l^2$. The stress is found from the ordinary relation $S_l = Mr_0/I$. Substituting the equivalent of M in this equation, $S_l = 3EI\Delta r_0/l^2I$, in which the moments of inertia cancel, thus giving the maximum stress in a straight cantilever in terms of the deflection at the end as $S_l = 3E\Delta r_0/l^2$. The moment of inertia,

TABLE XI.—CROSS-SECTIONAL AREA OF METAL FOR PIPE SCHEDULES, ASA B36.10*

Nominal pipe size, inches	Cross-sectional area of metal in pipe wall, square inches									
	Schedule 10	Schedule 20	Schedule 30	Schedule 40	Schedule 60	Schedule 80	Schedule 100	Schedule 120	Schedule 140	Schedule 160
⅛	.....	.....	.....	**0.072**	.....	**0.093**				
¼	.....	.....	.....	**0.125**	.....	**0.157**				
⅜	.....	.....	.....	**0.167**	.....	**0.217**				
½	.....	.....	.....	**0.250**	.....	**0.320**	.....	.....	.....	0.3836
¾	.....	.....	.....	**0.333**	.....	**0.433**	.....	.....	.....	0.5698
1	.....	.....	.....	**0.494**	.....	**0.639**	.....	.....	.....	0.8365
1¼	.....	.....	.....	**0.669**	.....	**0.881**	.....	.....	.....	1.107
1½	.....	.....	.....	**0.799**	.....	**1.068**	.....	.....	.....	1.429
2	.....	.....	.....	**1.075**	.....	**1.477**	.....	.....	.....	2.190
2½	.....	.....	.....	**1.704**	.....	**2.254**	.....	.....	.....	2.945
3	.....	.....	.....	**2.228**	.....	**3.016**	.....	.....	.....	4.205
3½	.....	.....	.....	**2.680**	.....	**3.678**				
4	.....	.....	.....	**3.173**	.....	**4.407**	.....	5.578	.....	6.621
5	.....	.....	.....	**4.304**	.....	**6.112**	.....	7.953	.....	9.696
6	.....	.....	.....	**5.584**	.....	**8.405**	.....	10.705	.....	13.32
8	.....	6.57	**7.26**	**8.396**	10.48	**12.76**	14.96	17.84	19.93	21.97
10	.....	8.24	**10.07**	11.90	**16.10**	18.92	22.63	26.24	30.63	34.02
12	.....	9.82	**12.87**	15.77	21.52	26.03	31.53	36.91	41.08	47.14
14 O.D.	10.80	13.42	16.05	18.31	24.98	31.22	38.45	43.17	50.07	55.63
16 O.D.	12.37	15.38	18.41	24.65	31.62	40.14	48.48	56.56	65.74	70.85
18 O.D.	13.94	17.34	24.11	30.79	38.98	50.23	61.17	70.28	80.66	89.34
20 O.D.	15.51	23.12	30.63	36.15	48.95	61.44	73.63	87.18	100.33	109.9
24 O.D.	34.36	27.83	41.39	50.31	67.89	87.17	106.03	122.33	142.11	157.5
30 O.D.	29.10	46.34	57.68							

* Areas shown in bold-face type in Schedules 30 and 40 are identical with areas for standard-weight steel pipe; those in Schedules 60 and 80 are identical with areas for extra-strong steel pipe.

as may be observed from this formula, has no effect on the stresses in the piping for all cases in which the lengths of pipe are straight. For curved pipes a change in the wall thickness has an effect on the stress modification factors which compensate for flattening of the cross section of a curved pipe during flexure. This effect of varying the wall thickness must be taken into account as explained on pages 816 to 818.

Outside diameter functions and moments of inertia for ASA B36.10 pipe schedules are given in Table X. For special thicknesses, the moment of inertia may be computed from the following formula:

$$I = \frac{\pi}{64}(D^4 - d^4) = \frac{\pi}{4}(r_0^4 - r_1^4)$$

where D and r_0 represent the outside diameter and radius, respectively, of the pipe, and d and r_1, the inside diameter and radius, all in inches.

CHAPTER VIII

STEAM POWER-PLANT PIPING

Basic Considerations of Good Design.—The merit of a power-plant piping layout, with due respect to economic considerations, should be judged on three basic points: (1) its mechanical design or ability to function properly (and efficiently) with respect to the mechanical equipment which it serves; (2) its convenience from an operating standpoint; (3) its appearance as a coordinated and symmetrical part of the entire plant. Although the relative importance of these basic points obviously falls in the order named, each has an important bearing on the acceptability of any layout. The refinement with which each is carried out should be based on economic considerations where extra cost is required to obtain the preferred result. In the case of a temporary job, it is sound business to eliminate all unnecessary refinements in design and make the installation in the cheapest way that will serve the purpose, but with a power plant having an expected life of 20 years or so, a more efficient, more convenient, and better appearing layout will be justified. While the monetary value of such features of (1) as reduced pressure drop, superior insulation, etc., can be evaluated readily and made a basis of economic consideration, the intangible advantages of (2) and (3) in greater operating convenience and improved appearance should not be ignored. The three basic points named are discussed at length in succeeding sections.

GENERAL FEATURES OF MECHANICAL DESIGN

Basic Principles.—The general principles of mechanical design, such as strength of parts, frictional resistance and flow of fluids, heat transmission, thermal expansion, etc., are discussed in other chapters of this handbook. The present section deals with the specific application of such principles to power-plant design and with the problem of obtaining piping layouts which will function properly in relation to the mechanical equipment they serve, *i.e.*, provide the proper circuits for fluid flow, necessary valves, by-

passes, drainage, etc., to ensure satisfactory operation. In modern power plants with the extra complications in piping incidental to the use of fuel-saving equipment, such as economizers, bleeder heaters and evaporators, deaerating heaters, and reheating boilers, the problem of securing satisfactory piping layouts is far more difficult than in the simpler plants of a few years ago. The present trend toward higher steam pressures and temperatures also adds to the necessity for greater refinement in design than previously existed.

Drawings and Line Diagrams.—On many small jobs it is common practice to set up a pipe bench in the field and cut and fit pipe on the job. This seems a simple and direct procedure, but the result is often a sloppy job at an increased cost. In almost every case it pays well in time, money, and general satisfaction to make a detailed layout on the drafting board before starting work. On more extensive jobs, the multiplicity of pipe lines now required for the interconnection of different apparatus demands careful study to define the function of each line and make its purpose clear in the layout, as well as to ensure that it is physically possible to install the piping in a neat and accessible manner without encroaching upon space required for equipment, walkways, stairs, light wells, or other structural requirements.

With involved piping layouts it is customary for someone to make a freehand sketch as a guide for the draftsman, or such a sketch may be made by the draftsman himself and referred to the proper authority for approval before the drawing proper is started. On a large job where considerable engineering study is involved, the engineer who works out the plant heat balance usually makes line diagrams pertaining to heat interchangers in connection with his studies. Such diagrams may be either isometrics or of the style resembling wiring diagrams. In either case they should show, by outline or conventional symbol, the equipment that is to be interconnected, the function of the different lines, and, usually, the condition and quantity of the fluid flowing through each. Such a diagram furnishes a good starting point for the piping studies proper.

A flow diagram of steam, condensate, and feed-water piping for a 75,000-kw 815-lb 900 F unit is shown in Fig. 1. This diagram might be termed a "functional diagram" since the boiler, turbine, condenser, and auxiliaries are represented by shapes which, although simple, indicate at a glance the function of each piece

of equipment. Somewhat simpler representations of pumps and motors might be used, but this diagram is deemed a good compromise between the artistic reproductions of turbines and boilers sometimes portrayed and the method employing symbols which have no physical significance.

The involved nature of many jobs makes imperative the use of a line diagram as a guide in laying out the piping, for explanation

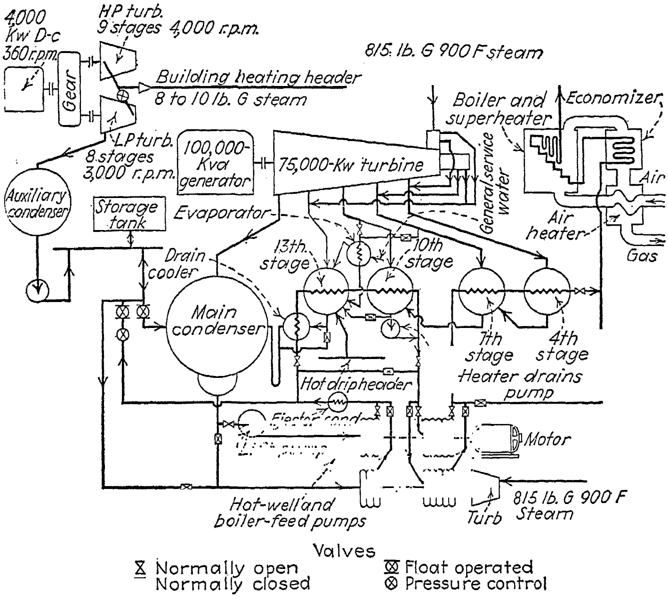


FIG. 1.—Flow diagram for 75,000-kw unit.

to the erection crew and for the intelligent operation of the system. When a final piping layout is finished and the drawings are nearly ready to be sent out for bids or to the field erection crew, it is often desirable to reproduce the line diagram in a corner of the layout sheet. Here it is readily available and proves a material aid to ready understanding of the piping system.

When the basic principles of interconnecting the equipment have been portrayed in line diagrams and duly approved, the next step is the preparation of assembly drawings in plan and

elevation showing structure and outlines of equipment with the piping laid out to scale. In the preparation of such drawings due consideration should be given to the principles of design discussed in following paragraphs. In addition to the assembly drawings, it is often advisable to prepare separate layouts for the more important lines drawn to scale and with piping located by dimension to the building structure, especially in cases where the piping is considerably congested. For large work, requiring heavy shop fabrication of bends, fillers cut to length, etc., complete and accurate detail drawings are essential. Great care should be taken in checking such details, since any errors are likely to involve considerable expense and serious delay in returning material to a distant shop for correction.

Determining Pipe Sizes.—Before proceeding beyond a preliminary drafting layout of piping systems, it is necessary to determine pipe sizes which allow reasonable velocities and friction losses. The maximum allowable velocity of the fluid in a pipe line is that which corresponds to the permissible pressure drop or friction loss from the point of supply to the point of consumption. In allowing for the loss through fittings and valves, it is customary to count each as equivalent to so many feet of straight pipe and take the total drop for the whole line as that corresponding to so many hundred feet of straight pipe. Methods of calculating pressure drop incidental to the flow of various fluids are discussed at length in the section on "Friction Loss in Pipes" (starting on page 81) and in the respective sections on specific fluids in Chap. II. From the standpoint of investment cost, it is desirable to keep the velocity as high as possible without exceeding the maximum allowable velocity or causing excessive operating losses as a result of unduly great pressure drop in pipes.

In many cases where a pipe line or branch is short, the labor of calculating pressure drop is not warranted. The values given in Table I are reasonable for use in such cases if we bear in mind that the lower velocities should be used for small pipes and the upper limits for large ones. These values represent good average practice and may be used as a guide in many cases where actual pressure drops are not computed.

Allowable steam velocities in turbine leads are considerably higher than in reciprocating-engine practice, since there is no pulsating inertia effect to consider with turbines. A steam receiver close to the steam chest of reciprocating engines is frequently

employed to steady steam flow and damp out pulsations in the line. Another factor affecting permissible steam velocities is erosive action on valve seats and similar exposed parts. This action is much more pronounced in the case of wet steam than with superheated, and velocities should be correspondingly lower when there is much moisture in steam.

TABLE I.—REASONABLE VELOCITIES FOR FLOW OF FLUIDS IN PIPES WHERE ACTUAL PRESSURE DROPS ARE NOT COMPUTED

Fluid	Pressure pounds per square inch gage	Use	Reasonable velocity, feet per minute
Water	25 to 40	City water	120 to 300
Water	50 to 150	General service	300 to 600
Water	150 up	Boiler feed	600
Saturated steam	0 to 15	Heating	4,000 to 6,000
Saturated steam	50 up	Miscellaneous	6,000 to 10,000
Superheated steam	200 up	Large turbine and boiler leads	10,000 to 15,000

In ordinary power-plant work (as distinguished from steam-transmission lines, central heat-distributing systems, etc.) where the runs are reasonably short and the valves and fittings are designed to offer slight resistance, it is customary to allow maximum velocities of 1,000 to 1,200 ft per min per in. of inside pipe diameter. The higher value applies particularly to diameters over 12 in. Velocities in steam lines to reciprocating engines or pumps without ample receivers should not exceed 75 to 80 per cent of those given above. The same rule applies to excessively long lines and those where fittings or valves offer unusual resistance. The velocity rules noted above seem to apply about equally well to both high- and low-pressure steam, since the reduction in friction at low pressures due to the reduced density about offsets the condition that less initial pressure is available to furnish the pressure drop.

Too high velocities in pipes carrying water or other liquids are objectionable from the more severe water hammer which then accompanies a sudden stoppage or abrupt change in velocity. The theory and magnitude of water hammer are explained at length starting on page 291. The water velocities given for various services in Table I usually will be found satisfactory from a standpoint of water hammer. As a general rule, it is not advisable to employ water velocities much exceeding 600 ft per min, both

because of the severity of water hammer and because the pressure drop mounts rapidly above that velocity.

An illustration of extremely high steam velocities, which are sometimes used to advantage, is offered in some central heating practice. Steam velocities of 50,000 to 75,000 ft per min are sometimes used in steam feeder lines leading from central heating plants to remote distributing centers in the network of service lines which serve the customers (see page 1004). The significance of the case of such feeders is that the high velocity is not in itself objectionable. There is no appreciable erosion of the pipe walls and no other bad effects. The practice of considering velocities at all is somewhat of a heritage from reciprocating-engine days. It is simpler to compute velocities than pressure drops, however, and there is, of course, a definite relation between them. A given velocity produces a higher pressure drop in a small pipe than in a large one, a fact which should not be overlooked. High velocities are accompanied by considerable noise which would be objectionable in heating systems in office buildings and dwellings but not in power plants.

In selecting pipe sizes for exhaust lines from auxiliary turbines and similar services where high velocities may be used to advantage, consideration should be given to the limiting velocities which can be obtained. This limiting or "acoustic" velocity has a definite value for each combination of steam density and pressure. For calculation of acoustic velocities as an aid in selection of pipe sizes for exhaust lines and safety valve vents (see pages 254 to 264).

The elimination of a number of seldom-used pipe sizes has been accomplished through the efforts of the ASA and the Simplified Practice Division of the U.S. Department of Commerce, to the combined advantage of manufacturers and consumers. It is often cheaper to use a larger pipe than to use an odd size. On the other hand, where space is at a premium, it may be necessary to use a smaller size than would otherwise be good practice. The calculation of velocity or pressure drop is a valuable check, but in the last analysis judgment is the deciding factor, and the blind use of general rules is impracticable.

Loss Due to Steam Leaks.—The value of steam which can be lost through a comparatively small leak becomes of appreciable moment when considered over a period of time. The amounts of steam which will escape through various size orifices at different pressures can be computed by any of the flow formulas

TABLE II.—LOSS DUE TO STEAM LEAKS

(The tables below are based on coal costing $4.50 per ton in the furnace, 1 lb having a heat value of 13,000 Btu and an evaporation of 8.4 lb of feed water per pound of coal. Grashof's formula used)

orifice, inches	Pounds steam wasted per month	Cost of coal per month	Cost of water wasted at 0.10 per 1,000 gallons	Total cost per month	Total cost per year	Capitalized value at 16 per cent on savings
			250-lb. gage			
1/2	1,780,000	$477.00	$21.36	$498.36	$5,980.32	$37,377.00
3/8	1,001,000	268.00	12.01	280.01	3,360.12	21,000.00
1/4	445,000	119.00	5.34	124.34	1,492.08	9,325.00
1/8	111,000	29.70	1.33	31.03	372.36	2,327.00
1/16	27,800	7.45	0.33	7.78	93.36	583.00
1/32	7,000	1.87	0.08	1.95	23.40	146.00
			300-lb. gage			
1/2	2,125,000	$573.00	$25.50	$598.50	$7,182.00	$44,887.00
3/8	1,195,000	322.00	14.34	336.34	4,036.08	25,225.00
1/4	531,000	143.00	6.37	149.37	1,792.44	11,200.00
1/8	132,800	35.80	1.59	37.39	448.68	2,804.00
1/16	33,200	8.95	0.40	9.35	112.20	701.00
1/32	8,300	2.24	0.10	2.34	28.08	175.00
			400-lb. gage			
1/2	2,804,000	$760.00	$33.65	$793.65	$9,523.80	$59,524.00
3/8	1,577.000	427.00	18.92	445.92	5,351.04	33,444.00
1/4	701.000	190.00	8.41	198.41	2,380.92	14,880.00
1/8	175,200	47.50	2.10	49.60	595.20	3,720.00
1/16	43,800	11.85	0.53	12.38	148.56	928.00
1/32	11,000	2.98	0.13	3.11	37.32	233.00
			600-lb. gage			
1/2	4,157,000	$1,141.00	$49.88	$1,190.88	$14,290.56	$89,312.00
3/8	2,338,000	642.00	28.06	670.06	8,040.72	50,254.00
1/4	1,039,000	285.00	12.47	297.47	3,569.64	22,310.00
1/8	259,700	71.25	3.12	74.37	892.44	5,577.00
1/16	65,000	17.85	0.78	18.63	223.56	1,397.00
1/32	16,300	4.47	0.20	4.67	56.04	350.00

given on pages 74 to 81. Table II, which was computed by Grashof's formula, gives the pounds of steam lost per month and the corresponding money value of the coal and water wasted.

Safety Codes.—Extensive safety requirements for power-piping systems are contained in Section 1 of the American Standard Code for Pressure Piping ASA B31.1, which covers the design, manu-

facture, test, and installation of power-piping systems for steam-generating plants, central-heating plants, and industrial plants. In designing the component parts of piping systems within the jurisdiction of this code, reference should be made to its provisions as representing a standard for minimum safe requirements, but it is not intended to indicate necessarily the best practice known to the art. Requirements of the Code for Pressure Piping are not compulsory since so far it has not been adopted as law by any state. It is in common use, however, and frequently is referred to in contract specifications and similar documents.

On the other hand, a number of states and municipalities have adopted the ASME Boiler Construction Code as law and therefore any piping within its jurisdiction must conform to Boiler Code requirements. Although it would seem that the Boiler Code should cover only such piping as is directly associated with the boiler, its provisions have been interpreted to include considerable main-steam and boiler-feed piping not directly pertaining to the boilers. Hence careful consideration should be given to the Boiler Code and the applicability of its rules to certain portions of many power-piping installations.

Considerable progress has been made in harmonizing the piping requirements of the two codes as evidenced by the formulas for pipe-wall thickness common to both, which are abstracted on pages 42 to 47. The basic assumptions as to design pressure differ between the codes, however, so that no one schedule of pipe wall thicknesses or series of fittings necessarily is acceptable under both.[1]

Selection of Dimensional Standards and Materials.—The selection of suitable dimensional standards for flanges, fittings, valves, and pipe for ordinary service conditions can be made from the American Standards reproduced in Chap. IV. Adjusted ratings at temperatures above and below 750 F for carbon and alloy steels are given in the steel-flange standards to govern their use under pressure or temperature other than the primary service ratings.

Selection of materials for temperatures above 750 F from the various grades of alloys described in ASTM Specifications A158, A157, A193, A194, and A206, is facilitated by reference to the

[1] Where piping comes under the jurisdiction of a particular code, the latest revision should be used. Copies may be obtained from the American Society of Mechanical Engineers, 29 West 39th St., New York 18, N.Y.

abstracts of these specifications given in Chap. IV and to the discussion of materials and working stresses given in Chap. III. The multiplicity of services in a large plant and the variety of dimensional standards and materials, possible joints, and different types of welding available make it desirable to prepare some form of design instructions for use in the drafting rooms and by the construction force to ensure that proper standards, design details, and materials are secured. A summary of the principal selections for an 850-lb 925 F plant is given in Table III.

It is advisable to express a word of caution regarding the connection of cast-iron flanges and fittings to steel flanges and fittings. The 25-lb and 125-lb cast-iron flanges have plain faces while the 250-lb flanges have raised faces which come out practically to the inner edge of the bolt holes, whereas those of steel flanges are narrower and stop some distance inside the bolt holes. The plain face or wide raised face on cast-iron fittings and flanges is a necessary precaution to prevent cracking the flange in drawing up the bolts. Numerous instances have been observed where cast-iron flanges have cracked when being bolted to steel flanges having a narrow raised face. In cases where it is necessary to bolt cast-iron and cast-steel flanges together, either the steel flange should be made with a wide raised face, or its raised face should be machined down flush with the flange edge. For the same reason, when lap-joint pipe is made up with cast-iron flanges, the lapped end should be brought out to the inner edge of the bolt holes. Failure to observe these precautions is liable to be costly in both replacements and time lost getting new material. Instances have been known where lap-joint pipe with steel flanges and a narrow lap has been bolted to cast-iron flanges on boiler-feed pumps with disastrous results. Cracking a pump-discharge flange means the replacement of the whole pump casing, frequently at a cost of several hundred dollars.

For reasons similar to those stated above, it is inadvisable to use alloy steel bolts in cast-iron flanges. Commercial carbon-steel bolts are amply strong for use in cast-iron flanges, and there is no occasion to risk cracking such flanges through the use of too strong bolts.

American Standard steel flanges are properly designed, both as regards dimensions and material, for use with a narrow raised face and alloy-steel bolts. Intelligent selection may be made from the three grades of alloy-steel bolts listed in ASTM Specification

TABLE III.—TYPICAL SELECTION OF MATERIALS AND DIMENSIONAL STANDARDS FOR AN 850-LB 925 F STEAM PLANT

	Pipe schedule and specification number	Style of joint, nominal sizes used, inches			Flanges and fittings, material and pressure standard			Valves, material and pressure standard		
		Screwed	Flanged	Welded	Screwed	Flanged	Welding	Screwed	Flanged	Welding
Air, compressed	40-A120	3—1	4+	3 — S[2] 2½ + B	Malleable iron 150	Cast iron[6] 125	Steel S-Sch 40[3] B-Sch 40[4] F-150[5]	Brass 125	Cast iron 125	
Blowoff:										
Boiler to last valve	80-A106	....	¾+	3 — S	....	Steel 1500	Steel S-Sch 80 F-1500			
Beyond last valve	80-A106	....	2½+	3 — S 2½ + B	....	Steel 600	Steel S-Sch 80 B-Sch 80 F-600			
Evaporator	40-A135	2 and 3	4+	3 — S 2½ + B	Cast iron 125	Cast iron[6] 125	Steel S-Sch 40 B-Sch 40 F-150	Brass 125	Cast iron 125	
Condensate:										
250 lb	40-A53[7]	½ to 2	2½+	3 — S 2½ + B	Steel 3000	Cast iron 250, steel 300	Steel S-Sch 40 B-Sch 40 F-300	Steel 300	Cast iron 250	
100 lb	40-A53	½ to 2	2½+	3 — S 2½ + B	Cast iron 125	Cast iron 125	Steel S-Sch 40 B-Sch 40 F-150	Brass 125	Cast iron 125	
Drain, equipment	40-A120[8]	3—	4+	....	Cast iron 125 Malleable iron 150	Cast iron[6] 125	Steel B-Sch 40 F-150	Brass 125	Cast iron 125	
Drip:										
Superheated	160-A206	....	2½+	3 — S 2½ + B	....	....	Steel S-Sch 160[10] B-Sch 160 F-1500	......	Steel[11] 1500	Steel[11] 1500

975-lb saturated..........	80-A106[9]	....	2½+	3 − S 2½ + B	....	Steel 1500	Steel S-Sch 80 B-Sch 80 F-1500	......	Steel 1500	Steel 1500
300-lb saturated..........	Th-80[12] W-40 A-53	½ to 2	2½+	3 − S 2½ + B	....	....	Steel S-Sch 40 B-Sch 40 ½ to 1½ F-600 2 + F-300	Steel 600	Steel 300	Steel ½ to 1½ 600 2 + 300
Exhaust:										
Turbine bleeder..........	40-A53[13] 12 + ⅜-in. wall	½ to 1¼[14]	2½+	3 − S 2½ + B	....	Steel 600 300 150 Cast iron 125	Steel S-Sch 40 B-Sch 40	......	Steel 600 300 150 Cast iron 125	
Auxiliary turbine..........	⅜-in. wall A-135	....	12	12B	....	Steel 150	Steel fabricated from plate and pipe			
Feed water:										
Connections to boiler.......	¾ to 4 Sch 80[15] 6-Sch 120 8-Sch 100	....	2½+	3 − S 2½ + B		3 to 8 Alloy[16] 900 ½ to 2½ Steel 1500	Steel S-Sch 80 2½ to 4 B-Sch 80 6-in. B-Sch 120 8-in. B-Sch 100 Alloy F-900	......		3 to 8 Alloy[16] 900 ½ to 2½ Steel 1500
System..................	80-A106	....	2½+	3 − S 2½ + B	....	Steel 900	Steel S-Sch 80 B-Sch 80 F-900	......	....	Steel S-Sch 80 B-Sch 80 900
Instrument:										
High pressure.............	160-A206 80-A106	....	¾+	3 − S	....	....	Steel S-Sch 80 F-1500	Steel[17] 3000	Steel 1500	
Low pressure.............	40-A120[18]	3− (flared)	4+	3 − S	Steel 3000	Cast iron[6] 125	Steel S-Sch 40 F-150	Brass 125		
Oil:										
Turbine lubricating.........	40-A53	....	¾+	3 − S 4 + B	....	Steel 150 Cast iron 125	Steel[19,20] S-Sch 40 F-150 F-400 F-600	......	Steel 150	Steel 150

TABLE III.—(*Concluded*)

	Pipe schedule and specification number	Style of joint, nominal sizes used, inches			Flanges and fittings, material and pressure standard			Valves, material and pressure standard		
		Screwed	Flanged	Welded	Screwed	Flanged	Welding	Screwed	Flanged	Welding
Oil (Cont.) Transformer	40-A120	3—	4+	3 − S 2½ + B	Cast iron 125	Cast iron[6] 125	Steel F-150	Brass 125	Cast iron 125	
Water: City and general service, inside.	40-A120[21]	3—	4+	3 − S 2½ + B	Malleable iron 150 galvanized	Cast iron[6] 125	Steel S-Sch 40 B-Sch 40 F-150	Brass 125	Cast iron 125	Steel 150
City and general service, outside.	40-A120[22]	3—	4+	3 − S 2½ + B	Malleable iron 150 galvanized	Cast iron[6] 125	Steel S-Sch 40 B-Sch 40 F-150	Brass 125	Cast iron 125	Steel 150
Steam: Main and auxiliary superheated.	¾ to 3 160-A206 4 and 6 120-A206 8+ 100-A206	....	2½+	3 − S 2½ + B	....	Alloy[23] 1500	Alloy[23] S-Sch 160[24] F-1500	......	Alloy[23] 1500	Alloy[23] 900
975-lb saturated	80-A106[9]	....	2½+	3 − S 2½ + B	....	Steel ½ to 2½ 1500 3 + 900	Steel S-Sch 80[10] B-Sch 80[10] ½ to 2½ F-1500 3 + F-900[25]	......	Steel ½ to 2½ 1500 3 + 900	Steel ½ to 2½ 1500 3 + 900
300-lb saturated	40-A53	....	2½+	3 − S 2½ + B	....	Steel 300	Steel S-Sch 40 B-Sch 40 ½ to 1½ F-600 2 + F-300	Steel 600	Steel ½ to 1½ 600 2 + 300	Steel ½ to 1½ 600 2 + 300

Vacuum	40-A135 -A120	3—	4+	3 — S 2½ + B	Cast iron 125	Cast iron 125 Steel 150	Steel S-Sch 40 B-Sch 40	Brass 125	Cast iron 125 Steel 150	Steel 150
Vent:										
Safety valve	40-A120 -A135	3—	4+	3 — S 2½ + B	Cast iron 125 Steel 3000	Cast iron[6] 125 Steel 150	Steel S-Sch 40 B-Sch 40 F-1500[26]	Brass 125	Cast iron 125	
Hotwell pump	Copper									

Gaskets.—Red rubber in joints up to 250 F, asbestos composition up to 800 F, profile serrated monel for 925 F.

Joint Facings.—ASA Standard raised faces, serrated for asbestos gaskets, ground face for monel gaskets.

Bolting.—Cast-iron flanges: use square- or hexagonal-head machine bolts with heavy hexagonal nuts. Steel flanges: for services up to 800 lb, 750 F, use bolt-studs, ASTM A96, Class A; for services up to 1,200 lb, 750 F, use ASTM A96, Class C; for temperature of 925 F, use ASTM A193, Grade B14 bolt studs with nuts to ASTM A194, Class 2H.

1 3— is used to indicate sizes 3 in. and smaller; 4+, 4 in. and larger, etc.

2 S indicates socket-welded; B, butt-welded.

3 S-Sch 40 refers to Schedule 40 fittings of the proposed American Standard for Steel Socket-welding Fittings, ASA B16.11 (see p. 505).

4 B-Sch 40 refers to Schedule 40 fittings of the American Standard for Steel Butt-welding Fittings, ASA B16.9 (see p. 502).

5 F-150, F-1500, F-600, etc., indicate standard of welding-neck flanges, ASA B16e (see p. 626), or slip-on welding flanges for 150 and 300 psi.

6 Screwed cast-iron companion flanges may be used in these services.

7 Seamless A53 or electric-resistance welded A135 also suitable.

8 Also A135, copper Class L with soldered connections extra-heavy cast-iron soil pipe and Class 50 water pipe.

9 Also seamless A53.

10 Carbon steel WF, A234.

11 Cast or forged carbon steel adequate because of intermittent operation and low average temperature of drip lines.

12 Th = threaded, W = welded seamless A53 or A106.

13 Seamless low or medium carbon A53, or A106 for two high-pressure bleed lines, electric-resistance-welded A135 for lower pressure lines.

14 Threaded nipples for highest pressure line = double-extra-strong, lower pressure lines ½ to ¾ in.; Schedule 80, 1 and 1¼; Schedule 40.

15 Thicknesses of seamless Grade B A106 pipe required to meet Boiler Code requirements for 975-lb boiler drum pressure.

16 Carbon-molybdenum alloy steel required by Boiler Code to permit use of 900-lb standard valves or fittings.

17 Screwed 3,000-lb valves remote from source.

18 Also copper tubing and thin-walled steel tubing with flared-tube connections or soldered copper connections.

19 Connections to turbine ¾ to 3 in., 600-lb, 4 in. and larger 400-lb with large male-female facing.

20 Welding-neck flanges for connection to cast-iron pump and oil-strainer flanges have raised faces omitted, except in the 3, 6, and 8 in. sizes.

21 Sizes ½ to 3 in. galvanized, 4 to 10 black; 12 in. and larger ⅜-in. wall.

22 Also Class 150 cast-iron, bell-and-spigot joints; lead (extra-strong quality) for city water; and copper Class K. See Chap. IV for standards.

23 Chrome-nickel-moblydenum, Grade WC 4, A127, or carbon-molybdenum, Grade WC 1, A217.

24 Carbon-molybdenum, Grade WF_1, A234.

25 Except terminating flanges on connections to boiler drum which are 1,500 lb.

26 Vent from economizer relief valve; 1,500-lb flanges and valve.

A96 and the bolting for temperatures from 750 to 1000 F given in ASTM Specification A193 (see Chap. IV, page 539).

The use of carbon-steel nuts with alloy-steel studs has been found generally satisfactory for temperatures up to 750 F, since the mechanical construction of the nut is stronger than that of the stud. However, owing to their greater tendency to splitting, bar-stock nuts machined from hot-rolled or cold-drawn bars, should not be used for severe service conditions.

For higher temperatures, low-sulphur nuts and oil-quenched nuts are used as described in ASTM specification A194 (see Chap. IV).

Selection of Valves.—A selection of the proper value for a particular purpose depends, first, on the operating pressure and temperature and, second, on the type of valve best suited for the use to which it will be put. In some cases, where either a globe or gate valve would serve equally well, the decision between these types may be based on price considerations. The more common types of valves are illustrated in Chap. IV.

In general, it is customary to use *gate valves* in locations where pressure drop through the valve is a consideration, and where the valve will be either wide open or entirely closed. Guard valves and shutoffs for boiler and turbine leads, etc., are almost always of the gate type. *Globe valves* are seldom used in water lines because of the large friction losses involved in water flow through such valves. Globe valves are commonly used in steam and air lines for throttling purposes, as the globe type permits closer regulation of flow. Throttling usually involves more or less cutting of the seat and disk, and these parts in globe valves are less expensive and more easily replaced than in gates. Among such uses for globe valves might be mentioned turbine and engine throttles, by-passes around traps or reducing valves, hand-feed regulation on boilers, etc. A gate valve should always be installed preceding a globe valve used for throttling.

It is always advisable to use *check valves* in individual pump or trap discharges before they join a common header, and where different lines are joined together to discharge into a common header. It should be borne in mind that a check valve cannot be counted on for closing a line off tightly against pressure working back through, but it will stop any considerable flow. In pump discharges where the header remains under pressure after the pump is shut down, a gate valve should be provided in addition to the check. It is also desirable to provide a small relief valve on the

pump suction to prevent pressure backing up through the pump and damaging the foot valve while the pump is shut down. Check valves are required in feed lines close to a boiler to prevent water or steam blowing back from the boiler if, for any reason, the feed line ruptures or its pressure fails.

With *reducing valves* it is desirable to select a size that is loaded somewhere near capacity under normal operation, as such valves are then more stable in their operation. If there is considerable seasonal variation in the load on a reducing station, as in furnishing steam for building heating, it is good practice to install a large and a small valve in parallel and use the one best fitting the load at any particular time. It is frequently desirable to install a hand-operated by-pass around a reducing valve so that service can be maintained while the reducing valve is being repaired. Such an arrangement with provision for shutoff to repair either

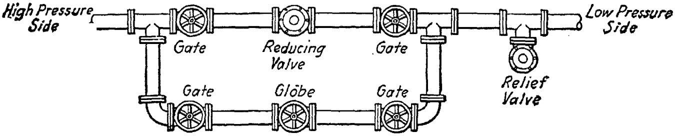


FIG. 2.—Pressure-reducing station.

the reducing valve or hand-throttling valve while the line is in operation is shown in Fig. 2. It is worth mention that filler pieces should be provided between a flanged reducing valve and the adjacent gates, as reducing valves usually are constructed so that it is impossible to remove all the end-flange bolts when they are bolted directly to other valves or fittings, owing to close clearance at the valve bonnets or fitting necks. For convenience in removing these valves from the line, it is desirable to use flanged connections at such points, even though the rest of the line is made up with screwed joints. Another point of caution is that when reducing from a pressure that requires the use of a heavy standard for flanges and fittings to a pressure with which a lighter standard is used or to which low-pressure equipment is connected, relief valves should be provided on the low-pressure side. The use of the heavier standard should be continued through to the last valve ahead of the relief valve, as it is possible to have full pressure up to that point. It should always be borne in mind that a reducing valve seldom or never has a tight shutoff, and, at times of negligible steam consumption, leakage through the valve

is apt to be enough to build up full-line pressure on the low-pressure side. The *Power Section* of the *Code for Pressure Piping* requires

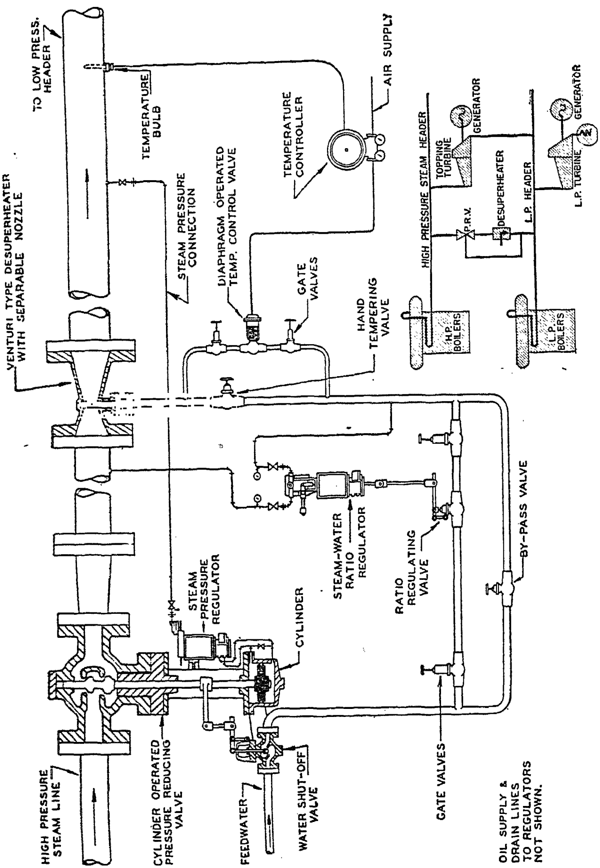


FIG. 3.—Typical pressure-reducing and desuperheating station of the "ratio" type.[1]

that the combined discharge capacity of the safety or relief valves

[1] This diagram was prepared expressly for the "Piping Handbook" by the Republic Flow Meters Company which manufactures the Smoot equipment illustrated. The sketch at the lower right-hand corner of Fig. 3 is reproduced by permission from an article by Mr. C. R. Earle in the April, 1938, issue of *Power Plant Engineering*.

shall be such that the pressure rating of the lower pressure piping will not be exceeded in case the reducing valve sticks open. An exception may be made in a steam plant used for district heating where attendants are always on duty. In this case, a pressure operated audible alarm may be installed in place of safety valves.

The advent of "topping" or superposed plants in which high-pressure high-temperature steam is passed through a back-pressure turbine to supply a number of lower pressure condensing turbines has evolved the need for extremely large pressure-reducing and desuperheating stations to enable steam to be supplied from high-pressure boilers if the back-pressure turbine is out of service or where reduced-pressure steam is required to operate low-pressure standby units.

A typical pressure-reducing and desuperheating station of the "ratio" type used extensively for this service is shown in Fig. 3. Other types known as "tank" and "atomizing" desuperheaters also are in common use.

Safety and Relief Valves.—Where more than one safety or relief valve is used on a boiler or other pressure vessel, it is desirable to set one or more of the valves to relieve at a lower pressure than the rest. This serves as a warning before too much steam is lost through all the valves opening at once, and also tends to facilitate repairs by confining any cutting action to the one or more valves that open first. In some cases an extra safety valve is set to blow before the others and is mounted above a gate valve, so that it can be removed for repairs while the boiler or steam line is in service. Of course, the capacity of this valve cannot be considered in meeting code or other safety requirements, since it might be shut off. Where the hazard involved does not require the installation of a full-size relief valve, it is sometimes desirable to install a small-size pop valve as a telltale to give warning when the usual working pressure is exceeded, so that the operator will attend to restoring normal conditions. A safety valve for use with an expansion fluid, such as steam or air, is distinguished from a relief valve in that a safety valve has an adjusting, or huddling, ring and chamber to control the amount the pressure blows down before the valve reseats. A safety valve with outside spring suitable for use with either saturated or superheated steam is shown in cross section in Fig. 36 on page 565 with the various parts labeled. A schedule of approximate relieving capacities for different size safety valves is given in Table LIX on page 566

for saturated steam. Actual values for any particular valve, or for other fluids, should be taken from the manufacturer's catalogue.

Safety Valves for Power Boilers.—The construction and method of installing safety valves for power boilers can best be explained by reference to Section I of the ASME Boiler Construction Code[1] which is incorporated in the safety laws of many states. The following material is taken from the 1943 edition of that code. For latest and more complete information, reference should be made to the latest revision of the Code itself.

Each boiler is required to have at least one safety valve and two or more safety valves if it has more than 500 sq ft of heating surface or if the steam-generating capacity exceeds 2,000 lb per hr. The safety-valve capacity for each boiler is required to be such that all the steam that can be generated will be discharged without allowing the pressure to rise more than six per cent above the highest pressure at which any valve is set and in no case more than 6 per cent above the maximum allowable working pressure. The maximum steaming capacity shall be determined by the manufacturer.

One or more safety valves on the boiler proper are required to be set at or below the maximum allowable working pressure. If additional valves are used, the highest pressure setting is required not to exceed the maximum allowable working pressure by more than 3 per cent. The complete range of pressure settings of all the saturated-steam safety valves on a boiler is required not to exceed 10 per cent of the highest pressure to which any valve is set.

Safety valves are required to be of the direct spring-loaded pop type. The maximum rated capacity stamped on the valve is required to be 90 per cent of the actual steam flow determined by tests in the presence of authorized inspectors at a pressure 3 per cent in excess of the pressure at which the valve is set to blow. The blowdown, *i.e.*, difference between opening and closing pressure, and capacity lift or distance the valve seat rises when blowing under a pressure of 3 per cent above the set pressure also are required to be stamped on the valve.

When two or more safety valves are used on a boiler, they may be mounted either separately or as twin valves on Y bases or duplex valves having two valves in one body. Valves mounted as twin valves and duplex valves shall be of equal sizes. Where not more than two valves of different sizes are mounted singly, the relieving capacity of the smaller is required to be not less than 50 per cent of the larger valve.

The safety valves are required to be connected to the boiler independent of other steam connection. Intervening pipe or fittings are required to be not longer than the corresponding tee fitting of the same diameter and pressure of the American Steel Flange Standard (see ASA B16e abstracted on pages 626–680).

The area of the connection between the boiler and safety valve is required to be at least equal to that of the valve inlet. No valve of any description is permitted between the safety valve and the boiler or on the discharge pipe between safety valve and atmosphere. The area of the discharge pipe, if used, is required to be not less than the area of the safety-valve outlet, or outlets

[1] Published by the American Society of Mechanical Engineers, 29 West 39th St., New York 18, N.Y.

if more than one valve discharges into a common discharge pipe. If a muffler is used on a safety valve, it is required to have sufficient outlet area to prevent backpressure interfering with the operation of the valve.

Safety valves are required to operate without chattering and to be set to close after blowing down not more than 4 per cent of the set pressure but not less than 2 lb in any case. For pressure between 100 and 300 lb, inclusive, the blowdown is required to be not less than 2 per cent of the set pressure. The blowdown adjustment is made and sealed by the manufacturer. The popping-point tolerance plus or minus is required not to exceed 2 lb for pressure up to and including 70 lb, 3 lb for pressure 71 to 300 lb, and 10 lb for pressure over 300 lb.

Every attached superheater is required to have one or more safety valves near the outlet. The safety valves may be located anywhere along the length of the outlet header. The superheater safety-valve capacity may be included in determining the size and number of valves required in a boiler, provided at least 75 per cent of the aggregate valve capacity is located on the boiler.

The principles involved in the protection of low-pressure piping supplied with steam from a higher pressure source are essentially the same as the principles used in the protection of unfired pressure vessels, except that the permissible pressure rise depends upon the particular service. The following material abstracted from Section VIII of the 1943 edition of the ASME Boiler Construction Code covers capacities, recommended types, methods of installation, and permissible adjustment of relief valves.

Pressure vessels shall be protected by such safety or relief valves and indicating and controlling devices as will ensure their safe operation. The relieving capacity of safety valves shall be such as to prevent a rise of pressure in the vessel of more than 10 per cent above the maximum allowable working pressure.

Safety valves shall be of the direct spring-loaded type. A lifting device is required for trying all safety valves except those used for relief of liquid pressure. Each safety valve shall be tested once every day or oftener, by raising the disk from its seat with the lifting device.

Pipe between safety valve or valves and the vessel which is to be protected shall have an internal cross-sectional area not less than the nominal area of the safety valve or valves. There shall be no intervening valve between vessel and safety valve. The safety-valve escape or vent pipe shall be full-sized and fitted with an open drain. No valve of any description shall be placed on the escape pipe (see discussion of safety-valve vent piping, pages 890–892).

Safety valves exposed to a temperature of 32 F or less shall have a drain at least 3/8 in. in diameter at the lowest point where water can collect, except that safety valves 3/4 in. in size and less may have drain holes as large as possible but not less than 3/16 in. in diameter. Safety-valve springs shall not be adjusted to carry more than 10 per cent greater pressure than that for which they were

Safety valves for compressed-air tanks shall not exceed 3 in. in diameter and shall be proportioned for the maximum number of cubic feet of free air that can be supplied per minute.

The information on relieving capacity, given in Table IV, which appeared in the 1937 and previous editions of Section VIII of the ASME Code, has been dropped from the Code but is continued here as of interest to piping designers.

TABLE IV.—MAXIMUM RELIEVING CAPACITY OF FREE AIR IN CUBIC FEET PER MINUTE FOR DIFFERENT SIZES OF SAFETY VALVES AT STATED PRESSURES

Diameter D, inches	Gauge pressure, psi															
	50	100	150	200	250	300	350	400	500	600	800	1,000	1,200	1,600	2,000	2,400
¼	12	20	27	33	38	43	48	53	61	70	84	97	109	128	147	160
⅜	17	27	36	44	51	58	65	72	83	95	115	133	149	176	197	215
½	20	32	42	51	59	67	74	111	129	147	177	205	230	270	304	330
¾	37	59	78	96	112	127	141	176	[illegible]	232	242	346	386	423	474	518
1	58	94	124	152	178	202	224	248	286	324	390	450	500	586		
1¼	84	135	180	221	259	293	325	352	405	443	509					
1½	114	186	248	302	354	400	444	478	528	568	634					
2	189	305	410	501	592	668	741									
2½	282	457	613	750	880	998	1,114									
3	393	638	856	1,050	1,230	1,398	1,557									

The foregoing table is based on the following formulas:

$$Q = 28PDl \text{ for 45-deg bevel-seat valves}$$

and

$$Q = 40PDl \text{ for flat-seat valves}$$

where Q = discharge, cu ft of free air per min measured at 14.7 psi abs and 60 F.
P = absolute pressure at which the safety valve opens (gauge pressure plus 14.7 psi at sea level).
D = diameter, in., of the inside edge of the bearing surface between the disk and seat.
l = [illegible] discharge capacity for satisfactory operation of the valve.

Pipe Thimbles.—Where pipes pass through walls or floors or where they are to be embedded in concrete or masonry, suitable pipe sleeves or thimbles should be provided. Such thimbles preferably should be made of standard-weight wrought pipe, either black or galvanized, to provide sufficient mechanical strength and durability. Where pipes pass through floors, the thimbles should extend at least 4 in. above the top of the floor in order to prevent wash water running down the hole. A schedule of the proper sizes of standard-weight pipe to use for thimbles for any size line, screwed or flanged and with or without flanges of different standard dimensions, is given in Table V.

MECHANICAL DESIGN OF PRINCIPAL PIPING SYSTEMS

Main Steam Piping.—The main steam lines constitute the most important piping in a power plant. Several methods of connecting

boilers and turbines are in common use. In the simplest type a boiler or two boilers, as the relative capacities of the boilers and turbines may dictate, are connected directly to their respective turbines as indicated in Fig. 4*a*. This is known as the unit boiler-turbine system. Although in normal operation steam flows

TABLE V.—PIPE THIMBLE SIZES[1]
(All dimensions in inches)

Nominal pipe size	Screwed pipe					Flanged pipe		
	Without insulation	With insulation				With or without insulation		
	Any weight pipe of iron pipe, size	Wool or hair felt 1 in. thick	Standard thickness 85 per cent magnesia	Double standard thickness 85 per cent magnesia	Cork covering 1½ in. thick	125- and 150-lb. flanges	250- and 400-lb. flanges	600-lb. flanges
¼	¾	3	3	6	4	.	.	.
½	1	4	4	6	6	.	.	.
¾	1¼	4	4	6	6	.	.	.
1	1½	4	4	6	6	.	.	.
1¼	2	6	6	6	6	6	6	6
1½	3	6	6	6	6	6	8	8
2	3	6	6	8	.	8	8	8
2½	4	6	6	8	.	8	10	10
3	4	8	8	10	.	10	10	10
4	6	8	8	10	.	10	12	12
5	8	10	10	12	.	12	14	16
6	8	10	10	12	.	12	16	16
8	10	12	14	16	.	16	18	20
10	14	16	16	18		18	20	24
12	16	18	18	20	.	24	24	24
14	18	20	20	24	.	24	26	26
16	20	20	24	24	.	26	28	30
18	24	24	24	26	.	28	30	32
20	24	24	26	28	.	30	32	34
24	....	..	..	..	.	34	38	40
30	....	..	..	..	.	40	46	48

[1] When flanges are in sleeves, special provision must be made for covering. Allow 1 in. extra on diameter of 125-lb flanges for lugs. Pipe thimble sizes given are for thimbles made of standard-weight wrought pipe up to 12 in.; above 12 in., of O.D. pipe with [illegible] wall. Pipe thimbles above 24 in. should be made of sheet metal, 10 gauge.

directly from boiler to turbine, it is usual to provide a crossover, indicated by the dashed line, to permit the operation of either turbine from the adjoining boiler or battery of boilers. In a plant having a multiplicity of turbines and boilers, these cross connections between adjoining units give the appearance of a continuous header system, but, because of the small size of the crossovers and

consequent high-pressure drop, steam is seldom transferred farther than to the adjacent unit.

The ring-header system shown in Fig. 4*b* is designed to permit the utmost flexibility in operation. The boiler leads and header are liberally sized so that steam may be supplied from any one of a number of boilers to a given turbine with approximately the same pressure drop as from the nearest boiler. Double valving is employed so that any one valve can be worked on without having to shut down more than one boiler or one turbine.

The new section of the main steam system shown in Fig. 5 shows a partial approach to the unit plan in that each pair of high-pressure boilers is direct connected to a turbine. Steam from any boiler may be by-passed, however, through a modest 10-in. ring

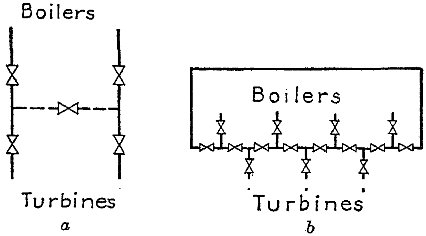


FIG. 4.—*a*, The unit boiler-turbine system; *b*, the ring-header system.

header that serves to transfer steam crosswise of the plant when occasion requires.

The old section of Fig. 5 illustrates the ultimate refinement of the ring-header system. Each boiler has available two distinct routes for reaching the take-off point to any turbine, both routes being of ample capacity to afford minimized pressure drops. This layout provides crosswise piping of a diameter larger than necessary if the direct leads from boiler to turbine can be utilized the greater part of the time, as is expected in the so-called "unit boiler-turbine" plan.

A typical steam header in a modern industrial plant is shown in Figs. 6 and 7. As may be noted, the boiler leads and connections supplying steam to process, heating, pumps, and the like are brought down to this header which is located near the floor so that all valves are readily accessible. The photograph in Fig. 7 shows an installation in the General Tire and Rubber Company's plant at Wabash, Ind.

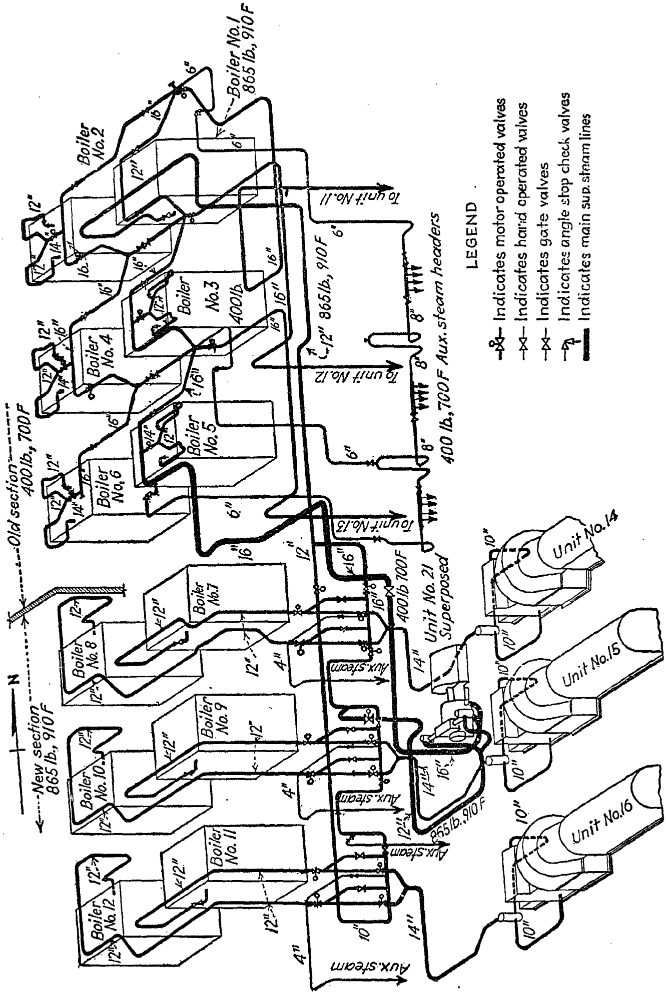
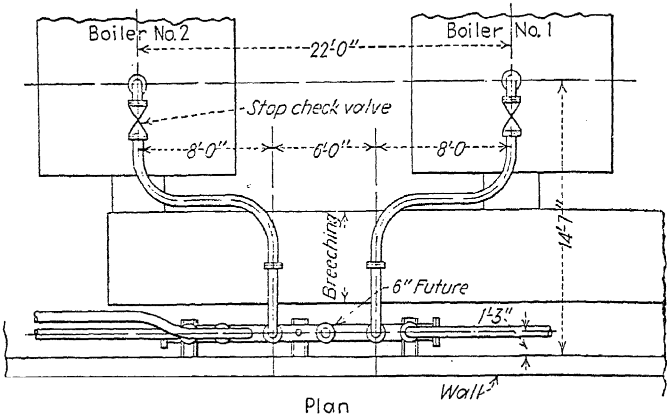


FIG. 5.—Main steam piping in old and new sections of Delray power plant.



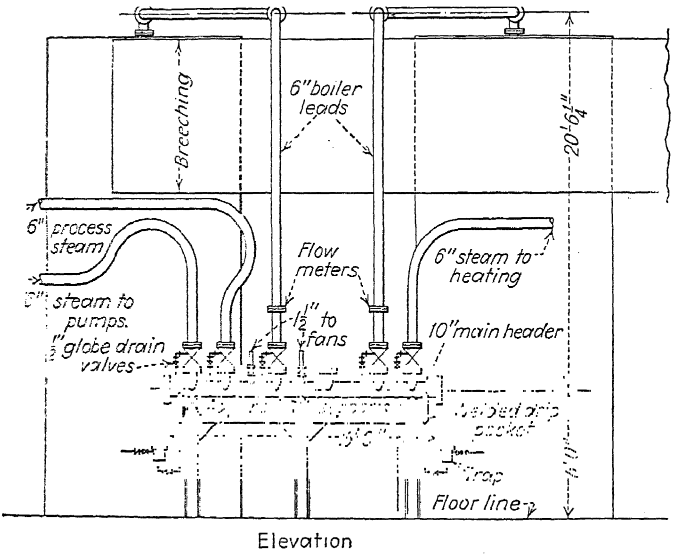


FIG. 6.—Typical plan and elevation of steam header in modern industrial plant. (*Courtesy of Valve World.*)

In the assembly drawing of Fig. 8, which is indicative of the recent trend of industrial power-plant layouts and piping arrangements along the lines of small central stations, the general use of welded joints has been an important factor in obtaining improved appearance. This plant was designed by Albert Kahn, Inc., for the Burroughs Adding Machine Company at Plymouth, Mich.

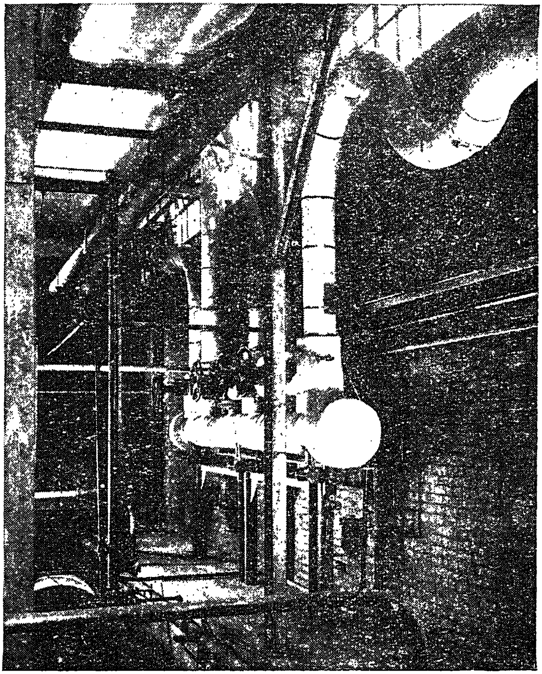


FIG. 7.—Steam header in industrial plant. (*Courtesy of Valve World.*)

The selection of pipe and valve materials for high-temperature service is described in detail in Chap. III. Suitable ASTM specifications for piping materials are abstracted in Chap. IV. Typical selections of insulating materials for power-plant piping are given in Table V, Chap. V.

Boiler connections for main superheated-steam piping are taken from the superheater outlet header through stop and check (combination) valves which are provided to prevent back flow of

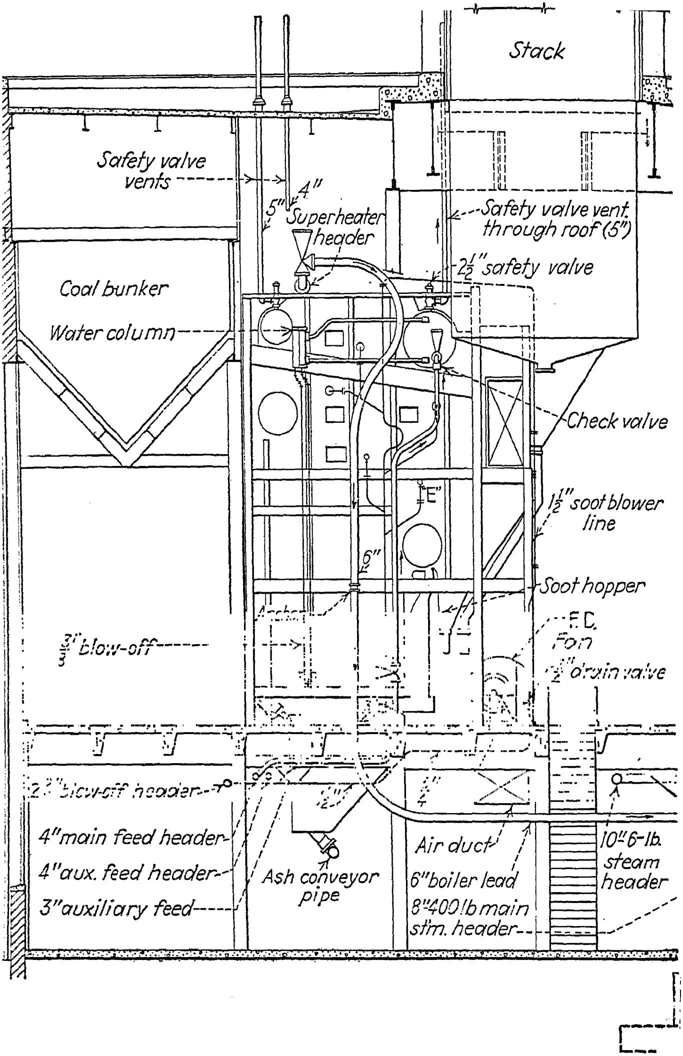
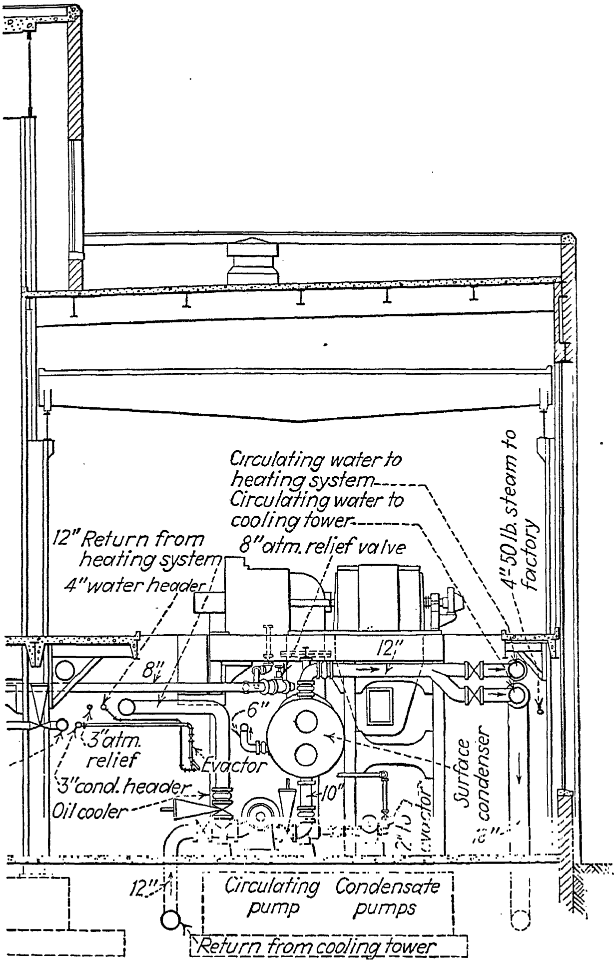


FIG. 8.—Assembly of steam piping in power plant of the



Burroughs Adding Machine Company at Plymouth, Mich.

steam from the mains into a boiler in case of tube rupture or other boiler trouble. A typical combination valve is shown in Fig. 9.

Auxiliary Superheated-steam Piping.—The auxiliary superheated-steam supply for operating auxiliary machinery is usually carried through a separate header system taking its supply from connections at fittings or manifolds in the main steam system

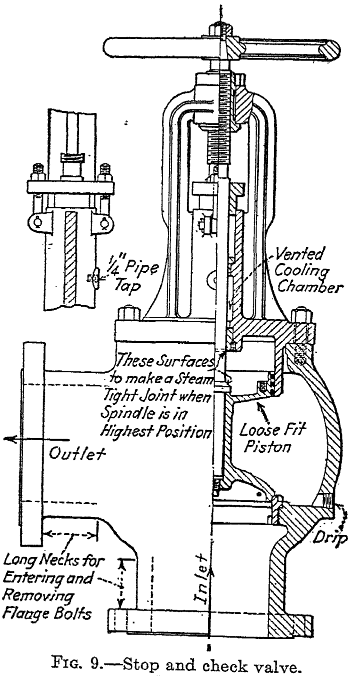


Fig. 9.—Stop and check valve.

as shown in Fig. 5. It is customary to install stop and check valves at the points of connection to the main system to prevent back feeding or short circuits through the auxiliary mains. The standards of construction for the auxiliary system should be on a par with the main superheated-steam system.

Auxiliary Saturated-steam Piping.—The auxiliary saturated-steam supply for soot blowing, building heating, and other miscel-

laneous plant uses is taken from the main saturated boiler drum through stop and check valves similar to those used for superheated steam on the superheated outlet. It is customary to provide such a connection on each boiler and to tie each line into a saturated-steam header for the plant. The saturated steam outlet on a boiler will vary from 2 to 4 in. nominal size, depending on the boiler capacity, and the saturated header will usually be about 3 to 6 in. nominal size. With boiler pressures of 300 lb gauge and higher, it is usually desirable to provide reducing valves or other throttling devices to reduce the saturated-header pressure to about 200 lb gauge, which is about right for soot blowing and can be further reduced as required for other plant uses. Auxiliary saturated steam is used for generator fire extinguishers, steam seals on turbine shaft packing, steam-jet air-removal equipment, thawing frozen coal in cars, thawing needle ice in intake canals, cracking scale in evaporators, and similar uses, including reserve supply for building heating. The normal supply for building heating is commonly bled from house-service turbines where these are available. The construction standards for the high-pressure auxiliary saturated system should ordinarily be equal to that for the main superheated-steam system, except where the difference in temperature may permit a lower pressure standard (see Chap. IV). It is permissible, however, to use a lower pressure standard where the line pressure is reduced, provided that part of the system is adequately protected with safety valves.

Auxiliary Exhaust Piping.—Exhaust piping from steam-driven pumps, blowers, etc., should be of a construction standard suitable for the conditions obtaining. Standard-weight pipe and American standard cast-iron flanges and fittings for 125 lb SSP are usually adequate for the purpose. In some cases where the steam-driven auxiliaries are intended for emergency use only, their turbines may be of a thermally inefficient type which allows considerable superheat to carry through in the exhaust. Under such circumstances it may be desirable to use American Standard steel flanges and fittings for 150 lb SSP. Where auxiliary exhaust lines are under vacuum, it is often advisable to use lapped pipe ends and loose flanges or welded joints. The exhaust from continuously operated steam-driven auxiliaries is usually led to feed-water heaters. In the case of emergency steam-driven auxiliaries, exhaust may be to the main condenser or to atmosphere, or to both.

Atmospheric Exhaust Piping.—Condensers are usually provided with atmospheric exhaust valves and piping leading to atmosphere to protect the condenser against excess pressure arising from failure of the circulating water supply. Atmospheric relief valves are sometimes fitted with hydraulic cylinders, as shown in Fig. 10, to open the valve before the absolute pressure in the condenser reaches atmospheric pressure. The valve is arranged to reseat at some predetermined pressure governed by design conditions. Atmospheric exhaust valves should have water-sealed seats to

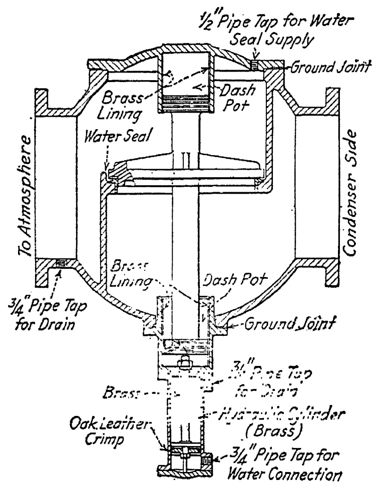


FIG. 10.—Atmospheric relief valve for condenser.

prevent air leakage into the condenser. Atmospheric exhaust piping may be thin-walled pipe or duct work of sufficient strength to withstand the small back pressure caused by friction loss in flow to atmosphere and to resist corrosion. This back pressure usually does not exceed 5-lb gauge. Where ducts are used, provision should be made to get inside for painting as a protection against corrosion. Provision of some kind is necessary to relieve expansion strains in the atmospheric exhaust line when the machine goes to atmosphere.

Safety-valve Vent Piping.—Except in the case of small low-pressure boilers, it is advisable to carry the vents from safety

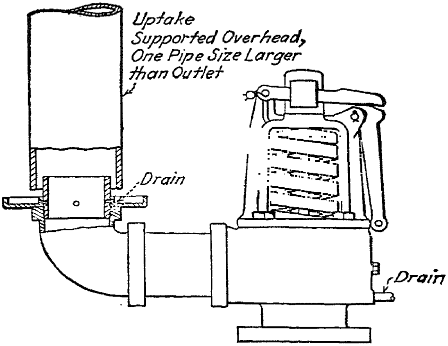


FIG. 11.—Safety-valve vent with umbrella fitting.

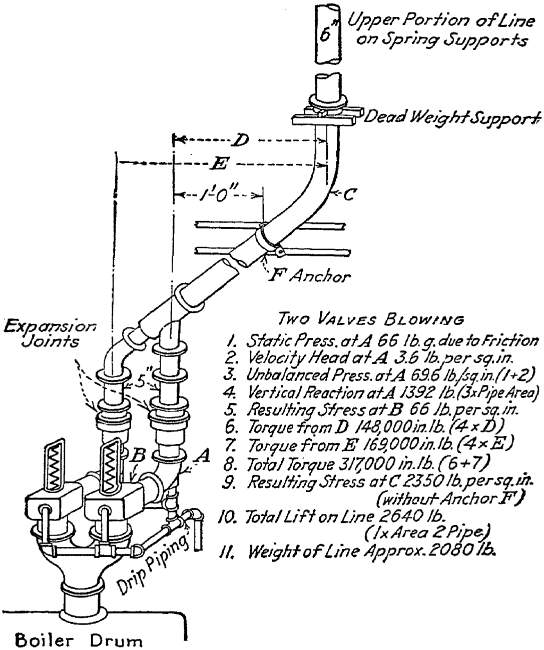


FIG. 12.—Reactions from blowing of safety valves.

valves to the outside of the building and extend them at least 6 ft above the roof to avoid scalding anyone happening to be near at the time the valve blows. Where separate vents are used for each valve and the run is short, the vent pipe may be rigidly attached to its safety valve and carried through the roof in a pipe thimble screened with suitable flashing. Where the vents from two valves are combined into a single pipe, or where there is a long run to atmosphere, some form of flexible connection should be employed to relieve expansion strains. The umbrella type of flexible connection shown in Fig. 11 is extensively used, since it is both simple and inexpensive. Slip joints are sometimes used, as shown in Fig. 12. While the slip joints require a certain amount of maintenance, they have the advantage of providing a tighter seal against steam blowing out into the room than the umbrella-type connections. Safety-valve vents should be dripped at the lowest point in the line or in the valve body just above the seat to prevent accumulations of water in the escape pipe.

Where the vent pipe is long enough to offer considerable frictional resistance to the escaping steam, consideration should be given to the reacting forces set up. A typical problem of this kind occurring with a water-screen drum is illustrated in Fig. 12. Stress at point C is relieved in this case by providing an anchor at F to take the thrust occurring when the valve discharges.

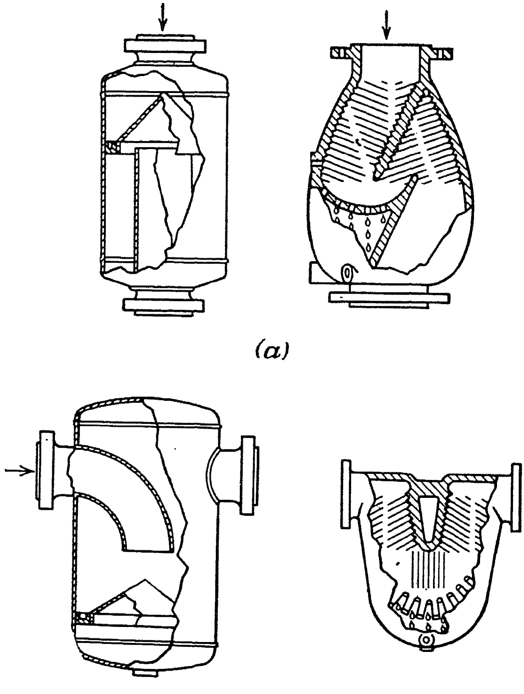
Drips and Drains.—Steam and air lines must be properly pitched and dripped between valves and at pockets in the line to get rid of accumulations of water. Although there is no considerable amount of water to drip from a superheated steam line while it is in operation with a steady flow of steam through it, yet it is just as necessary that provision be made here as in the case of a saturated-steam line. Superheated lines with a valve closed at one end and no flow must be dripped and, during the period of warming up any line after a shutdown, large amounts of condensation have to be removed. It has been the authors' observation that it is impossible to maintain superheat at the remote end of a long line under full pressure unless there is a considerable steam flow through the line. In cases where an attempt is made to keep such a line warm by allowing steam to escape from a small bleeder valve at the remote end, the temperature of steam in the line will soon fall to the saturation point. The absence of superheat is, of course, obvious during warming up a line from a cold or semicold condition.

The amount of condensate removed as drip from a steam line varies, of course, with the quality or superheat of the steam, the effectiveness of insulation, the length of line dripped, whether the line is initially hot or cold at the time of observation, etc. With such conditions known, it is possible to compute the heat loss or the warming-up loss and estimate the amount of drip for the given steam conditions. In the case of an extensive piping system, such a computation would involve considerable work. According to data published by the National Electric Light Association, the condensation collected as drip from the entire superheated and saturated steam-piping systems of a central station should not exceed 0.25 per cent of the entire water fed the boilers, and frequently is much less. On the other extreme is a central-heating boiler plant and distributing system, where the entire line condensation amounts to 5 to 15 per cent of the total water fed the boilers. Condensation in the central-heating plant itself may not exceed 0.5 per cent of the boiler feed. Small- and medium-sized industrial plants usually will have a line loss not exceeding 1 per cent of the total boiler feed.

Where a drip pocket is installed in a saturated-steam line, it is good practice to make the opening into the pocket the full size of the line, as shown in Fig. 16 on page 896, as a precaution against condensation being carried by. In reciprocating-engine practice, especially where saturated steam is used, it is advisable to have a separator located close to the throttle. It is frequently convenient to make such separators sufficiently large to serve the dual purpose of separator and steam receiver. Vertical and horizontal welded-steel separator receivers of this kind are illustrated at the left in Figs. 13*a* and *b*, respectively. Provision in separators for removing entrained moisture is frequently much more elaborate than shown in Fig. 13, and such separators are extensively illustrated in piping-supply catalogues.

Where a partial dam is created in a line by the presence of a globe valve or similar obstruction to the flow of line condensation, a drip connection and trap should be provided. Any pockets in the line should be dripped and care taken to see that gate valves are not installed with their stems below the horizontal because the bonnets will act as pockets if the stems are turned down. Disregard of these precautions is apt to result in severe water hammer or damage to equipment through slugs of water being carried on through. For the same reasons it is always advisable to warm

up a line slowly from the cold condition by merely cracking the main valve, or by opening a small by-pass around it. The main valve should be opened fully, and then in a cautious manner, only after the line is well warmed.



(a)

(b)

FIG. 13.—Steam receivers and moisture separators: (a) vertical types; (b) horizontal types.

Steam traps for removing collected drips from separators, drip pockets, tapped fittings, etc., are of numerous types such as bucket, float, tilting, thermostatic, and the like. Some commonly used varieties of bucket, float, and impulse traps are shown in Fig. 35 on page 564, and thermostatic traps in Figs. 10 and 13 on pages 954 and 958.

Traps should be located if possible so that condensation will flow into them by gravity, thus ensuring positive drainage. Where

a pipe line is in a trench below floor level, it is sometimes impossible to locate the trap below the pipe without placing it in a more or less inaccessible pit. Under such circumstances it may be permissible to place the trap above the floor, provided line pressure is sufficient to lift condensation to that elevation. This is never so satisfactory, however, as having gravity drainage to the trap. Where it is necessary to locate a trap above the line it drips, operation can be made more positive by providing lift fittings, such as those shown in Fig. 14. The action of lift fittings is to discharge alternate

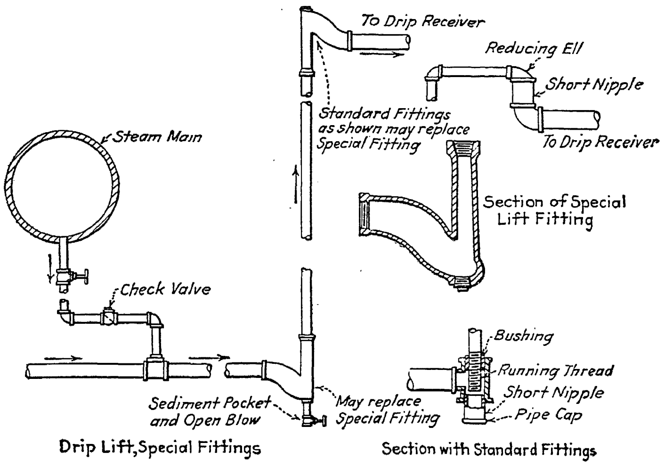
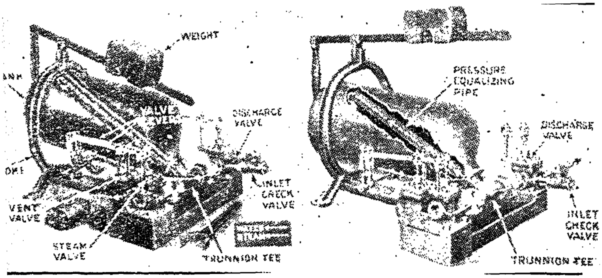


FIG. 14.—Drip lifts.

slugs of water and steam, thus lightening the weight of the discharge column and permitting higher lift.

The elevation to which a direct-acting trap will discharge may be anything up to that which corresponds to the minimum line pressure, proper allowance being made for frictional losses through the drip piping. Where the discharge elevation or back pressure exceeds the line pressure, so-called "lifting" traps are sometimes resorted to. Lifting traps may be either the float or tilting variety, having a three-way valve arrangement and an additional pressure connection from some source (steam or compressed air) which will furnish sufficient pressure to discharge the contents of the trap intermittently while connection from the line being dripped is

temporarily cut off. A lifting trap of the tilting variety is shown in Figs. 15*a* and *b*. A common application for lifting traps is the removal of condensation from dry vacuum lines and exhaust lines under vacuum. A special application of lift traps in connection



(*a*) (*b*)

FIG. 15.—Three-valve lifting trap, tilting type: (*a*) filling position; (*b*) emptying position.

with direct boiler returns systems is described on page 901. An explanation of the operation of a three-valve lifting trap of the tilting variety is given there.

The valve seats of traps are easily cut by dirt or pipe scale lodging in them, and strainers, or sediment pockets, such as that shown

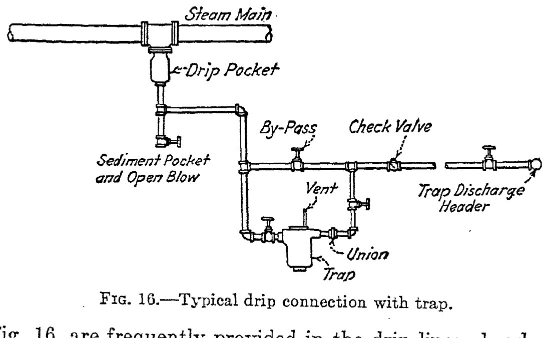


FIG. 16.—Typical drip connection with trap.

in Fig. 16, are frequently provided in the drip lines ahead of the trap to intercept such material. Where a trap is installed so that the steam line does not drain by gravity through the trap when the steam pressure is off, an open-drain connection and valve should be provided to let condensation run out on the floor or through a

funnel into the sewer. Often there are places in pipe lines which should be provided with open blows rather than traps to look after temporary conditions in warming up the line or to drain a dead end after the closure of its supply valve. Aside from the first cost of traps, they are a continual source of trouble and expense to maintain, and their use should be avoided wherever practicable through the use of open blows or other devices.

As a substitute for steam traps, many engineers prefer manually operated condensation receivers equivalent to that shown in

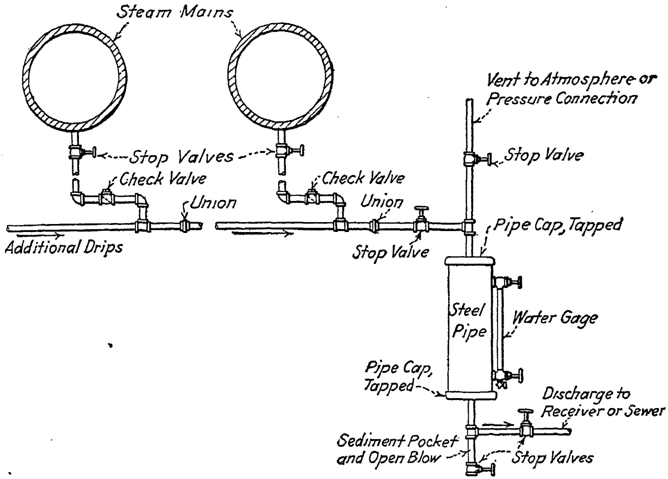


Fig. 17.—Manually operated condensation receiver.

Fig. 17. With this arrangement, drips are allowed to collect in the receiver until they reach some predetermined level which can be observed in the gauge glass. Periodic inspection reveals when this level is reached and steps are taken then to empty the receiver. Such receivers are applicable to either pressure or vacuum lines, the chief difference in the installation under the two circumstances being in the method of emptying. Where it is used to drip lines having sufficient pressure, the receiver can be emptied by merely opening the discharge valve. If, under low pressure or vacuum, the valve in the drip line entering the receiver must be closed first, the discharge valve opened next, and then either (1) an atmospheric vent may be opened, if the elevation of the receiver is enough to

cause the contents to flow by gravity to the point of disposal; or (2) a valve may be opened in a line that applies steam or compressed-air pressure above the surface of the liquid in the receiver, thus forcing it to flow to the point desired.

In instances where there is small difference between line pressure and that existing where the drips are to be discharged, it may be advantageous to let the discharge take place through a water leg or "siphon trap," as it is sometimes called, which is illustrated in a conventional way in Fig. 18. A siphon trap of this kind renders unnecessary the use of a mechanical trap and requires

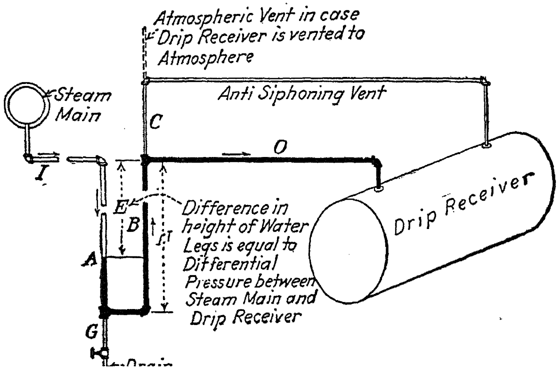


Fig. 18.—Water leg or siphon trap.

no maintenance. It consists of two legs *A* and *B*, which may be close together or any distance apart but the lengths of which must be sufficiently great to prevent pressure acting through pipe *I* from forcing the water level down to the bottom of leg *A*. *C* is a vent pipe connecting with the vessel into which *O* discharges. In case *O* discharges against atmospheric pressure, *C* may be vented directly to atmosphere. The purpose of vent *C* is to prevent siphoning out the water leg in case *I*, *A*, *B*, and *O* should run completely full of condensation for a time and then have the supply suddenly diminish. In ordinary operation the leg *B* is filled with water which is constantly overflowing, and *A* with steam and water, the total pressure in both legs being equal. The siphon trap is applicable only to small pressure differences, as it requires approximately 2.3 ft of vertical space *E* for each pound per square inch pressure differential.

In steam lines a liberal pitch should be given toward drainage points. In all horizontal pipes where there is considerable velocity, pitch should be in the direction of steam flow. Low-pressure, low-velocity, building-heating-system pipes, however, are frequently pitched against steam flow, in which case it is customary to use the next size larger pipe than would otherwise be chosen. With long lines subject to considerable expansion both horizontally and vertically, the position of the line in both the hot and cold positions should be determined, and the pitch made sufficient to look after both. The amount of sag in the pipe between supports also enters into the question of pitch, as explained on pages 744–749, where a formula and chart (Figs. 11 and 12) are given for obtaining sufficient pitch to offset such deflection. Important lines should be leveled with surveyor's instruments to ensure that the calculated pitch is actually obtained. A minimum pitch of 1 in. in 20 ft to 1 in. in 10 ft is usually sufficient for adequate drainage.

Where two or more traps discharge into a common header it is necessary to provide a check valve in the discharge of each trap before it joins the header. It is also advisable to install a stop valve in each discharge line between the check and the header to facilitate repairs to any trap or check valve. Discharges from high- and low-pressure traps should not be connected to the same header, unless this is done at or close to a low-pressure drip receiver.

Provision should be made for the gravity drainage of water from pipes and equipment or tanks containing water during periods of shutdown for repairs or where necessary to prevent freezing when not in use. In case water so removed is plant condensate or boiler feed, it may be desirable to provide a drain line leading to storage, otherwise discharge should be made to the sewer or equivalent. The size of drain required will depend on what is considered a reasonable time to empty the piping or equipment. The time required can be calculated by the method described on page 290. In the case of a large, elevated, water-storage tank for fire protection, a reasonable time might be several hours, while in the case of a boiler-feed line, boiler or condensate storage tank, etc., 1 to 2 hr at the most should be long enough. In making provision for such drainage, attention is called to the necessity for providing a vent to atmosphere above the water line.

Drip Disposal.—The most advantageous means of returning hot drips to the boiler or feed-water circuit will depend on individual plant arrangements. Discharge to the hot tanks of plants

heating their feed water with auxiliary exhaust steam and having hot-water storage is frequently good practice. Where this is done, the high-pressure traps or condensation receivers can be arranged to discharge directly into a header leading to the hot tanks. The low-pressure drips also can be discharged to the hot tanks either

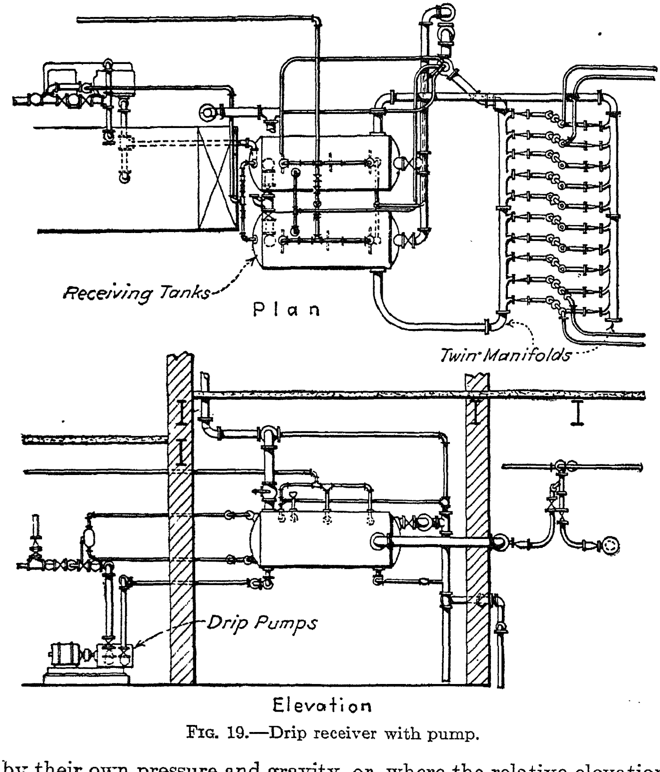


FIG. 19.—Drip receiver with pump.

by their own pressure and gravity, or, where the relative elevation of hot tank and traps require it, by the use of lifting traps or pumps.

Drip Receiver with Pump.—The present tendency is toward plants operating on a regenerative feed-heating cycle which, as a general rule, is installed without hot storage tanks. In this case a good way of returning all the plant drips to the boiler-feed system is to provide a large receiver tank, vented to atmosphere

through a back-pressure valve, into which all the drips discharge. The temperature in this tank can be kept down, if necessary, by spraying in cool water from a hot-well pump discharge line or equivalent supply. The accumulation of water in the receiver tank is removed by a float-controlled pump discharging into the boiler feed-pump suction, or, if deaeration is desired, into the shell of one of the series of bleeder heaters. Selection of the proper heater is made on a temperature pressure basis with a view to having enough temperature differential between the drips and

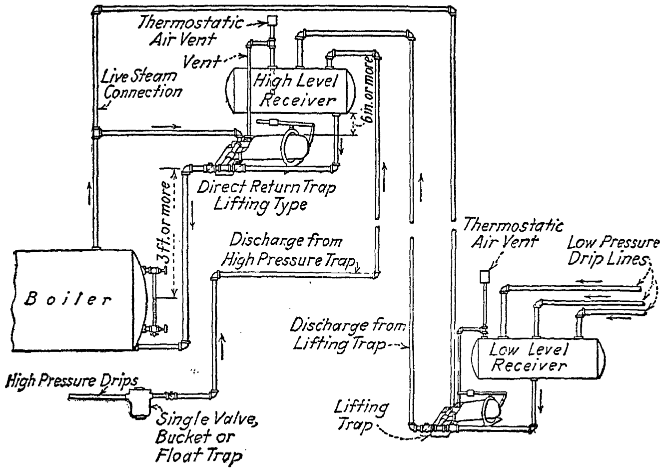


FIG. 20.—Direct return system for plant drips.

heater shell conditions to insure considerable flashing. A temperature differential of 40 to 50 F is usually adequate for the purpose. The importance of the service usually warrants installing such receiver tanks and pumps in duplicate. An installation of this kind is illustrated in Fig. 19.

Direct Return Traps.—A system for the direct return of drips to a boiler without the use of pumps is shown diagrammatically in Fig. 20. This operation of this system depends on the use of three valve-lifting traps similar in principle to that illustrated in Figs. 15*a* and *b* on page 896, and it may be applied to the return of either high- or low-pressure drips as indicated. Returns from both low-pressure and high-pressure traps are discharged to a

high-level receiver which feeds by gravity a direct return trap located 3 ft or more above the boiler line. The direct return trap, which is of the three-valve-lifting type, alternately fills by gravity from the high-level receiver and empties, through the application

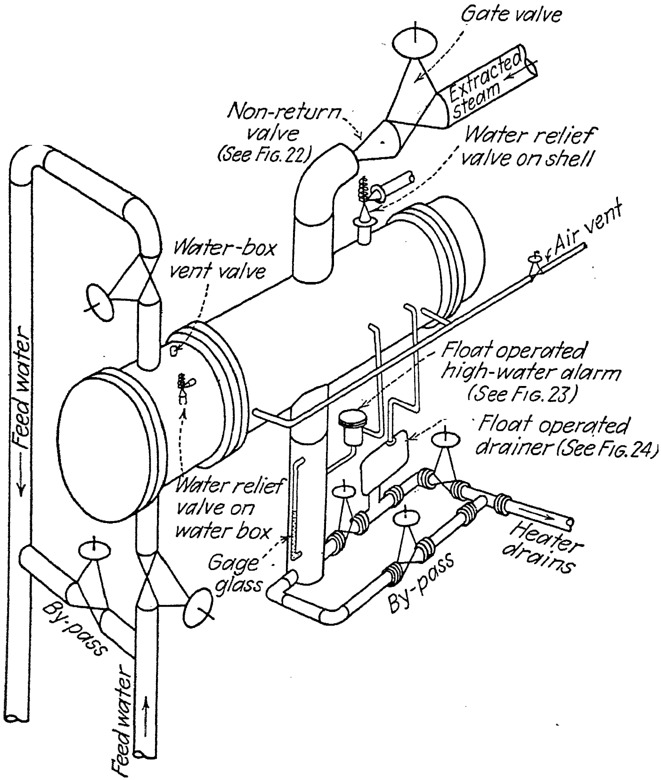


Fig. 21.—Typical bleeder-heater piping.

of boiler steam pressure, into the boiler drum. Check valves must be installed in the lines as indicated, and automatic air vents provided to release accumulations of air from the receivers. All stop valves have been omitted from the diagram to keep it as simple as possible. The direct return trap system is applied ordinarily in small plants only.

The operation of a *three-valve-lifting trap* such as that shown in Figs. 15*a* and *b* is explained as follows. As its name implies, it has three valves, *viz.*, inlet, vent, and discharge valves, which are operated by the tilting action of the trap. A check valve in the inlet pipe ahead of the trap is also essential to its operation. Water enters from the inlet pipe when the trap is in the filling position with vent open to atmosphere, as shown in Fig. 15*a*. The weight of condensation collecting in the trap causes it to overbalance the

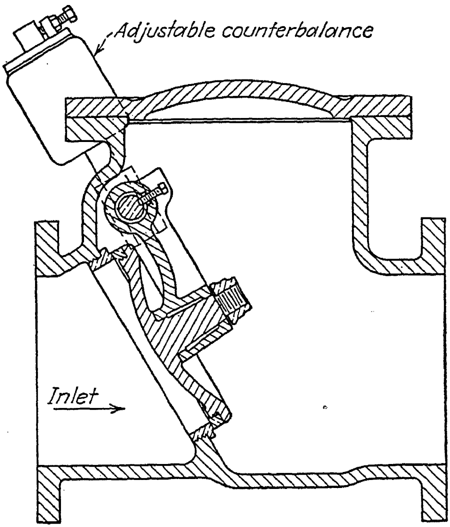


Fig. 22.—Counterbalanced nonreturn valve used in bleeder-heater piping shown in Fig. 21.

counterweight and assume the position shown in Fig. 15*b*. This closes the vent valve and opens the discharge valve and the high-pressure admission valve. Through the action of high-pressure air or steam above the water surface in the chamber, condensation is forced out the discharge valve and through the discharge pipe to a higher elevation than would be reached by normal line pressure.

Simple Steam Loops and the Holly Loop.—The use of steam loops to return water to boilers in small plants without the use of a trap, pump or injector is described in "Steam Power Plant Engineering," by G. F. Gebhardt, 6th ed., page 702. The Holly loop

which is an application of the same principles to larger plants also is described by Gebhardt.

Bleeder Heater Piping.—A general idea of bleeder heater piping can be obtained from Fig. 1 on page 863 where the various lines required are shown diagrammatically. A more detailed sketch of typical bleeder-heater piping is shown in Fig. 21, with sectional views of the special valves and fittings in Figs. 22 to 24. Figure 22 is a free-flow reverse-current valve with a counterweight to

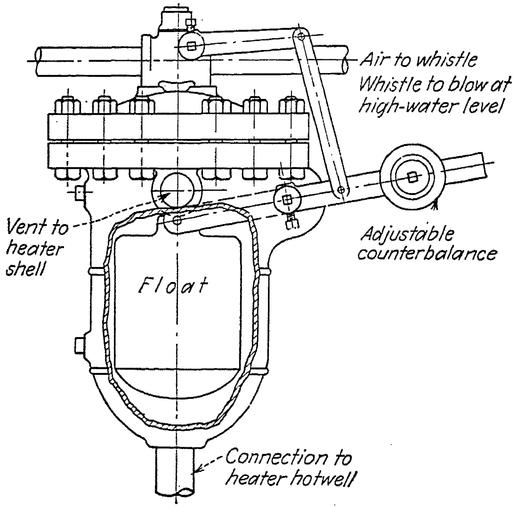


Fig. 23.—High-water alarm used in connection with bleeder heater shown in Fig. 21.

effect closure. Figure 23 is a high-water alarm in case the condensate drainer fails to function, or in case of tube rupture flooding the heater. Figure 24 is a float-operated drainer for controlling the flow of condensation from the heater shell. Water relief valves on the heater shells are provided to guard against excessive hydrostatic pressure on the shell resulting from tube rupture.

The arrangement described is only one of several systems which have been applied to safeguard the operation of bleeder heaters. A water leg or siphon trap, equivalent to that shown in Fig. 18 discharging to the main condenser, or the next lower pressure heater, can be provided to prevent flooding. Sometimes a

pilot valve with float control, shown in Fig. 25, is provided as a relay for operating the hydraulic actuating cylinder on an auto-

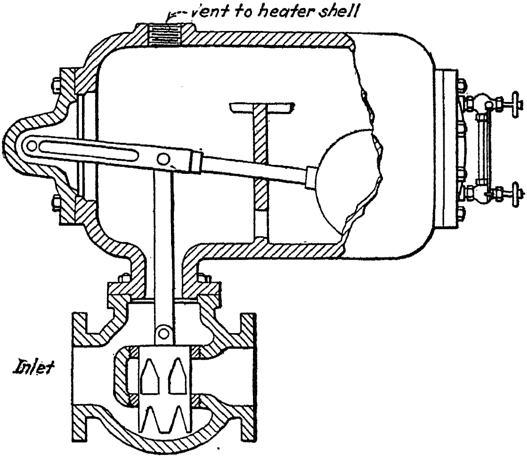


FIG. 24.—Condensation drainer for bleeder heaters.

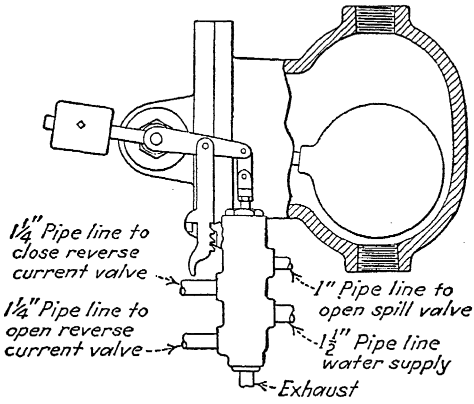


FIG. 25.—Pilot valve with float control.

matic spill valve to the main condenser as well as an actuating cylinder placed on the reverse current valve.

The principal lines required are: bleeder steam line from turbine, water circuit in and out of heater, condensation drain, air vent,

and relief valve. The choice of material for each line depends on the pressure and temperature of the fluid flowing through it. Some of the chief features to be considered are as follows:

The *bleeder steam line* from turbine casing should be of generous size and particular care taken to avoid restrictions or types of valves which cause appreciable pressure drop. This is especially important in bleeder lines since any drop in steam pressure is attended with a corresponding decrease in the saturated-steam temperature obtaining in the heater shell. This in turn results in a lowered outlet water temperature and less heat reclaimed from finding its way to the main condenser. An economic study of the thermal loss resulting from pressure drop in bleeder lines will reveal that extra investment to reduce frictional loss will pay big dividends. Some form of nonreturn or check valve is required in bleeder lines to serve a dual purpose: first, in case the turbine governor trips, it prevents back feeding of steam from the bleeder heater with the attendant possibility of overspeeding the turbine; and second, it prevents water entering the turbine in case the heater shell floods, due to tube rupture or other causes. Certain types of check valves are unsuitable for use in bleeder lines because their construction is such as to produce a material pressure drop. For this reason, a balanced swing check, such as that illustrated in Fig. 22, is preferable to the poppet type sometimes used.

Where a *heater drains* pump is not provided, some type of float-operated drainer is desirable to control the flow of condensation from the heater. A compact type which eliminates a stuffing box for the valve rod is shown in Fig. 24. Heater drains are frequently cascaded successively to the shells of the lower pressure heaters for the sake of the increased economy resulting from the delivery of flash steam where it can be absorbed to best advantage. Where the difference in shell pressure between two heaters is not too great, a water leg or siphon trap, similar in principle to that illustrated in Fig. 18, is sometimes used as a substitute for the mechanically controlled drainer. The flow through pipes of flashing mixtures of saturated water and steam, such as exist in heater drain lines, is discussed under "Blowoff Piping" on page 915.

Provision for *air vents* from bleeder heaters is necessary. In the case of surface heaters, such vents are usually taken off from an air box where the air is refrigerated and dried by contact with tubes situated in the coolest pass of the heater. Air vents should be

of ample size to evacuate air from the shell quickly in starting up the unit. It is then customary to restrict the passage so that

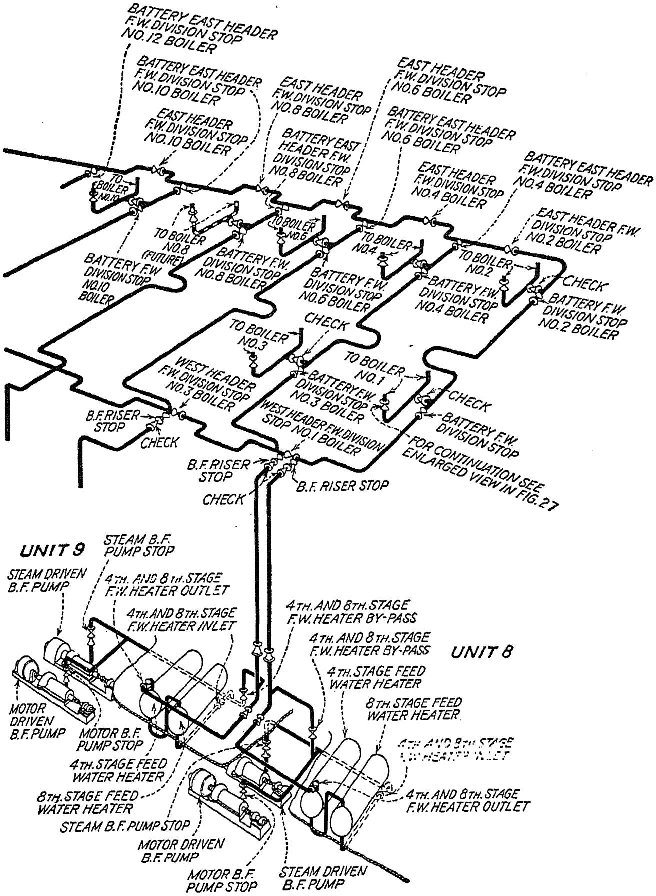


FIG. 26.—Boiler-feed system, Conners Creek Plant.

excessive amounts of steam will not be drawn off with the air. Heaters operating at pressures above atmospheric pressure may be provided with a thermostatic vent to atmosphere, and those

under vacuum vented to the main condenser. Better economy is obtained, however, by cascading the air vents successively to the shells of lower stage heaters.

Boiler-feed Piping.—Pipe, fittings, and valves in boiler-feed lines are usually of the same weight and pressure standard as used for the main steam piping. But, the materials may be quite different for steam temperatures above 750 F. Feed piping between the boiler drum and the first valve should be of the weight specified in the ASME Boiler Construction Code for the rated working pressure of that boiler. Check valves should be provided in the individual pump discharges and close to where feed lines enter a boiler.

Because of the vital necessity of getting water to boilers, dual feeding arrangements in various degrees of complete duplication are used. An ample number of boiler-feed pumps is the prime requirement. Where motor-driven pumps are used with stoker-fired boilers it is customary to provide enough steam-driven pumps in addition to look after any emergency condition arising from failure of the electricity supply. With small boilers, injectors (Fig. 30) are sometimes used instead of stand-by pumps.

In many plants there are complete dual systems of boiler-feed piping. The more recent trend, however, has been toward less complete duplication, principally because of the proven reliability of modern welded piping systems. Figure 26 shows the complete boiler-feed system at the Conners Creek Plant of The Detroit Edison Company. Figure 27 shows connections at the boiler.

The use of automatic-feed water regulators, such as those shown in Figs. 28 and 29, to admit water to the boiler as required to maintain a constant water level is common practice. Hand regulation is customary on the auxiliary feed service, and a by-pass should be provided around the automatic regulator on the main feed. Boiler-feed piping should be firmly secured with sway bracing to damp pulsations due to water hammer, even where centrifugal pumps are used. Proper sizes for boiler-feed piping can be selected by reference to the pressure-drop data given on pages 269 to 289.

Boiler-feed Injectors.[1]—Injectors are made in many forms, but Fig. 30 shows the typical arrangement and illustrates the method of operation. Steam is admitted through the valve *M*, by turning the handle *K*, and enters the expand-

[1] Reproduced from "Steam Boiler Engineering," by permission of the Heine Safety Boiler Company.

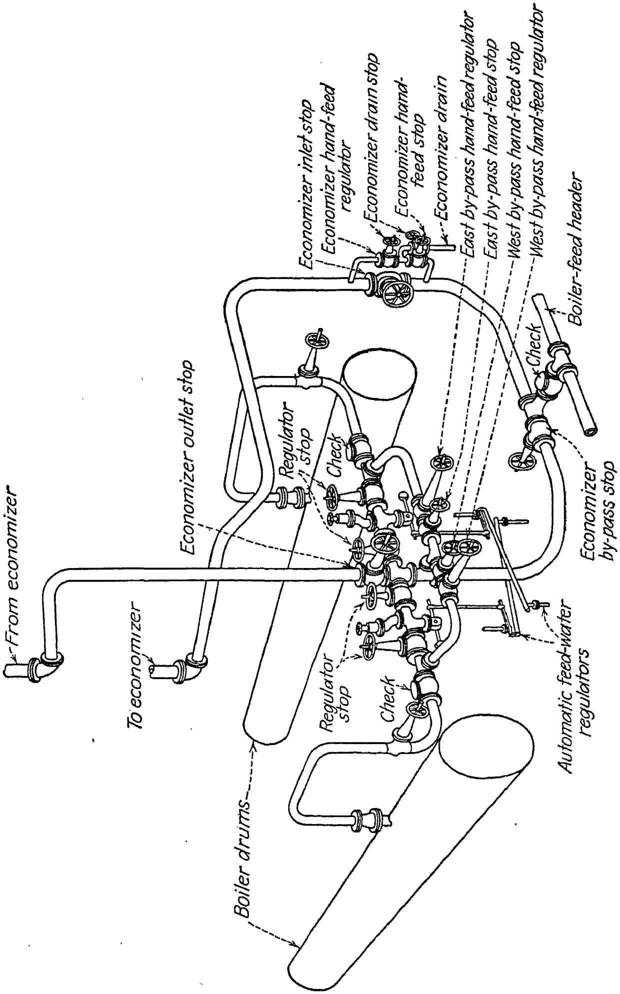


FIG. 27.—Boiler-feed connections at boiler drum, Conners Creek Plant.

ing nozzles where the pressure is reduced and the velocity greatly increased. The steam jet is then guided to the contracting nozzle or lifting tube V. In passing from the first to the second nozzle it carries along the air in the chamber and creates a vacuum. The water to be pumped rises in the suction pipe and fills the chamber. The steam and water thus enter the lifting tube, passing to

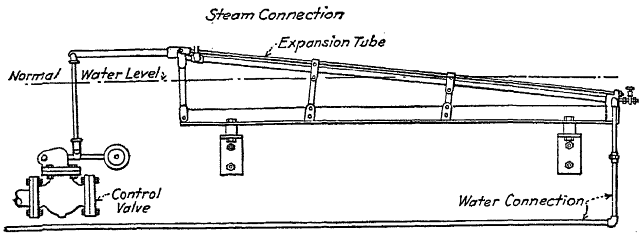


FIG. 28.—Feed water regulator of expansion type.

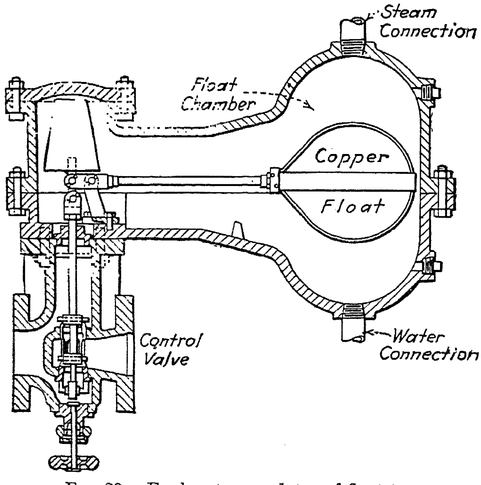


FIG. 29.—Feed water regulator of float type.

the mixing nozzle C, and the steam is condensed. When the water and steam have reached the delivery nozzle D, the steam has been condensed and the water is traveling at a high velocity imparted to it by the steam. The delivery nozzle is increased in cross-sectional area, reducing the velocity and hence increasing the pressure of the water. Consequently its head is sufficient to overcome the resistance of the feed valve, and the water enters the boiler. The steam has thus imparted kinetic energy to the water; this energy is converted from velocity to pressure in the delivery nozzle. The water is heated through the condensation of the steam.

The action of the injector depends not only upon the impact of the jet of steam, but also upon its efficient and complete condensation, which must occur during its passage through the combining tube. At 180-lb boiler pressure the water must attain a terminal velocity of 163 ft per second to balance the pressure, and something more to lift the check valve and enter the boiler. If the total length of the converging combining tube is 7½ in., the interval of time during which the steam can be condensed is only 0.008 of a second and the acceleration is 4 miles per second per second.

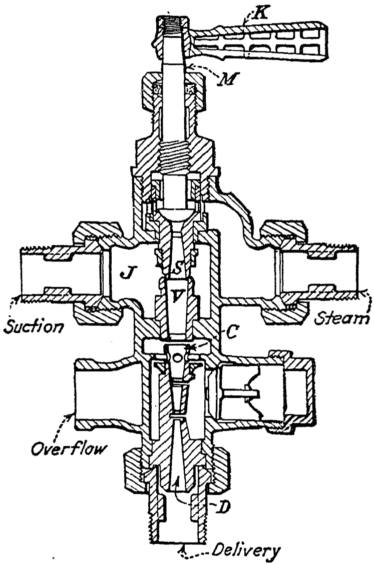


FIG. 30.—Boiler-feed injector.

Anything that tends to diminish rapid condensation operates against mechanical efficiency. An increase in the temperature of the water supply, moisture, or superheat in the steam all tend to reduce the proper ratio between the weight of the water delivered into the boiler and that of the motive steam. The steam must undergo instant and complete condensation, and its velocity must reach a maximum at the instant of impact with the water.

A comprehensive paper on "Characteristics of Injectors" by R. M. Osterman appears in the *Trans. ASME*, 1928. This paper deals with the theory and design of injectors.

Suction Lift of Boiler-feed Pumps.[1]—In the case of a boiler-feed pump or other pump handling hot water, consideration must

[1] Reproduced from "Steam Boiler Engineering," by permission of the Heine Safety Boiler Company.

be given to the decreased lifting ability of the pump as the water temperature is increased.

For cold-water service, that is, water at 60 to 70 F, feed pumps give satisfaction with a vertical suction lift as high as 15 ft. Generally, however, the suction lift of the feed pump is decreased by the temperature of the water.

The atmospheric pressure, which is equivalent to a head of 34 ft of water, forces the water into the pump. In practice, deductions must be made for the loss of head at the pipe entrance, pipe friction, valve friction, acceleration of

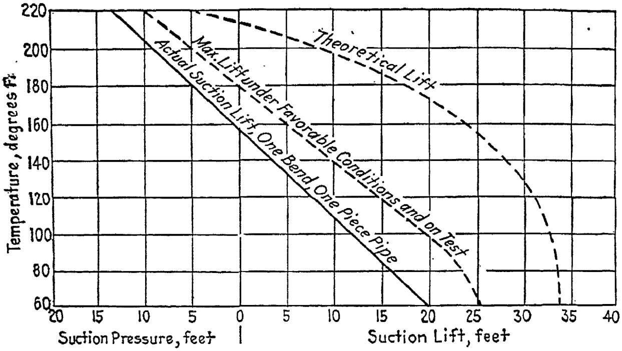


FIG. 31.—Suction lift or suction head at different temperatures.

water to its highest velocity, and pressure necessary to prevent vaporization of hot water. For example:

	Feet
Entrance loss, say	2.0
Suction pipe friction	2.5
Acceleration, or velocity head	2.0
Pressure to prevent vaporization at 120 F	3.9
Assumed lift	15.0
	25.4
Available head for lifting suction valves and as a factor of safety for contingencies	8.6
Total	34.0

The velocity head of 2 ft is a typical figure for a centrifugal pump, in which the water velocity through the eye of the impeller will be about 12 ft per second.

Figure 31 shows curves of suction lift or suction head for different water temperatures. The right-hand curve represents theoretical conditions as in the steam tables, or the pressure to prevent vaporization of the water. The curve in the middle represents the maximum suction lift or maximum suction head. For ordinary piping, the left-hand curve should be used.

If the capacity is too high for a pump or suction pipe handling hot water, the velocity head will be increased and the water handled will be vaporized. If the suction pressure is too low, or the lift is too high, the hot water will be vaporized. Vaporization causes knocking in the discharge lines and greatly

reduces the capacity and efficiency of a direct-acting pump. The capacity will also be decreased with centrifugal pumps, since the water passages will be filled partly with vapor and partly with water.

Condensate Piping.—The suction and discharge piping of hot-well pumps is generally referred to as condensate piping. Condensate piping usually consists of standard-weight black pipe and American Standard 125-lb cast-iron flanges and flanged fittings. Copper pipe is sometimes used for flexibility. Typical piping arrangements for a hot-well pump are shown in Fig. 32. The vent or equalizing pipe is necessary to permit water to flow

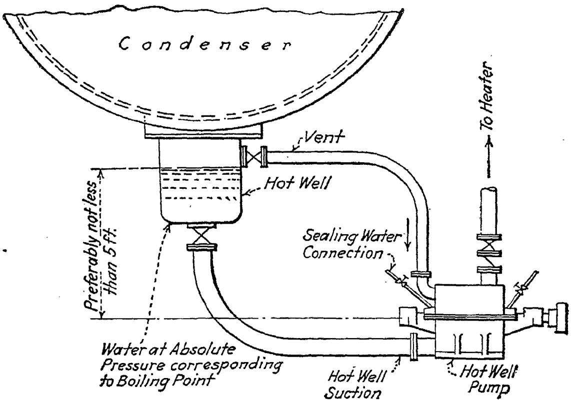


FIG. 32.—Hot-well pump piping.

from the condenser (which is under high vacuum) into the pump casing where it can be picked up by the impeller.

Blowoff Piping.—Boiler blowoff connections vary in size from 1 to 2½ in. depending on the size of the boiler. Where several blowoff connections join into a common header, the header is often 4 in. or larger. Evaporator blowoff connections are frequently made as large as 4 in. to furnish easy egress for scale accumulations. The ASME Boiler Construction Code requires that where blowoff lines from more than one boiler are connected to a common header, a guard valve be provided to prevent workmen being scalded in any boiler which is down for repairs. Where there are several blowoff connections on one boiler, each should have its own blowoff valve, and a single guard valve may be provided where the combined lines from that boiler join a common header serving other boilers. As an alternate arrangement,

some engineers prefer to provide a blowoff valve and a guard valve in tandem on each blowoff connection. A typical blowoff valve and guard valve are illustrated in Fig. 33. Among the principal design features in blowoff valves are quick opening and free passage for sediment or scale. Guard valves for blowoff lines are frequently of the conventional gate-valve type. In order to protect the

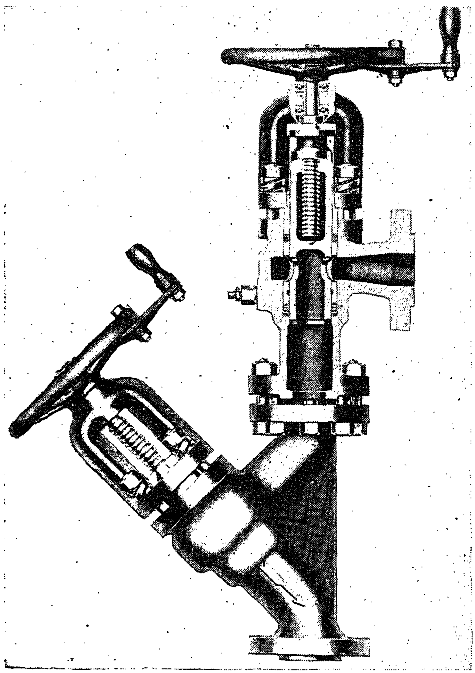


FIG. 33.—Blowoff valve and guard valve.

seating surfaces of the guard valve and ensure its continued tightness, it is customary to open the guard valve wide before cracking the blowoff valve proper.

Boiler blowoff piping between guard valve and blowoff tank or circulating water overflow canal where pressure is reduced to approximately that of the atmosphere is required by the Code for Pressure Piping to be designed as for the following steam pressures.

Boiler Pressures, Psi Gauge	Design as for Steam, Psi Gauge
901 to 1,500	600
601 to 900	400
250 to 600	250
Below 250	250[1]

[1] Except that for boiler pressures in excess of 150 lb, steel fittings of ASA 150-lb standard shall be used.

Flashing mixtures of water and steam occur in boiler blowoff lines, in evaporator and feed-water-heater drain lines, and in other piping where hot water flows at a pressure less than the saturated pressure corresponding to the temperature of the water entering the line. The actual increase in specific volume depends on the end-pressure condition which, in turn, depends on the pipe size and length of travel and is independent of the pressure in the receiving vessel (or atmosphere) so long as the latter is below the calculated end pressure.[1] Owing to the characteristic of saturated water to flash as the pressure falls, the pipe tends to choke with vapor so that the back pressure in the line remains a substantial part of the initial pressure and is not reduced almost to atmospheric or receiver pressure as might wrongly be assumed.

This fact should be kept in mind in choosing piping weights for boiler blowoff lines. The foregoing pressures for blowoff piping quoted from the Code for Pressure Piping have been found through experience to provide sufficient margin for usual combinations of saturated pressure and length of line. Those wishing to make a specific check of conditions can refer to the basic methods given in the Benjamin and Miller paper, or to "Charts for Estimating Capacity and Outlet Pressure of Boiler Blow-off Lines" derived from their methods.[2]

In the case of heater and evaporator drain lines it often is desirable to put the flow-controlling device, whether drainer, trap, or orifice,[3] near the outlet end of the line so as to avoid flashing by maintaining saturation pressure as far as possible. Where

[1] See "The Flow of a Flashing Mixture of Water and Steam through Pipes," by M. W. Benjamin and J. G. Miller, *Trans. ASME*, Vol. 64, No. 7, pp. 657–669, October, 1942.

[2] See "Tube Turns Catalog and Engineering Data Book," No. 111, published by Tube Turns, Inc., Louisville, Ky.

[3] For designing orifices see "The Flow of Saturated Water through Throttling Orifices," by M. W. Benjamin and J. G. Miller, *Trans. ASME*, Vol. 63, No. 5, pp. 419–429, July, 1941.

severe flashing with attendant high velocity exists in lines carrying a mixture of steam and water, there is apt to be excessive erosion, especially at elbows and bends, which may eat through the walls in the course of a couple of years.[1]

Blowoff piping from the boiler drum to the guard valve should be of the weight prescribed in the ASME Boiler Construction Code for the rated working pressure of that boiler. Adequate protection should be provided for blowoff piping where exposed to hot gases within the boiler setting.

Blowdown Tanks.—Where a canal is not available to receive the blowdown from power boilers, it is frequently necessary to provide a blowdown tank, such as that illustrated in Fig. 34. Local ordinances generally require that such tanks be provided to receive and cool the blowdown before it is discharged into a public sewer. Blowdown tanks are usually low-pressure chambers vented to the atmosphere, where blowdown can be collected and cooled by mixture with city or general service water, and then allowed to flow gently into a sewer under gravity head. In Fig. 34 blowdown enters the tank tangentially through a water mixer into which city (or general service) water is introduced under a moderate pressure of about 40 lb gauge. If blowdown is discharged directly into a tank of cold water without provision for proper mixing, severe water hammer will result. An alternate arrangement to that shown is to supply cold water direct to the tank and introduce blowdown through an internal fitting called a "noiseless water heater" or "hydro-kineter" similar in principle to the canal fitting shown in Fig. 36.

In order to avoid water hammer and secure satisfactory cooling of the blowdown, sufficient city water must be provided to lower the temperature of the mixture below 212 F. The theoretical amount required can be computed by the method outlined on page 137. The actual amount of city water required with the arrangement shown in Fig. 34 is usually two to three times the theoretical, since blowdown is not continuous and sufficient cold water must be flowing to look after the instantaneous flow of blowdown. As a result, the temperature of the tank overflow may be as low as 130 F rather than the permissible maximum of 212 F. Meters are sometimes installed on the city water supply

[1] See "The Flow of a Flashing Mixture of Water and Steam though Pipes," by M. W. Benjamin and J. G. Miller, *Trans. ASME*, Vol. 64, No. 7, pp. 657–669, October, 1942.

and overflow to keep a record of the amount of blowdown and the quantity of city water used. The blowdown is obtained by taking the difference of the two meter readings.

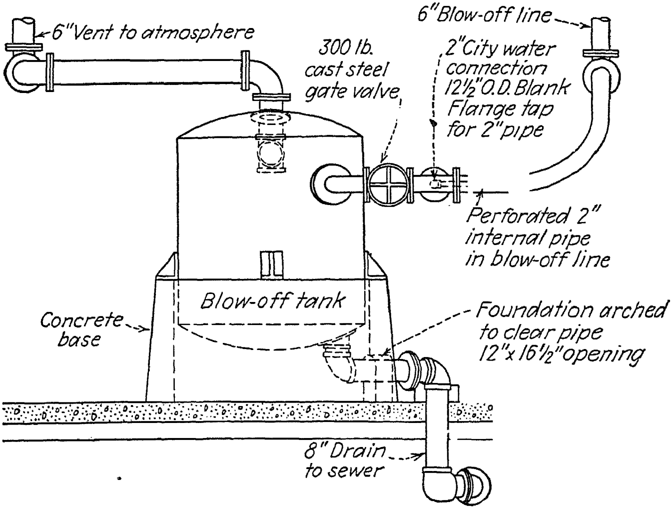


FIG. 34.—Blowdown tank.

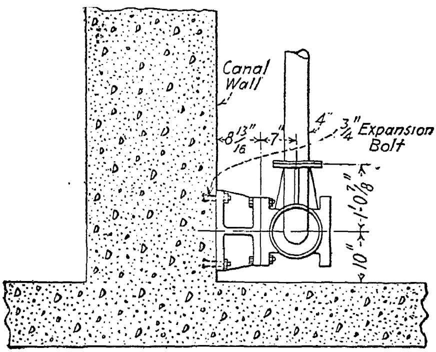


FIG. 35.—Installation of blowoff fitting in canal.

The tank itself serves the purpose of a vented chamber at atmospheric pressure, floating on the line between the blowoff line

and the sewer. Its size is of little importance provided the overflow pipe is adequate as to diameter and elevation above the sewer connection to handle the total amount of blowdown and cooling water. If a drain cannot be provided large enough to handle this rate of flow, the tank must be made of dimensions

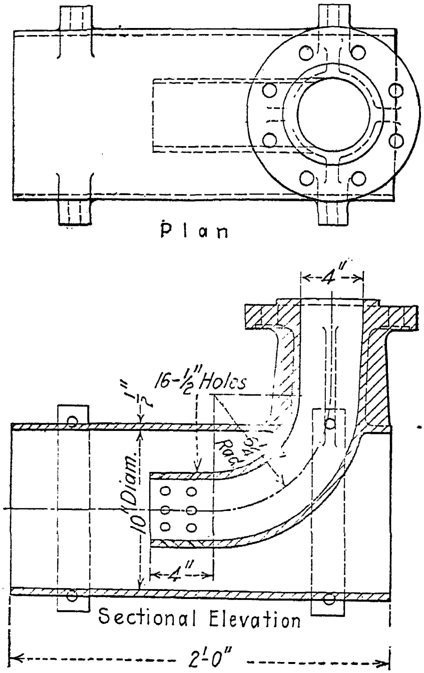


FIG. 36.—Detail of canal fitting for blowoff line.

ample to accommodate the blowdown and cooling water collected during the blowing down operation of one valve. The contents of the tank then can be allowed to drain off before blowing the next valve.

Canal Blowoff Fittings.—Where a canal is available to receive blowdown, it is customary to dispense with the blowdown tank and inject the blowdown directly into the canal with a mixing fitting, such as that illustrated in Figs. 35 and 36. This saves

considerable investment cost, is easier to operate, and eliminates any expense for furnishing cooling water to the tank.

Continuous Blowoff.—In boiler plants where there is a large amount of make-up water, it is often desirable to blow off a small amount of water continuously rather than a large quantity at intervals. The chief advantage of a continuous blowoff is that it permits the heat to be extracted from the blowoff water. A satisfactory arrangement of a continuous blowoff system is shown

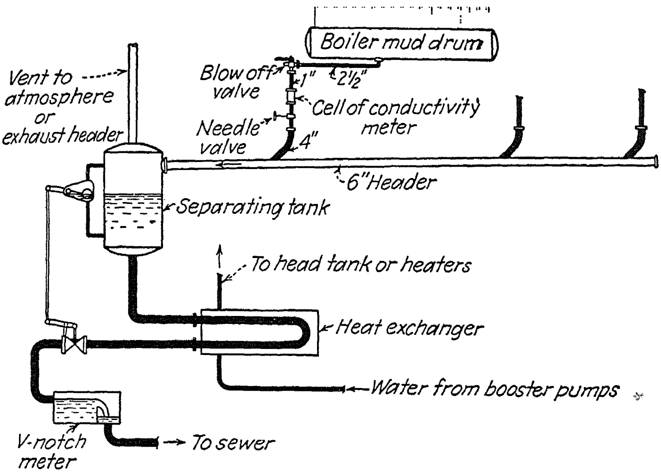


FIG. 37.—Diagrammatic arrangement of continuous blowoff system.

diagrammatically in Fig. 37. A needle valve is used to control the flow from each boiler into a header of generous size where, because of the reduction of pressure, a portion of the water flashes into steam. The header empties into a separating tank from which the flash steam is vented to an exhaust header. The remaining water flows by gravity through a heat exchanger where its temperature is reduced on its way to the sewer.

Instrument Piping.—The pressure standard chosen for instrument piping should, in general, agree with that of the line to which it is connected. Instrument piping connected to superheated steam piping need not be made of materials capable of withstanding full temperature, provided it is used in such a way as to constitute a dead end where line temperature does not obtain.

The fittings at the point of connection, however, should be suitable for the maximum temperature. Where high pressure is involved, it is not advisable to use pipe smaller than ½ in. nominal size on account of the greater danger of mechanical breakage with very small pipe. The use of ¾-in. Schedule 160 source nipples with ½-in. valves counterbored to receive ¾-in. pipe at the inlet end and ½-in. pipe at the outlet is a preferred construction (see

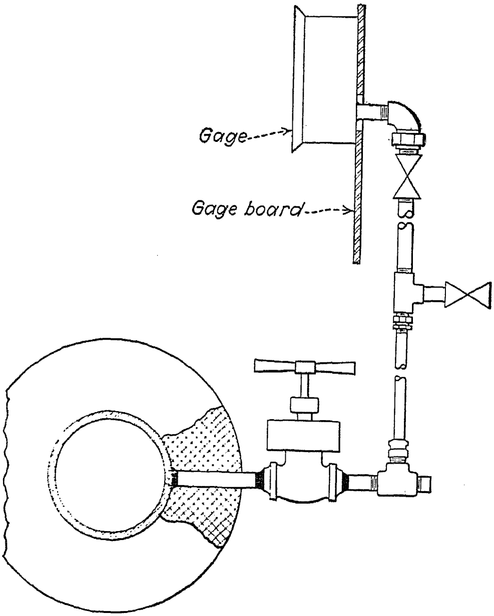


FIG. 38.—Typical assembly of instrument piping.

Fig. 38). Copper tubing is sometimes used for instrument piping because of the ease with which it is bent and its neat appearance. Difficulty in properly supporting long runs of copper tubing and higher cost are factors which tend to curtail its use. Thin-walled steel tubing with flared tube connectors or seamless steel pipe is recommended for temperatures above 406 F.

A typical assembly drawing of instrument piping showing source connection and lines to gauge board for high-pressure and high-

temperature installations is given on Fig. 38. The source nipple corresponds to Schedule 160 pipe. Nipple is made of carbon steel for temperatures up to 850 F and of carbon molybdenum for temperatures from 850 to 925 F. The source valve conforms to the 1,500-lb standard or to whatever standard is used for the main line.

The instrument piping proper begins at the outlet of the source valve. For line temperatures above 406 F and pressures from 200- to 1,200-lb, seamless-steel thin-walled tubing and steel-flared tube connectors or seamless-steel pipe and forged-steel screwed fittings are used. For temperatures below 406 F and pressures from 200 to 1,200 lb, copper tubing and brass-flared tube connectors are usually employed. Low-pressure lines, pressures 0 to 200 lb, under dead-end service, such as connections to pressure gauges, copper tubing, and brass-flared tube connectors are used for temperatures up to 700 F. For continuous service at these pressures for temperatures above 406 F, steel tubing and steel-flared tube connectors are required. For service under vacuum, copper tubing and soldered type brass fittings are used. Steel bar-stock valves are used at gauge and on open blow.

The thicknesses of steel and copper tubing suitable for service at pressures up to 1,200 psi under the temperature limitations described above are tabulated as follows:

Nominal size	Outside diameter	Wall thickness, inches	
		Seamless copper	Seamless steel
5/16 O.D.	0.3125	0.030	
1/2 I.P.S.	0.840	.....	0.147
5/8 O.D.	0.625	0.070	0.060
3/4 O.D.	0.750	0.080	
1 I.P.S.	1.315	.....	0.179

Steam gauges, thermometers, flow meters, etc., should be located where they are readily accessible, can be easily read, and are so connected as to ensure accurate readings. Standard *steam gauges* have a 1/4-in. male connection and are generally provided with a stop cock or valve at the gauge. The Bourdon tubes in steam gauge may become softened when subjected to temperatures of more than 150 F, so that steam or very hot water should not come in direct contact with the tube unless stainless-steel tubes or other

temperature-resistant material is used. A gooseneck siphon, or loop (Fig. 39) is used to maintain a protective water seal between

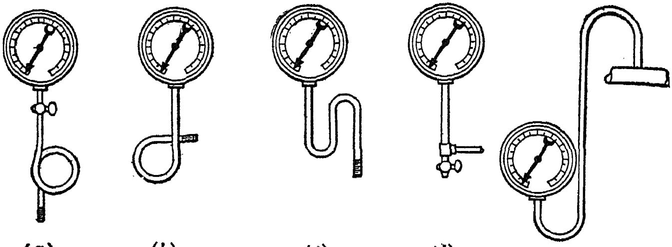


FIG. 39.—Siphons for steam gauges.

the gauge and steam supply. When the gauge is exposed to freezing a pet cock (Fig. 39*d*) should be provided for draining water from the siphon. This pet cock should not be opened when the gauge is in service, as the water seal would be lost and the Bourdon tube would be liable to damage from contact with the steam. If a gauge is placed below a pipe line (Fig. 39*e*) allowance must be made for the head of water in the seal in order to obtain correct readings. Such a correction can be made by multiplying the head of water in feet by 0.433, thus reducing it to pounds per square inch, which should be deducted from the gauge readings, or the position of the needle should be changed accordingly.

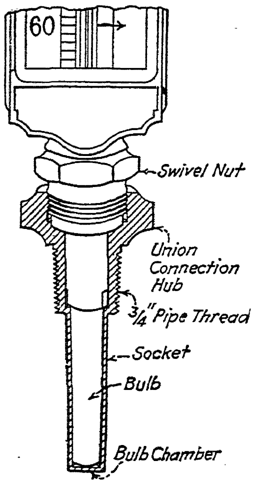


FIG. 40*a*.—Straight thermometer with separable-socket connection.

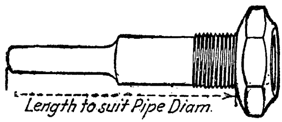


FIG. 40*b*.—Separable socket for thermometer shown in Fig. 40*a*.

Where subject to vibration, gauges should be securely attached to minimize its effect. Repeated jarring or continuous vibration

will cause wear of the rack and pinion and result in inaccurate pressure indications.

Flow meters and *Venturi meters*, which are properly classed with instrument piping, are discussed on pages 59–62. Operating *thermometers* are commonly inserted in piping by means of the separable socket fitting shown in Fig. 40*b*. An indicating thermometer

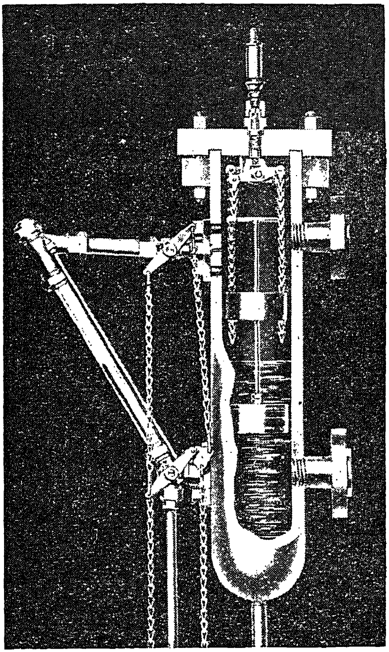


FIG. 41.—Boiler water column.

is shown in place in such a fitting in Fig. 40*a*. The same fitting can be used to receive the sensitive elements of recording thermometers, thermostats, etc. Separable sockets are made in different lengths to suit the diameter of the pipe in which they are installed. The thread which screws into the pipe is usually ¾ in. nominal size. For pressures above 600 lb and temperatures above 750 F some form of welded sockets should be used.

A typical *water column* for a power boiler for pressures up to 400 lb is shown in Fig. 41. This column is provided with a water-

gauge glass, trycocks, and two floats to operate a whistle for high- and low-water alarm. The gauge glass is provided with a blowdown and chain-operated cocks for shutting off the gauge-glass connections in case the glass breaks. A flat gauge glass is supplied for higher pressures. Suitably shaped funnels having a drain pipe are sometimes used to catch water discharged from the trycocks, and a blowdown line provided for the column and gauge glass as shown in Fig. 42. According to the requirements of the ASME Boiler Construction Code, the gate valves or cocks in the steam and water lines connecting the column to the boiler drum must be provided with a locking device in the open position.

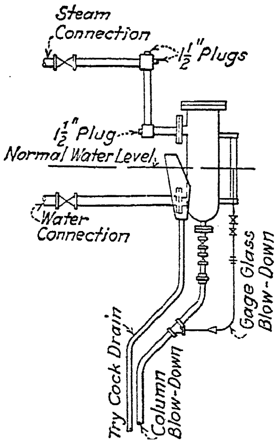


FIG. 42.—Water-column piping.

Oil Piping.—Lubricating, insulating, or fuel-oil piping in a power plant usually operates at pressures below 125 lb gauge and standard-weight black pipe and 125-lb American standard cast-iron or 150-lb malleable-iron or steel fittings are used. Valves 2 in. and smaller are usually brass. Oil, especially when hot, has a decided tendency to leak out around pipe threads and special care must be taken to see that all threaded connections are properly fitted. The white or red lead oil paste commonly used in making up threaded joints for air, steam, or water is not suitable for oil. "Key" paste, litharge and glycerin, or shellac are generally satisfactory for making up threads in oil piping although shellac should not be used where the oil temperature exceeds 150 F. Glue, red glyptol, and permatex are also used as thread paste for oil piping.

Tinning the threads of the steel pipe and brass valves and applying solder make an effective method of sealing threaded joints that is sometimes employed in turbine oil-supply lines. The use of socket-welded joints is particularly desirable for oil lines (see welding details, pages 489, 490, 505). One-sixteenth-inch-thick asbestos or composition gaskets are suitable for flanged connections.

Natural rubber should not be used for this purpose because of its tendency to rot when in contact with oil.

It is desirable that valves in oil lines be of the rising-stem type so that it can be told at a glance whether the valve is open or shut. This is particularly important in oil lines because the result of having a valve in the wrong position may not be evident for some time, and then only after damage to important equipment. Where threads are welded to ensure tightness, it is convenient to use flanged valves so that they can be removed readily for replacement or repairs.

Proper sizes for oil piping can be selected by reference to the pressure-drop data given on pages 210 to 221 and 1322 to 1334.

Drain connections on oil piping and tanks should be made through funnels so that leakage or accidental drainage can be detected readily. Municipal regulations forbid the drainage of oil into public sewer systems, and state laws forbid the pollution of streams with oil. On this account, scrupulous care should be taken to lead all oil drains to a sump where proper disposal can be made. If fire hazard is involved, this sump should be made a closed tank where combustion can be extinguished by smothering with steam or other means.

Compressed-air Piping.—Compressed-air systems in power plants usually operate at 90 to 100 lb gauge pressure for which standard-weight pipe and 125-lb cast-iron or 150-lb malleable fittings are entirely suitable. Pipe and screwed fittings are frequently galvanized as a protection against internal corrosion which is rather active, owing to the moisture present in the air. Compressed-air lines should be either dripped with traps of the bucket, float, or tilting variety or provided with manually operated condensation receivers (Fig. 17, page 897). The latter device is generally preferred for this service since it does not require frequent attention, because the amount of condensation collected is relatively small. Gate valves are generally used in the headers and principal branches, and globe valves at hose connections and other outlets. A large receiver tank is sometimes provided at a reciprocating-compressor outlet to steady pulsations and to serve as a separator for removing moisture. It is good practice to bring the suction line for a compressor from outdoors where the air is generally cooler and drier than in the plant. The inlet of an outdoor suction pipe should be properly shielded to prevent the entrance

of rain, snow, or debris. Proper sizes for compressed-air piping can be selected by reference to the pressure-drop data given on page 165. One-sixteenth-inch-thick red-rubber gaskets are suitable for use in flanged joints. Threaded joints are made up with white or red lead or some of the proprietary pastes on the market.

General Service and City Water Lines.—The operating pressure for general service and city water lines in a power plant is usually between 50 and 125 lb gauge, depending on the elevation which the water must reach. Where the city-water pressure required in the building exceeds that existing in the city mains, it is necessary to install booster pumps. General service water, so called, is usually pumped from the canal connecting with the river or lake on which the plant is situated. General service pumps are usually motor driven with one or more steam-driven spare units for emergencies. An emergency general service header is sometimes provided to supply water to indispensable equipment, such as oil coolers, water-cooled transformers, reserve boiler-feed or storage tanks, fire-protection lines, and the like, in case of failure of the usual supply. This emergency line is usually kept under pressure at all times and suitably isolated from the ordinary header by check valves. The steam-driven emergency pump can be arranged to discharge directly into the emergency header.

The suction of each pump should be equipped with a foot valve and strainer and a small line provided from the discharge header serving all pumps to prime the pump by filling the suction pipe and pump casing. In the case of large general-service and circulating-water pumps, it is frequently convenient to dispense with a foot valve and prime the pump by means of a vacuum line connected to the pump casing. By means of this vacuum line, water can be drawn up through the suction pipe until it fills the pump casing, thus priming the pump. Each pump discharge should be provided with a check valve, and a gate valve between the check valve and header. A duplex strainer, such as that shown in Fig. 43, is desirable in the discharge header to catch gravel and other small debris which may be drawn up from the canal through the coarse screen around the foot valve. Standard duplex strainer screens have a mesh with perforations varying from $\frac{3}{16}$ to $\frac{3}{8}$ in. in diameter depending on the pipe-line size, although diameters $\frac{1}{64}$ in. or smaller are furnished on special order. Standard-weight pipe and American Standard 125-lb cast-iron fittings and valves are

generally used in city-water and general-service lines. Gate valves are usually preferred to globes in water lines because of their lower pressure drop. Galvanized wrought pipe is commonly used in the small sizes, and black pipe for larger, although galvanized pipe is sometimes used for all sizes. Pipe threads on water lines are commonly made with red or white lead, or with some of the pipe thread compounds offered under proprietary names. One-sixteenth-inch-thick red-rubber gaskets are generally used in flanged joints.

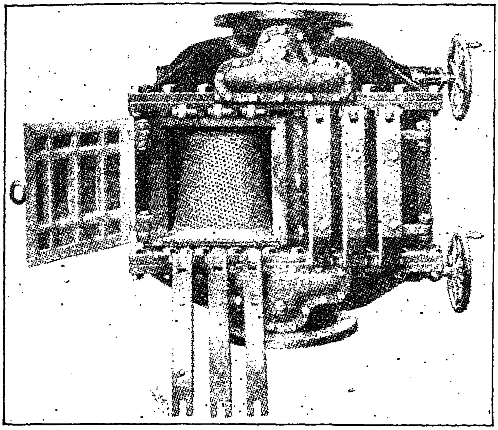


FIG. 43.—Duplex strainer.

Proper sizes for water piping can be selected by reference to the pressure drop data given on pages 268 to 289. Usual velocities in power plant work are given on page 865.

Welded Construction.—The development of welding processes and methods of quality control described in Chap. IV has made it possible to construct welded piping systems for high-pressure high-temperature service which possess a degree of reliability difficult, if not impossible, to obtain with bolted flanged construction. A welded branch connection with special ring and side-plate reinforcement is shown in Fig. 44. This style of reinforcement was developed by The Detroit Edison Company to take advantage of the findings of an investigation[1] that it had spon-

[1] The results of an investigation made by The Detroit Edison Company were presented at the December, 1937, meeting of the ASME in a paper entitled

sored at the University of Michigan. The valve shown in the photograph has a welded bonnet, welding ends, socket-welded by-pass, and an inspection opening closed with a welding cap. The extensive use of welding in the assembly of valves has become feasible through improved seat-facing materials such as stellite or colmonoy overlays which stand up indefinitely under severe

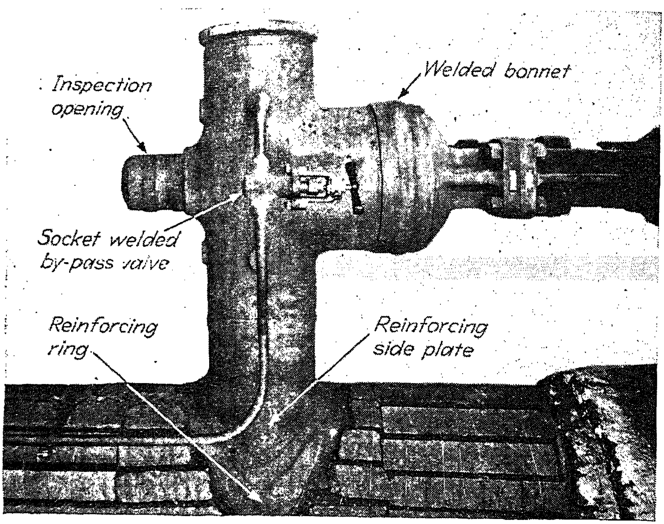


FIG. 44.—Reinforced welded branch connection and welded-bonnet valve.

service conditions. Thus, Fig. 44 serves to show in a single illustration several of the important developments made possible through welded construction.

Methods of beveling header and branch are illustrated by sketches in Fig. 45. The choice of method depends to some extent upon the equipment available for cutting and fitting the mating parts.

Dry-vacuum Piping.—Standard-weight black pipe and Class 125 American Standard cast-iron flanges and fittings are generally used for dry-vacuum piping. Sufficient strength is required to

"Investigation of Stress Conditions in a Full-size Welded Branch Connection," by F. L. Everett and Arthur McCutchan, *Trans. ASME*, FSP60-12, Vol. 60, pp. 399–410, 1938.

withstand collapsing pressures due to atmosphere, to resist corrosion, and to carry the dead weight between supports. Red-rubber

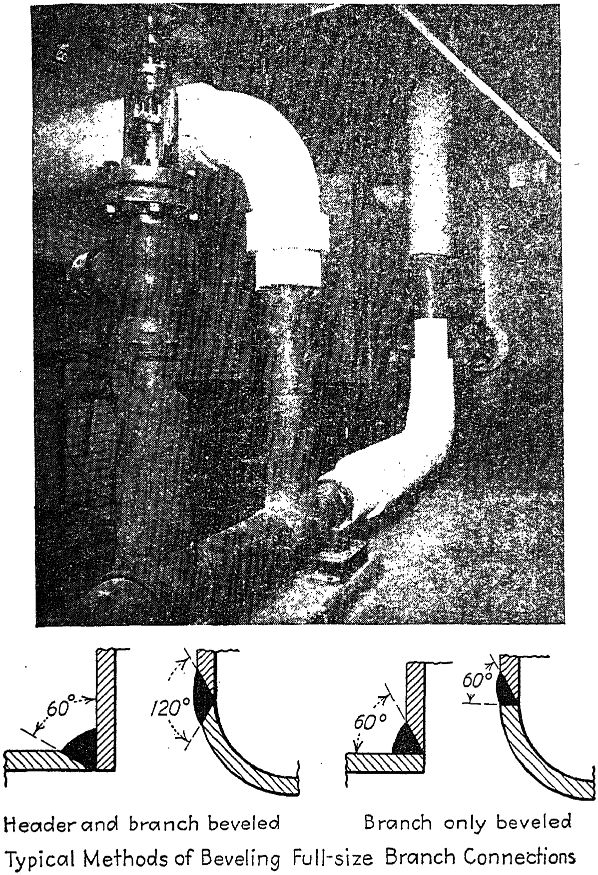


Fig. 45.—Welded steam header and details of beveling full-size welded branch connections.

gaskets $\frac{1}{16}$ in. thick are suitable in flanged connections. Threaded joints are made up with red or white lead or some of the proprietary paste compounds.

OPERATING CONVENIENCE

In laying out power-plant piping, provision for convenience in operating and maintaining piping and equipment comes next in importance after the mechanical-design features discussed in preceding paragraphs. Before attempting to generalize on how to obtain a convenient arrangement, it is necessary to point out several examples of what is good or bad practice from a standpoint of convenience. The design methods which should be employed to ensure getting satisfactory operating convenience are identical with those discussed under "Appearance."

Accessibility.—It is of paramount importance that valves which are to be operated frequently, or operated on short notice in case of emergency, be made readily accessible. If the handwheel of a valve is much more than 6 ft above the floor, it is impossible for a man of average height to operate it unless a chain wheel, platform, walkway, permanent ladder, or some equivalent arrangement is provided. In case a valve is located below the operating floor, it is necessary to provide an extension stem or a floor stand. Where a valve is so located that a man cannot operate its handwheel conveniently from the floor, this condition should be recognized in the design and provision made for its operation. The positions of all valve stems and handwheels should be indicated on piping drawings, rather than leaving the orientation of the valve to chance or the erecting crew. Cognizance should be taken of the necessity for taking large valves apart for maintenance, and for replacing small valves which are damaged or worn out. With large valves, it is sometimes necessary to lift the parts with the aid of chain falls, which should be taken into account in the design. In the case of a screwed valve, it is desirable to provide one or more unions so that it can be removed for replacement. The necessity for unions at strategic points in screwed piping is a feature which never should be neglected.

Overhead piping should be placed where it is accessible for painting, repair, or alteration without the erection of too much temporary scaffolding. Where practicable, all piping should be made reasonably accessible from some existing floor or walkway in the plant. Before locating piping in an inacessible place in a power plant, consideration should be given to finding another location where it will be accessible, or else providing a permanent walkway or other means of access.

Clearance.—Care should be taken to see that piping is installed so that it will not interfere with dismantling mechanical equipment, obstruct light wells, encroach on stairs, etc. Operating aisles and passageways should be kept free from obstructions. Horizontal runs of piping preferably should be suspended overhead, with a minimum clearance of 6 ft 6 in. underneath. Where piping must be installed close to the floor, it should be arranged

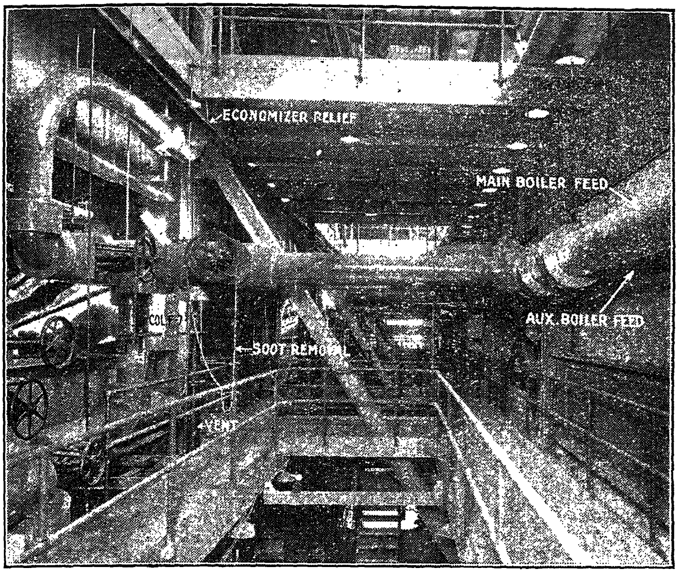


FIG. 46.—Accessibility and clearance.

to leave what passageways are required for operation, and stiles provided over the pipes at convenient locations.

Illustration of Accessibility and Clearance.—Figure 46 is included to illustrate some of the above-mentioned points concerning accessibility and clearance. This is a photograph of the boiler-feed piping at the upper drums of a large Stirling boiler, which shows the arrangement of piping with respect to a lightwell and the walkway from which valves are operated. Some of the valves, which are too high for a man to reach from the walkway, are provided with chain wheels. It will be noted that pipes are run around

the outside of the lightwell so as not to infringe on it, and that overhead pipes are high enough above the walkway for head clearance. It was impossible to include enough of the boiler-feed lines in this view to make the purpose of each valve readily apparent, although it would be so to a man standing on the walkway. The necessity for making the function of each line and valve readily understood is discussed in the next paragraph.

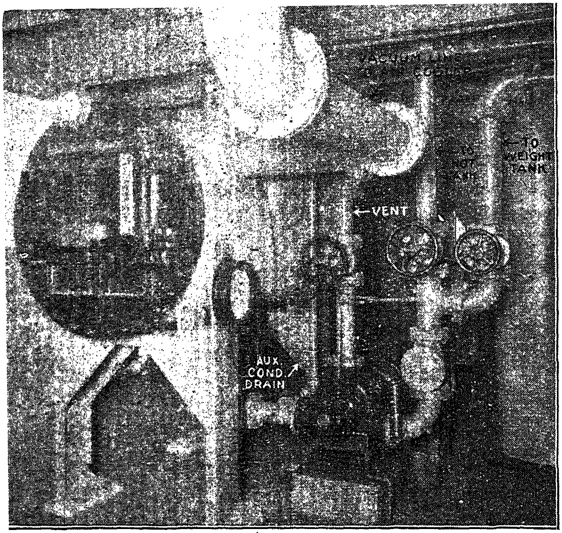


Fig. 47.—How to make the purpose of valves obvious.

Making the Purpose Obvious.—It is important, for the convenience of the operating force and to minimize the possibility of making mistakes, that piping be arranged so as to make the purpose of each line and valve as obvious as possible. If proper attention is given this feature in making a layout, it is nearly always possible to place each pipe so that its route is readily followed. The use of a systematic color scheme[1] with individual colors for each kind of service is of material assistance in identifying pipes in the plant.

[1] Reference may be made to ASA Standard A13 for a "Scheme for Identification of Piping Systems." Copies may be obtained from the American Standards Association, 29 W. 39th St., New York 18, N.Y.

An alternate method is to stencil the name of the service at frequent intervals on each pipe.

Valves in general, and by-pass valves in particular, should be installed so that their purpose is evident at a glance. It is desirable that by-pass valves be grouped together where all can be seen at once, rather than placed in scattered locations close to separate pieces of equipment.

The possibility of an error in operation can be still further reduced by stenciling the purpose of each line on the pipe close to its valve. A simple illustration of this point is furnished in a photograph, Fig. 47, of the piping for a small deaerator. One of the chief features shown in this instance is the grouping of valves at the pump rather than in scattered locations.

APPEARANCE

A piping layout should present a neat, coordinated, symmetrical, and general pleasing appearance which can best be described as good architecture. Certain definite principles can be stated which will aid in producing these results.

Clean-cut Detail.—There are certain details in the layout of individual systems which greatly affect the ensemble. Pipes should be run parallel to the axes of the building, and 45-deg turns and offsets avoided. From the standpoint of appearance, the use of elbows is generally superior to bends, although very neat work can be done with copper tubing for instrument and lines and oil piping.

Coordination with Structural Design.—Set aside certain definite lanes for piping to approach equipment, provide a pipe gallery at a suitable elevation in the boiler house, and a vertical portion of some bay adjacent to the turbine house for vertical runs of pipe, such as main steam, boiler feed, and atmospheric exhaust. Sufficient pipe trenches are required below the condenser room and boiler-house basement floors to accommodate building heating system returns, drains, hot-well pump suction lines, and the like. A room at subbasement level for the heating returns pumps with pipe trenches leading to it is often desirable.

Coordination with Equipment Location.—Arrange the plant equipment in a series of major repeating groups each comprising a main turbine, auxiliary equipment, and a group of boilers with their relative locations identical in the several groups. This

principle permits a nearly identical piping layout for each group, with consequent economies in design and erection, as well as a symmetrical and uniform appearance.

The choice of the hand of equipment (right or left hand as viewed from the same end) requires consideration. In general, single units which are repeated throughout the plant (air pumps, for example) should be of the same hand, but paired units (oil coolers, boiler-feed pumps, etc.) should be of opposite hands, because of the more symmetrical arrangement of piping which results. It is wrong in principle to arrange the main unit condensers and circulating pumps alternately right and left hand when the turbines cannot be so arranged. Where this is done, nevertheless, the alternate hand arrangement should stop at that point, and the remaining main unit auxiliaries, stage heaters, etc., should be of the same hand throughout.

Coordination between Piping Systems.—Lay out the several piping systems comprehensively rather than individually. To accomplish this, it is necessary to coordinate the various systems y means of assembly drawings. The assemblies should be completed before the individual systems are detailed. All piping going in the same direction should, in general, be arranged in a single plane. The longitudinal headers, of which there are always several, should be grouped together. Small piping, which cannot be shown on the major assembly drawings, should be installed under the direction of a competent designer.

Use of Models.—The use of models is of great help in studying layouts, and every major project should be modeled as the design progresses. After the line diagrams and preliminary piping layouts have been made, it is desirable to make a model of parts of the plant where piping is particularly congested. The use of models aids in securing a symmetrical layout of pleasing appearance which is conveniently arranged and at the same time reveals interferences and close clearances. Models may be as extensive or as curtailed as any particular case requires, but they should be made to scale and show the relative positions of piping, equipment, and building structure. The proper use of models pays for their cost many times over by (1) eliminating interference between building steel and piping or equipment; (2) providing sufficient headroom and dismantling space; and (3) obtaining good appearance and operating convenience. Models should be revised progressively as the design advances and before final working plans

of the various drafting groups are approved, until a generally satisfactory layout is assured.

Draftsmen sometimes are inclined to feel that a model is not required in so far as detection of interferences is concerned. Their view is that a competent draftsman carries the equivalent of a model in his mind when making a piping layout. The operators, engineers, and executives, however, find models useful in criticizing proposed layouts with a view to securing more workable locations of piping and equipment. The better understanding of just how

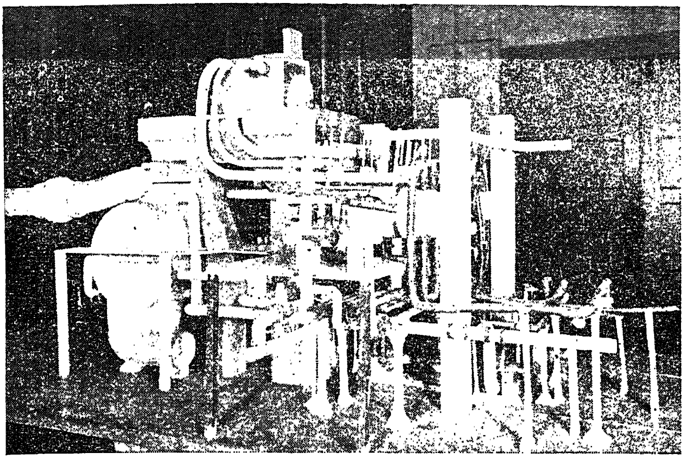


FIG. 48.—Model of piping—75,000-kw unit.

the final design will appear does much to eliminate dissatisfaction with the layouts finally agreed upon.

The materials used in constructing models vary with the inclination of the modelmaker. In the model shop of a large utility, plywood is used for floors, partitions, and boiler walls; white pine for equipment, columns, stairways, coal downspouts, etc.; wood, wire, tubing, or extruded wax[1] for piping. Models made out of transparent plastics in the form of sheets and rods have been used in the chemical industry.[2]

[1] See article on molding pipe and fittings to scale out of beeswax by an extrusion process by C. T. Van Dusen in the Feb. 23, 1926, issue of *Power*.

[2] See article on "Scale Models," by Willard Allphin in *Chemical Engineering News*, Apr. 23, 1943, pp. 556–558.

The convenience of a piping layout from an operating point of view is of great importance. The arrangement of a by-pass, for instance, should be such that the purpose of each valve is evident at a glance. In cases where there is a repetition of certain units, such as boiler-feed pumps, at intervals throughout the plant, the positions of valves, crossovers, etc., should be as nearly as possible identical. Such duplication is desirable not only from the viewpoint of appearance, but it is of immense value to the operating force in the ease with which any valve can be located in case of emergency. The extra attention given such matters in design is amply repaid by the greater convenience of a well-thought-out arrangement. It is also frequently desirable to use more pipe and fittings than required for the smallest initial investment, in order to obtain simplicity in operation and repetition of detail at similar units. A model is of great assistance in working out these features of design, as it reveals at once the advantages in different valve locations, and the space left available for access to or dismantling of equipment. Figure 48 illustrates a model study of a particularly congested spot. Unfortunately, a photograph cannot do justice to the model in this case.

The amount of detail shown in a model and the accuracy with which it is made to scale should depend on the kind of plant that it represents, the congestion of piping in the plant, and the desirability for operating convenience and neat appearance.

COSTS AND METHODS OF ESTIMATING

The total cost of piping for steam-power plants can be estimated fairly closely by using over-all figures of dollars per kilowatt of installed plant capacity, if the size and type of plants compared are not too greatly dissimilar. In estimating the costs of the separate piping systems in the plant, however, there is likely to be considerable variance, because of the differences in the design and extent of the corresponding piping systems in the plants which are compared. To overcome this difficulty, the relation between the material costs and the erection labor costs are useful. Thus, given a particular system, the material can be readily listed from the drawings and priced, and the labor estimated as a percentage of the material.

The direct material and labor costs for the piping systems of a typical large modern power plant in dollars per kilowatt of installed generating capacity are given in Table VI. The labor costs include

erection of pipe, welding, insulating, painting, etc., in the plant. In addition, the shop fabrication of bends, welded manifolds, and subassemblies is included in the labor costs. Where welding is used, as in this installation, the labor costs represent a much larger percentage of the total cost than in the older flanged construction employing cast-steel manifolds, cast-steel fittings, Van Stoned joints, and the like.

The use of welding in conjunction with simplification of piping layouts and general advances in piping design has made possible a considerable reduction in total cost of piping over other methods. Parts of this reduction in cost per kilowatt is, of course, incidental to the use of larger steam generating and turbine units. Overhead costs, including engineering design, dráfting, and supervision are not included in the costs given in Table VI. An allowance of from 50 to 100 per cent for overhead and contractors' profit should be made in arriving at the probable total cost of a complete piping installation erected in place.

TABLE VI.—COSTS OF STEAM POWER-PLANT PIPING FOR LARGE CENTRAL STATION, 850 LB, 925 F

Piping system	Cost per kilowatt[1]	Material, percentage	Labor, percentage
[illegible]	$0.09	29	71
Blowoff..	0.16	70	30
Condensate	0.45	64	36
Drains, equipment	0.12	32	68
Drip, steam lines	0.20	51	49
Exhaust	0.48	75	25
Feed water	0.69	77	23
Instrument	0.33	58	42
Oil	0.10	46	54
General service water	0.51	47	53
Main superheated	0.78	73	27
Auxiliary superheated	0.16	67	33
Auxiliary saturated	0.27	62	38
Vacuum	0.08	52	48
Vent	0.18	61	39
Valve control	0.06	43	57
Miscellaneous	0.30	40	60
Painting all lines	0.04	28	72
Total cost per kilowatt of installed generating capacity	$5.00	63	37

[1] Not including engineering, drafting, superintendence, contractor's profit, or other overhead which probably would amount to 50 to 100 per cent additional. Building and yard piping, such as steam heating, roof conductors, storm and sanitary drains, sewers, plumbing, etc., are not included in these figures.

Building Heating.—The cost of a two-pipe steam-heating system for the boiler and turbine houses of a large power plant is detailed

in Table VII. The cost includes bleeder connections from house-service turbines with a live-steam pressure-reducing station for stand-by, all supply and return headers and branch circuits, vacuum pumps, etc., as well as the radiation itself. The costs of material and labor are direct costs which do not include engineering, drafting, contractor's profit, or other overhead.

TABLE VII.—DIRECT COST OF BUILDING-HEATING SYSTEM, 400,000-KW POWER PLANT, 20,000 SQ FT OF RADIATION[1] INSTALLED IN BOILER AND TURBINE HOUSE

Item	Cost	Percentage of total installed cost	Cost per sq ft of radiation
Material:			
Radiation	$ 5,600	18.64	$0.280
Radiator hangers, wall type	760	2.53	0.038
Paint	65	0.22	0.003
Pipe, valves, traps, and fittings	8,400	28.03	0.420
Insulation	1,000	3.30	0.050
Returns pumps, meters, and tanks	1,775	5.92	0.122
Hangers, bolts, and miscellaneous	400	1.33	0.020
Total material	$18,000	60.00	$0.900
Labor:			
Total	$12,000	40.00	$0.600
Total	$30,000	100.00	$1.500

Total cost is $1.50 per sq ft of installed radiation or 7½ cents per kw of installed capacity.

[1] Actual installed surface in direct-column radiation, two-pipe steam-heating system with vacuum returns. Office building, warehouses, garages, screen house, etc., are not included.

Plumbing.—The cost of plumbing and drains such as floor and roof drains is given in Table VIII.

TABLE VIII.—COST OF POWER-PLANT PLUMBING, FLOOR DRAINS, AND ROOF DRAINS

Average cost per kilowatt of capacity	Percentage of material	Percentage of labor
$0.175	53	47

For the underground pipes outside of buildings, including city water lines, sewers, drains, service water, etc., there should be allowed approximately $0.12 per kw of capacity. Water and oil piping for large transformers located on concrete mats out of doors costs about $0.10 per kw.

The labor costs on which all the above figures are based are as follows.

Foremen	$1.07 to $1.35	per hour
Pipe fitters	1.02	per hour
Hanger men	0.72	per hour
Pipe coverers	1.02	per hour
Helpers	0.62	per hour
Common labor	0.42	per hour

The cost of cast-iron fittings for 125- and 250-lb pressure was between 4½ and 6 cents per pound.

The division of the material and labor costs among the pipe and fittings, the hangers, and the insulation is of value in making estimates. This subdivision is given in Table IX.

TABLE IX.—DIVISION OF MATERIAL AND LABOR COSTS FOR PIPING, AVERAGE FOR ENTIRE STEAM POWER PLANT

Item	Material, per cent	Labor, per cent
Pipe and fittings	90	67
Hangers	6	11
Insulation	4	9
Fabrication		
Handling and miscellaneous		
	100	100

CHAPTER IX

BUILDING HEATING SYSTEMS

By J. H. Walker[1]

HEAT REQUIREMENTS

Temperatures.—Before making the calculations for the radiators and piping of a heating system, it is necessary to select the inside and outside temperatures to be used as a basis for design. The inside temperature naturally depends upon the type of building. In residences 70 F is usually assumed for most of the rooms, with 75 F for bathrooms and sometimes for dressing rooms. In factories 50 to 60 F is satisfactory where heavy work is to be done and 65 F for light work. In offices, schools, and stores 70 F is the usual assumption and in storage spaces 40 to 50 F.

The outside temperature should not be chosen as the lowest recorded temperature for the locality because it is not necessary to design for such an infrequent condition. A temperature 10 to 15 degrees above the lowest recorded temperature is a safe figure. Local practice varies considerably in this respect, and enquiries as to the best practice are advisable.

Heat Transmission.—When a building is maintained at a temperature above that of the outside air, heat is constantly being dissipated from it in two ways: (1) by conduction of heat through the walls, roof, and floor of the structure and (2) by the continual leakage of air into and out of the building.

The amount of heat which will flow through a wall depends upon the thermal conductivity of the materials of which the wall is built and upon the surface resistances. If we have a wall composed of successive layers of different materials of thicknesses x_1, x_2, x_3, etc., and having thermal conductivities k_1, k_2, k_3, etc.,

[1] Honorary Member, National District Heating Association; Member, ASHVE, ASME; Superintendent of Central Heating, The Detroit Edison Company.

then the transmission coefficient for the entire wall is

$$U = \frac{1}{\frac{1}{f_1} + \frac{1}{f_2} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3}, \text{ etc.}}$$

where U = heat loss, Btu per square foot of wall area per hour, per degree F difference in temperature.

$x_1, x_2,$. = thickness of the successive layers composing the wall,

k_1, k_2, k_3 = coefficient of thermal conductivity of the materials, Btu per square foot of surface per inch of thickness per hour per degree F difference in temperature.

f_1 = coefficient of heat transfer between the (still) inside air and wall surface, Btu per hour per square foot of surface per degree F difference in temperature between air and wall surface.

f_2 = coefficient of heat transfer between outside wall surface and outside (moving) air, Btu per hour per square foot of surface per degree F difference in temperature between air and wall surface.

By means of this formula and using established values for the various coefficients, the heat loss for most kinds of wall construction can be computed.

Transmission Coefficients.—Values of U for various standard forms of building construction are given in Table I.

Calculation of Conduction Losses.—The heat losses by conduction are represented by the formula

$$H_w = WU(t_r - t_o)$$

where W = area of wall, or glass, or roof, sq ft.

U = transmission coefficient (see above).

t_r = room temperature, F.

t_o = outside temperature, F.

Infiltration.—The simplest method of computing the heat loss due to infiltration is to assume a certain number of air changes per hour. For ordinary rooms with one side exposed, one air change per hour is usually assumed. For greater exposure one and one-half to two air changes may be assumed; and for rooms such as entrance halls, stores, etc., three air changes per hour should be assumed.

TABLE I.[1]—VALUES OF U. COEFFICIENT OF HEAT TRANSMISSION
(Btu per square foot per hour per degree F temperature difference between air inside and air outside)

Type of construction	Wall thickness, inches	U, 15-mile wind
Walls		
Solid brick wall		0.50
	12	0.36
	16	0.28
Brick wall, 4 in. thick, hollow tile inside, plastered	6 (tile)	0.34
	8	0.33
	10	0.32
Concrete, with or without stucco finish	6	0.79
	10	0.62
	16	0.48
	20	0.41
Concrete blocks		0.56
Concrete blocks, cores filled with cork		0.41
Cinder blocks		0.42
Cinder blocks, cores filled with cinders		0.31
Cut-stone veneer with hollow tile, plaster	6 (tile)	0.35
	10	0.33
Stucco, hollow tile, and plaster	8 (tile)	0.38
	12	0.29
Wood wall clapboards or shingles, sheathing, studding, lath, and plaster		0.26
With rock-wool fill between studding		0.072
Stucco, sheathing, studding, lath, and plaster		0.32
With rock-wool fill between studding		0.076
Brick veneer, sheathing, studding, lath and plaster		0.28
With rock-wool fill between studding		0.074
Hollow glass block		0.49
Windows		
Single, wood or metal sash		1.13
Double		0.45
Solid wood doors		
1 in., nominal thickness		0.69
1 in., nominal thickness with glass storm door		0.42
2 in., nominal thickness		0.46
2 in., nominal thickness with glass storm door		0.32
Floors on ground		
All types of concrete floor on ground, and basement walls below grade		0.10
Ceilings		
Metal lath and plaster only		0.69
Metal lath and plaster, floor above		0.30
Metal lath and plaster, 3⅝ in. rock-wool fill, no flooring		0.079
Roofs		
Wood shingles on sheathing		0.46
Concrete 4 in., paper, tar, and gravel		0.72
Wood 1 in., paper, tar, and gravel		0.49
Flat metal		0.95

[1] Reproduced by permission from "Heating, Ventilating and Air Conditioning Guide," 1943, which contains more extensive tables. Copies may be obtained from the American Society of Heating and Ventilating Engineers, 51 Madison Ave., New York 10, N.Y.

The heat loss due to air infiltration in Btu per hour is

$$H_a = \frac{V(t_r - t_o)n}{55.2}$$

where V = volume of room, cu ft.

n = number of air changes per hour.

If accurate results are desired, it is preferable to use some method which takes into account the length of crack around the windows and doors.[1]

Computation of Heat Loss.—The total heat loss is the sum of the loss through the walls, glass, etc., plus the loss due to infiltration, or

$$H = S_wU_w(t_r - t_o) + S_gU_g(t_r - t_o) + S_rU_r(t_r - t_o) + \frac{V(t_r - t_o)n}{55.2}$$

where H = heat loss, Btu per hr.

S_w, S_g, and S_r = respectively, the net areas of exposed wall, glass, and roof areas, sq ft.

U_w, U_g, U_r = the coefficients of heat transmission for the respective materials.

t_r, t_o, V, and n are as defined before.

If there are partitions enclosing unheated spaces, the heat loss through them must be added, also the heat loss through ceilings with a cold attic above and the loss through floors laid on the ground. The values of t_o in such cases must be assumed and the usual assumptions are

	Degrees F
t_o for attics	25 to 40
t_o for cold cellars	32
t_o for earth below floors	50

Usually, an allowance of 15 per cent is added to the heat loss from the sides of the building exposed to the prevailing winter winds.

The total heat loss from each room is computed separately by the foregoing method and the heat losses are used as a basis for computing the amount of radiator surface to be installed.

RADIATORS, CONVECTORS, AND UNIT HEATERS

Radiators.—There are two forms of cast-iron radiators, differing as to the method of joining the sections. In the screwed-nipple

[1] See "ASHVE Guide."

type the sections are joined by nipples having right- and left-hand threads. In the push-nipple type they are connected by accurately fitted tapered nipples and drawn together by means of tie rods. Radiators were formerly made in both steam and water types, the sections of the latter being connected at both top and bottom and those of the former at the bottom only. The steam-type radiator has been discontinued by most manufacturers, since the water type serves both purposes.

Radiator Heating Surface and Tappings.—A simplified schedule of stock sizes of cast-iron radiators has been formulated by radiator manufacturers in cooperation with the U.S. Department of Commerce, and issued as Simplified Practice Recommendation

TABLE II.—SIZES AND HEATING SURFACE OF CAST-IRON RADIATORS, TUBE TYPE

Number of tubes per section	Catalog rating per section,[1] sq ft	Section dimensions[2]				
		A Height,[3] inches	*B* Width, inches, Min	*B* Width, inches, Max	*C* Spacing, inches	*D* Leg height,[2] inches
3	1.6	25	3¼	3½	1¾	2½
4	1.6	19	4 7/16	4 13/16	1¾	2½
	1.8	22	4 7/16	4 13/16	1¾	2½
	2.0	25	4 7/16	4 13/16	1¾	2½
5	2.1	22	5⅝	6 5/16	1¾	2½
	2.4	25	5⅝	6 5/16	1¾	2½
6	1.6	14	6 13/16	8	1¾	2½
	2.3	19	6 13/16	8	1¾	2½
	3.0	25	6 13/16	8	1¾	2½
	3.7	32	6 13/16	8	1¾	2½

[1] The square foot of equivalent direct steam radiation is defined as the ability to emit 240 Btu per hr, with steam at 215 F in air at 70 F. These ratings apply only to radiators installed exposed in a normal manner; not to radiators installed behind enclosures, grilles, etc. (See ASHVE code for testing radiation.)

[2] See Fig. 1.

[3] Over-all height and leg height of radiator as made by some manufacturers is 1 in. greater than shown in columns *A* and *D*. Radiators may be furnished without legs. Where greater than standard leg heights are required, this dimension shall be 4½ in.

WALL RADIATORS

Size of Section, Inches	Square Feet of Surface
14 × 16, approximately	5
14 × 22	7
14 × 29	9

R174-43. In addition to an approved list of sizes and dimensions therefor, this recommendation specifies hydrostatic test pressure, tappings, painting, etc. The rated external surfaces of radiators for various heights and widths, as given in this simplified practice recommendation, are reproduced in Table II and Fig. 1. Table II also contains common, although not standardized, sizes of wall radiators.

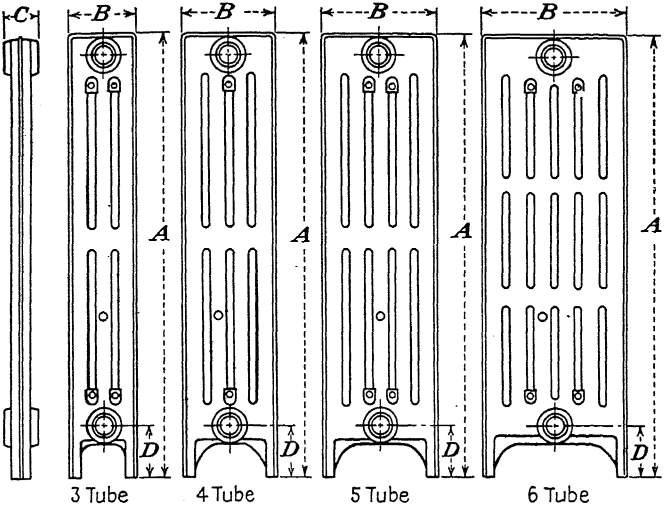


FIG. 1.—Types and dimensions of cast-iron radiators.

The end sections of cast-iron radiators usually are tapped for a 1½-in. pipe thread and furnished, as specified, with bushings having openings the size of which depends on the size of the radiator. The sizes of the reduced openings for radiators intended for use with different systems of piping should be somewhat as shown in Table III on page 946.

Heat Emission from Radiators.—Heat is emitted from an ordinary radiator partly by radiation and partly by convection. The amount of heat emitted per square foot of surface depends upon the shape of the radiator and the proportion of exposed radiating surface. The approximate heat emission from various types of

Table III.—Radiator Bushings

One-pipe steam

Size of Radiator, Square Feet	Pipe Size of Tapping, Inches
Up to 24	1
24 to 60	1¼
60 to 100	1½
Above 100	2

Two-pipe steam

	Supply	Return
Up to 48	1	¾
48 to 96	1¼	1
Above 96	1½	1¼

Hot water

	Supply	Return
Up to 40	1	1
40 to 72	1¼	1¼
Above 72	1½	1½

For vapor systems, supply, ¾ in.; return, ½ in.; air-valve tapping, ⅛ in. on all radiators.

Table IV.—Approximate Heat Emissions, Condensation Factors, and Equivalent Ratios for Various Types of Radiating Surfaces with Room Temperature of 70 F[1]

Fluid	Radiation	Heat emission, Btu per square foot per hour	Equivalent square feet of column or tube radiation		Square feet equivalent to one square foot column steam	Condensation factor, pounds per square foot per hour
			Steam[2] 240 Btu	Hot water 150 Btu		
Steam at 215 F	Column or tube	240	1.00	1.60	1.00	0.25
	Wall or pipe coil	300	1.25	2.00	0.80	0.31
	Bare pipe	360	1.50	2.40	0.67	0.38
Water at 170 F	Column or tube	150	0.62	1.00	1.60	
	Wall or pipe coil	190	0.80	1.27	1.26	
	Bare pipe	225	0.94	1.50	1.07	

[1] These are average values which do not take into account the actual number or height of columns, tubes, or coils in a radiator, or whether the surface is horizontal or vertical. For more complete data, reference may be made to the "ASHVE Guide."

[2] This column also represents heat emission relative to column steam radiation.

radiation when supplied with steam at 215 F or water at 170 F and with a room temperature of 70 F is given in Table IV.

Convectors.—Although direct radiators transmit heat by both radiation and convection, the heat-emitting unit may be built into the wall and concealed so that the greater percentage of the heat is transmitted to the room by convection through the circulation of air over the unit. This type of heating element is called a "convector." Heating elements commonly used in concealed heaters are of extended surface type made of copper, brass, cast iron, or steel. The heating element is placed low in the enclosure to secure a maximum stack effect. Control of the heat output is effected by means of a damper in the air outlet or in the stack.

Heat Emission from Convectors.—For a given unit, the heat output of a convector varies, depending on the height of the enclosure. A standard code for testing and rating concealed gravity-type radiation has been formulated by the American Society of Heating and Ventilating Engineers,[1] which has been adopted by the Convector Manufacturers Association for rating convectors. Capacities are established for various lengths and depths of unit and height of enclosure, and for several styles of installations, which are available from manufacturers' catalogues. Although published ratings frequently are given in terms of square feet of equivalent direct radiation, convector units are entirely different from direct radiation, both structurally and in their heating effect, so that the output should be considered in terms of the Btu capacity of the unit.

Radiator and Convector Performance.—The correction factors in Table V give the amount of direct radiation or convection surface required at other than standard conditions to equal one square foot of standard equivalent direct steam radiation (240 Btu). Standard conditions are assumed as 215 F steam temperature, 70 F air for radiators, and 65 F inlet air for convectors. When used for water, it should be noted that the factors in the table correct heat outputs to a 215 F base only.

Unit Heaters.—A convector in which the air circulation is produced by a fan is termed a "unit heater." As distinguished from a complete indirect-heating system or a ventilating system, unit heaters usually are installed without ductwork and at, or near, the points where heat is to be delivered. Adjustable louvers

[1] ASHVE Standard Code for Testing and Rating Concealed Gravity Type Radiation.

TABLE V.—CORRECTION FACTORS[1] FOR DIRECT CAST-IRON RADIATORS AND CONVECTOR HEATERS[2,3]

Steam pressure, approximate	Heating medium temp. F, steam or water	Factors for direct cast-iron radiators							Factors for convectors						
		Room temperature F							Inlet air temperature F						
Abs. uum i in. Hg		75	70	65	60	55	50		75	70	65	60	55		
22.4	150	2.58	2.36	2.17	2.00	1.86	1.73	1.62	3.14	2.83	2.57	2.35	2.15	1.98	1.84
20.3	160	2.17	2.00	1.86	.73	1.62	1.52	1.44	2.57	2.35	2.15	1.98	1.84	1.71	1.59
17.7	170	1.86	1.73	1.62[4]	52	1.44	1.35	1.28	2.15	1.[illegible]	1.84	1.71	1.59	1.49	1.40
14.6	180	1.62	1.52	1.44		28	1.21	1.15	1.84	1.71	1.59	1.49	1.40	1.32	1.24
10.9	190	1.44	1.35	1.28		1.15	1.10	1.05	1.59	1.49	1.40	1.32	1.24	1.17	1.11
6.5	200	1.28	1.21	1.15	10	1.05	1.00	0.96	1.40	1.32	1.24	1.17	1.11	1.05	1.00
Psi															
1	215	1.10	1.05	1.00	0.96	0.92	0.88	0.85	1.17	1.11	1.05	1.00	0.95	0.91	0.87
6	230	0.96	0.92	0.88	0.85	0.81	0.78	0.76	1.00	0.95	0.91	0.87	0 83	0.79	0.76
15	250	0.81	0.78	0.76	0.73	0.70	0.68	0 66	0.83	0.79	0.76	0.73	0.70	0.68	0.65
27	270	0.70	0.68	0.66	0.64	0.62	0.60	0.58	0.70	0.68	0.65	0.63	0.60	0.58	0.56
52	300	0.58	0.57	0.55	0.53	0.52	0.51	0 49	0.56	0.54	0.53	0.51	0.49	0.48	0.47

[1] Based on conversion factors of "ASHVE Guide" equal to $[(215 - 70)/(t_f - t_a)]^{1.3}$ for radiators, and $[(215 - 65)/(t_f - t_a)]^{1.5}$ for convectors, where t_f = fluid temperature, t_a = air temperature. Reproduced by permission from "Heating, Ventilating and Air Conditioning Guide," 1944.

[2] To determine the size of a radiator or a convector for a given space, divide the heat loss in Btu per hour by 240, and multiply the result by the proper factor from the above table.

[3] To determine the heating capacity of a radiator or a convector under conditions other than the basic ones with the heating medium at a temperature of 215 F, and the room temperature at 70 F in the case of a radiator, and the inlet air temperature at 65 F in the case of a convector, divide the heating capacities at the basic conditions by the proper factor from the above table.

[4] This factor based on the heat-emission ratio of nominal steam radiation to nominal hot-water radiation would be $240/150 = 1.60$.

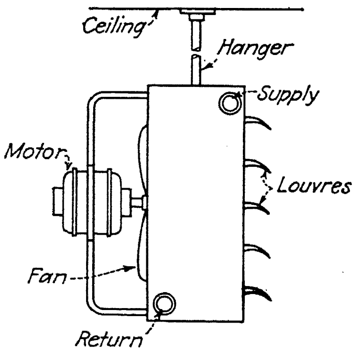


FIG. 2.—Horizontal-blow suspended unit heater. (*From "Heating, Ventilating, and Air Conditioning Guide,"* 1943.)

or directional outlets may be used to secure proper air distribution. Heating surface and fan usually are combined in a factory-built

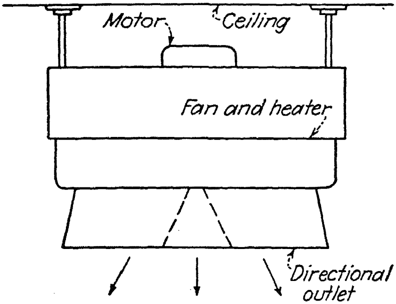


FIG. 3.—Vertical-blow suspended unit heater. (*From "Heating, Ventilating and Air Conditioning Guide,"* 1943.)

unit. The fan is used to create sufficient velocity of air discharge to transmit the air a considerable distance, which may be utilized to obtain air distribution in a large room. Unit heaters may be secured to discharge air either horizontally or vertically. Heat-transfer surface may be steel, cast iron, or nonferrous material, generally with extended surface. Unit heaters are intended primarily to handle all recirculated air but may be installed to draw either part or all outdoor air. The suspended units shown in Figs. 2 and 3, which are common types, are installed overhead, whereas the floor type shown in Fig. 4 is mounted on the floor and discharges air at about a 6-ft elevation. Steam is the usual heating medium, but ratings usually are given for both steam and hot water.

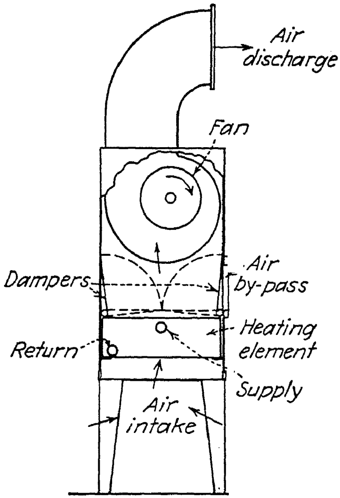


FIG. 4.—Floor-mounted unit heater, housed-type fan. (*From "Heating, Ventilating, and Air Conditioning Guide,"* 1943.)

Heat Emission of Unit Heaters.—Most manufacturers publish ratings which are determined in accordance with the standard method of testing and rating unit heaters formulated by the ASHVE.

Heat-transfer Surface.—The air in an indirect heating or ventilating system may be heated or cooled by forced convection

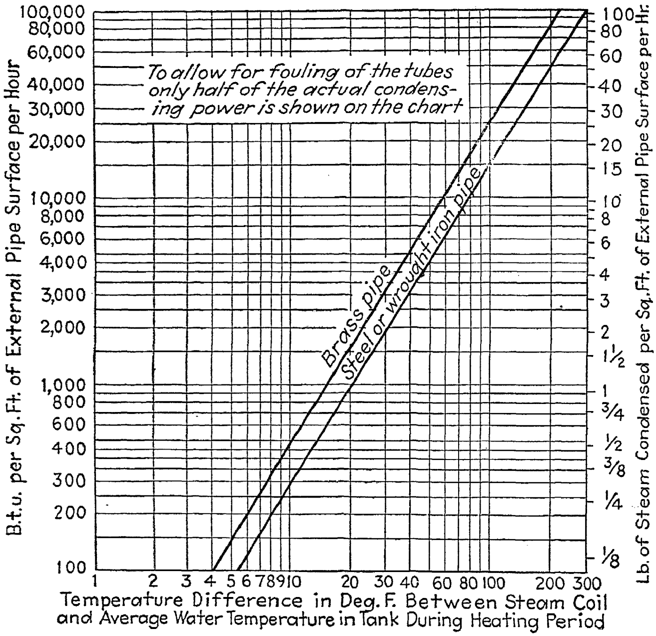


FIG. 5.—Heating capacity of brass and steel pipe coils in water storage tanks.

using extended surface steel or nonferrous coils. Cast-iron heaters of the Vento type have been used extensively for heating air but are being superseded by fin-tube type units. The extended surface units are built in a number of types, and to accommodate high-pressure steam as well as hot water, cold water, or refrigerants. The rate of heat transmission for given conditions, the final temperature of the air, the pressure drop through the unit when handling a specified quantity of air, etc., as well as piping details,

are available from manufacturers' catalogues. Proper air venting and provision for rapid removal of condensate are important considerations in piping for steam coils.

Heat Emission of Submerged Pipe Coils.—In some installations steam coils immersed in tanks are used for water heating. Water-heating capacities and corresponding boiler loads for such coils of brass or steel pipe can be read from Fig. 5. These values are for horizontal pipe coils, are based on tests made by the American Radiator Company, and contain an allowance of 50 per cent for fouling of the pipes. Consequently, the maximum load with clean new pipe may be twice that shown, which should be taken into account in sizing the boiler. All pipe surfaces are external.

STEAM-HEATING SYSTEMS

Types of Systems.—In a steam-heating system the piping and radiators must be arranged with a view to performing successfully three functions: (1) the conveying of steam to the radiators; (2)

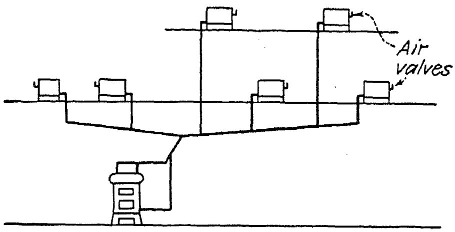


Fig. 6.—Single-pipe steam system—mains pitching toward boiler.

the removal of air from the radiators; and (3) the draining off of the condensation from the radiators and returning it (usually) to the boiler. Steam-heating systems may be divided roughly into two general classes according to the manner in which the condensation is drained from the radiators. In the *single-pipe system* the steam is conveyed to the radiator through a pipe which enters the radiator at the bottom of one of the end sections. The condensation flows out through this same pipe. In the *two-pipe system* separate return piping carries away the condensation and in many cases the air also.

An elementary form of single-pipe system is shown in Fig. 6. The single-pipe system is simple and can be installed at a low cost. Its disadvantage is that the air from the radiators must be vented

into the rooms through a multiplicity of air valves which are delicate and often fail to function properly. Figure 7 shows a better design of one-pipe system since the condensation flows through the mains in the same direction as the steam.

Figure 8 shows a single-pipe system with overhead distribution. Buildings over three stories high are usually heated by an overhead

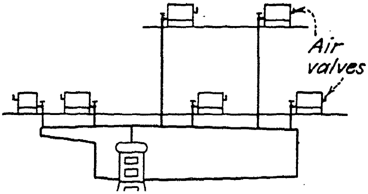


FIG. 7.—Single-pipe steam system—mains pitching away from boiler.

system because the condensate draining down the risers flows in the same direction as the steam and causes less disturbance than with upward flow. Single-pipe systems usually are limited to residences and small buildings.

There are several kinds of two-pipe systems but those in most common use are the so-called "vapor system" and the "vacuum system." In the former, each radiator is equipped at its dis-

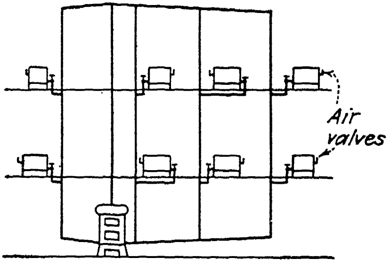


FIG. 8.—Single-pipe steam system—overhead distribution.

charge end with a thermostatic trap which allows air and water but no steam to escape into the return line. Systems of this type preferably should be provided with an automatic return trap to ensure positive return of the condensation to the boiler and to prevent water from backing out of the boiler. The term "vapor" originated from the fact that steam will circulate in a system of this

kind at a pressure of only a few ounces above atmosphere. The piping is illustrated in Fig. 9.

The vacuum system is piped almost exactly like the vapor system, except that a vacuum pump is connected to the return lines

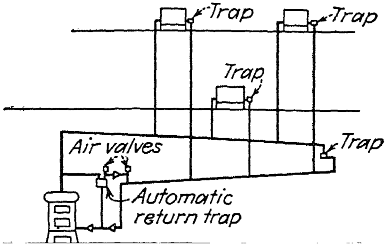


FIG. 9.—Typical up-feed vapor system with automatic return trap.

to assist the circulation and drainage and to pump the returns back into the boiler. A heating system supplying more than 10,000 sq ft of radiation usually is equipped with a vacuum pump. In vacuum-heating systems it is possible to dispense with the

TABLE VI.—PIPE SIZES FOR SUPPLY AND RETURN LINES

Pipe size	Square feet of equivalent radiation for 1 oz. pressure drop per 100 ft. of pipe							
	1	1¼	1½	2				3½
Supply mains and risers, uniflow[1]			125	190	385	635	1,165	1,740
Upfeed risers, one pipe system			100	150	290	465	800	1,145
Dry return lines	150	325	675	1,060	2,300	3,800	7,000	10,000
Wet and vacuum return lines	400	700	1,200	1,900	4,000	6,700	10,700	16,000

Pipe size	Square feet of equivalent radiation for 1 oz pressure drop per 100 ft of pipe							
	4	5	6	8	10	12	14	16
Supply mains and risers, uniflow[1]	2,460	4,600	7,500	15,500	28,500	45,500	55,000	85,000
Upfeed risers, one pipe system	1,500	2,200	3,200					
Dry return lines	15,000	23,000	37,000	78,000				
Wet and vacuum return lines	22,000	39,000	62,000					

[1] Includes supply mains and downfeed risers, steam and water flowing in same direction; branches to risers, dripped; and upfeed risers in 2-pipe systems. For contraflow (steam and water flowing in opposite directions) use next size larger pipe.

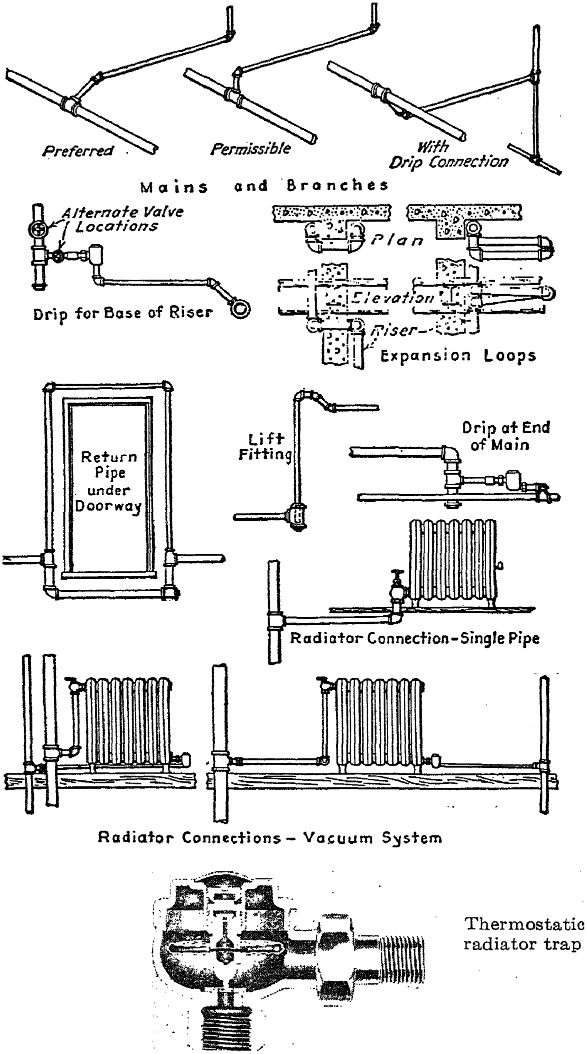


FIG. 10.—Piping details—steam heating system.

thermostatic trap on the outlet of each individual radiator and substitute a properly proportioned orifice in each steam inlet.

Figure 10 shows accepted arrangements for various details of piping in heating systems.

Pipe Sizes.—Pipe sizes are chosen from the following considerations: (*a*) The steam must be conveyed without undue pressure drop. (About 1 oz drop per 100 ft is good practice.) (*b*) If condensation is flowing in the opposite direction in the same pipe, the steam velocity must be low enough to avoid interference. (*c*) Velocities must not be so great as to cause noise. (*d*) Return piping must be large enough so as not to be clogged by moderate amounts of dirt. (*e*) A factor of safety must be allowed to cover defective installation, obstruction due to unreamed ends, etc. Tables VI and VII, giving the pipe sizes for various conditions, are based on these considerations, tempered by experience.

TABLE VII.—SIZE OF RADIATOR CONNECTIONS

Size of radiator, sq. ft.	One pipe radiators		Two pipe radiators[1]	
	Radiator connection and tapping	Horizontal branch	Horizontal supply branch[2]	Horizontal return branch[3]
20	1	1	1	¾
30	1¼	1¼	1	¾
40	1¼	1¼	1¼	¾
60	1¼	1¼	1¼	¾
80	1½	1½	1¼	¾
100	1½	2	1½	¾
150	1½	2	1½	¾
200	. . .	. . .	2	1

[1] See manufacturers' catalogues for size supply valve or return trap required. Vertical connections to be same size as valve or trap.

[2] Low-pressure steam or vapor with either gravity or vacuum return.

[3] Low-pressure steam or vapor with gravity return. With a vacuum system, a ¾-in. return branch is ample for any radiator of less than 200 sq ft. No pipe size used smaller than ¾ in.

Mains and Branches.—Proper provision for the linear expansion of the pipes is very important in a heating system. Roughly the amount of expansion is 1½ in. per 100 ft of pipe.[1] Failure to provide for it adequately results in noise as the pipes strain in warming up and in leaky joints or broken fittings.

Expansion is taken care of by the bending of pipes or by sliding-sleeve expansion joints. Sometimes arrangements are made to allow threaded joints to turn and thus absorb movement, but this is not regarded as the best practice.

[1] See also p. 756.

to handle the large amount of condensate and often is of the float type with a thermostatic air bypass. Acceptable piping arrange-

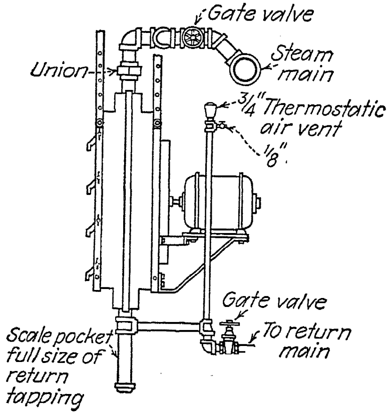


FIG. 12.—Unit heater connection to a gravity steam system using a wet return.

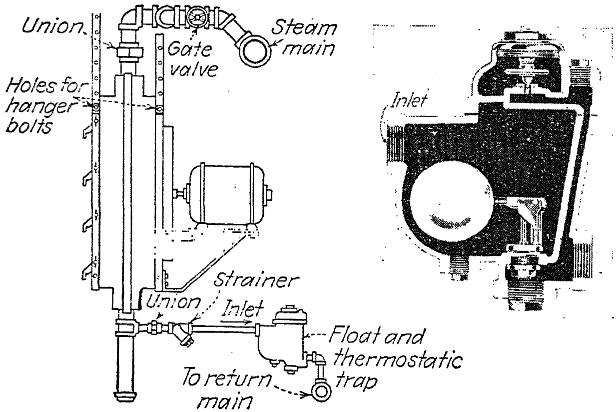


FIG. 13.—Unit heater connection for a vacuum or vapor steam system, showing detail of a float and thermostatic trap.

ments,[1] as used in connection with two types of steam systems, are shown in Figs. 12 and 13.

[1] See "ASHVE Guide" for piping details for other types of steam heating systems.

Boiler Connections.—Figure 14 shows a correct method of arranging the connections to a cast-iron heating boiler. Figure 15 shows the proper method of connecting two boilers in parallel. This is a piece of work which must be carefully done, otherwise

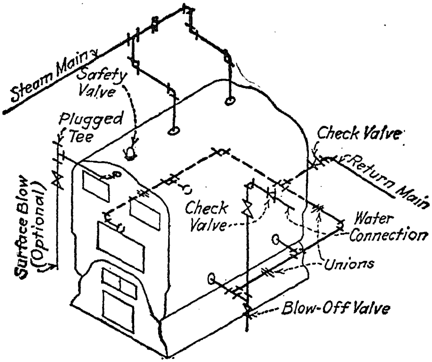


FIG. 14.—Boiler connections.

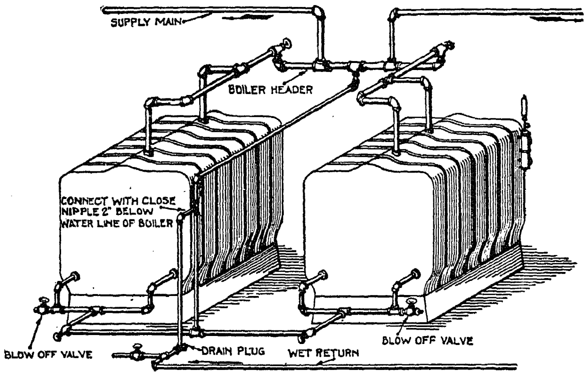


FIG. 15.—Method of connecting two boilers. Safety valves not shown.

the two boilers will not maintain the same water level and much trouble may result. The piping should be of generous size and the connections to the two boilers must be absolutely identical and symmetrically arranged. Valves may be installed as indicated, if desired, but it is better to leave them out unless it is important that one of the boilers can be cut out of service while the other is

operated. The safety valves are not shown. They are connected directly to the boiler.

The two methods of connecting wet-return lines are illustrated in these cuts. It is always desirable to guard against the water leaving the boiler through the return pipe in case of a leak. In Fig. 14 a check valve is installed in the return pipe, and in Fig. 15 the so-called "Hartford loop" is shown. The return line rises vertically to a point representing the lowest safe water level and there ties into a loop connecting the steam header and the return header. Where two boilers are connected in parallel, it often is advisable to install an individual Hartford loop on each.

If the return pipe is dry, *i.e.*, above the water line, these provisions are unnecessary, although a check valve is often installed.

Reducing and Relief Valves.—Where steam is supplied from a *district heating system* at a street pressure exceeding the safe working pressure of the building-heating apparatus, the Code for Pressure Piping[1] requires that a pressure-regulating valve be installed in the customer's premises close to the point of supply. If the street pressure exceeds 50 psi and is above the safe working pressure of the building steam-using apparatus, a relief valve or valves are required, except that if two pressure-reducing valves in series are used, both set below the safe working pressure of building equipment, no relief valve is necessary. If installation of a relief valve is not feasible, a trip stop valve may be used or, if the building is continuously attended, an alarm valve or signal may be installed instead. Relief valves shall be capable of discharging sufficient steam at full flow so that the safe pressure rating of the lower pressure piping and equipment is not exceeded.

Boiler Code Requirements.—For both steam and hot-water boilers, extensive requirements for pressure-gauge piping, water-gauge connections, safety and relief valve sizes, and other data on pipe connections, boiler fittings, and appliances are given in the ASME Boiler Construction Code for Low-pressure Heating Boilers.[1] These rules should be followed to ensure a safe installation and must be complied with in those localities where the ASME Boiler Code has been adopted as law.

HOT-WATER HEATING SYSTEMS

Types of Systems.—In a hot-water system, the circulation of water may be produced by the difference in density of the hot

[1] Copies may be obtained from the American Society of Mechanical Engineers, 29 West 39th St., New York 18, N.Y.

water in the supply pipe and the cooler water in the return pipe. In this case it is called a "gravity system," or the circulation may be produced by a pump in the circuit, this being termed "forced circulation." The piping is arranged substantially the same for both except that smaller sizes are used with forced circulation.

The simplest type of gravity-flow hot-water system with an open expansion tank is shown in Fig. 16. The mains are located in the basement, and there are separate supply and return connections to each radiator. In Fig. 17 is shown a similar system with a so-called "reversed return." The length of the circuit through each radiator on the same floor is approximately the

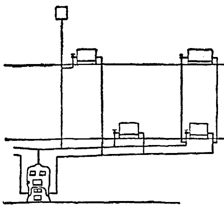


FIG. 16.—Hot-water system.

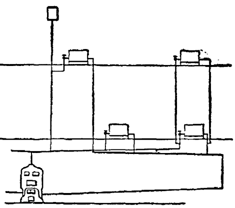


FIG. 17.—Hot-water system with reversed return.

same in this system, and the operation, therefore, tends to be more uniform.

The overhead system of distribution is often used in large buildings with the supply main circling the attic and the return main located in the basement.

Single-pipe Systems.—The systems shown in Figs. 16 and 17 are two-pipe systems and, until recently, have been the most common types. They are being superseded to a large degree in residence work by the single-main system, particularly when forced circulation is used. The single-main system has one main circling the basement, and the return connection from each radiator joins the main at a point downstream from the supply connection. Special fittings are sometimes used to aid the flow to the radiator. When the single main system is used with gravity circulation the downstream radiators should be of larger size to compensate for the lower water temperature; but with forced circulation, the water temperature to all radiators in a small system is substantially the same. Figure 18 shows a single-main system for a small building. It uses relatively small pipe sizes and is low in cost. The flow-

control valves illustrated are built like check valves and prevent flow through the heater unless the pump is operating. This permits the boiler to be maintained at operating temperature and the domestic hot water is adequately supplied by the indirect heater.

Special tees for the supply and return branches serve to aid the flow through the radiators.

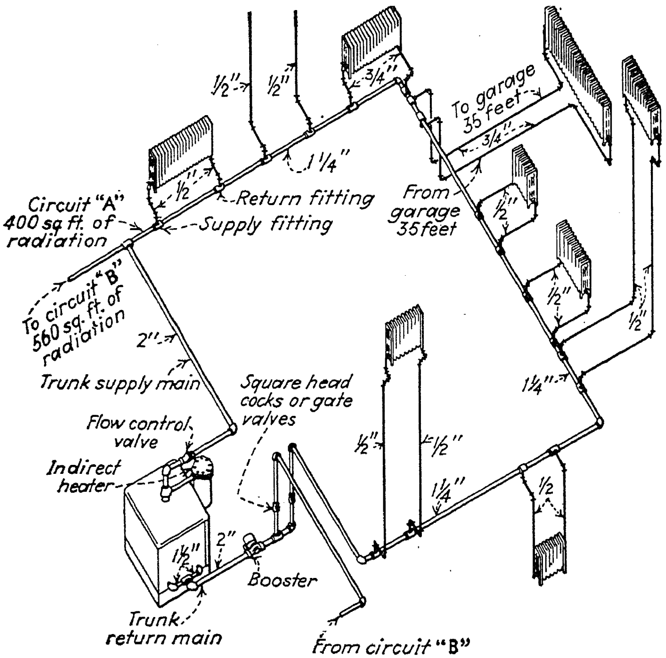


FIG. 18.—Single-main forced-circulation system. Expansion tank and relief valve not shown.

Forced Circulation.—In systems using forced circulation the head produced by the pump is usually so much greater than the gravity head that the latter can be neglected. The design procedure is similar to that used for gravity systems except for the magnitude of the heads.

In large buildings the piping is designed for friction heads up to 20 psi. In small systems, it is good practice to use a pump

having a head of 2½ to 10 ft, which permits a drastic reduction in pipe sizes and makes the single-main system practicable with resulting decrease in cost. The head for which a system should be designed is that which gives the best economic balance between the cost of the piping and the cost of the power for pumping. The circulating head having been chosen, the pipe sizes are selected so as to give an over-all friction head about equal to the circulating head.

Open and Closed Systems.—Thermal expansion of the water in the pipes and radiators usually is provided for by an expansion tank. On being heated from 40 to 200 F, water expands about 4 per cent. The volume of the expansion tank should be about twice the actual expansion, or about 8 per cent of the volumetric contents of the system including boiler, radiator, pipes, etc. For a system provided with direct radiation, the tank capacity in gallons should be approximately equal to the radiator surface in square feet divided by 40. In computing the water content of a hot-water heating system, column and tube radiation may be assumed to hold 1 pt of water per sq ft. In the open-tank system the tank is placed above the highest radiator and vented to the atmosphere. The water temperature must therefore be kept below the boiling point and seldom above 190 F.

If the tank is placed in the basement and a cushion of air under pressure maintained in it, the average water temperature in the radiators can be raised to as high as 215 F, in which case the radiators can be rated at 240 Btu per hour of heat output (the same as with steam) as compared with the 150 Btu per hour commonly assumed for the gravity system. The details of installation of pressure tanks are best obtained from the manufacturers of the special devices that are needed for their operation.

The expansion tank may be eliminated altogether if automatic spring-loaded valves are used to relieve overpressure and to admit city water when the pressure is low. The relief valve is usually set at about 28 lb pressure.

Theory of Flow in a Gravity System.—The head available for producing circulation through any radiator in a gravity system is proportional to the elevation h_r of the radiator above the boiler and to the temperature difference between the flow and return. The frictional resistance to flow varies almost directly as the length l of the circuit from the boiler and the radiator. It is, therefore, evident that the radiators for which the ratio h_r/l is

least will be the most unfavorably situated and the pipe must be chosen so as to ensure flow to them.

The head available for producing circulation is

$$h_l = h_r \frac{(D_R - D_F)}{D}$$

where h_l = head available for circulation, ft.
h_r = height of radiator above boiler, ft.
D_R = density of water in return pipe, lb per cu ft.
D_F = density of water in flow pipe, lb per cu ft.
D = average density of water.

The density D is approximately $D_R + D_F$

The head may be expressed in "milinches" (thousandths of an inch)

$$H_l = h_r \times 24{,}000 \times \frac{D_R - D_F}{D_R + D_F}$$

where H_l is the head in milinches of water. Values of H_l for various flow and return temperatures are given in Fig. 19.

Friction of Flow.—F. E. Giesecke has derived the following formula for calculating loss of head in hot-water heating systems. This formula is based on flow tests using 90 F water made at the Texas Engineering Experiment Station on *ordinary standard-weight* (*Schedule* 40) *steel or iron pipe.*[1] Data represent, not friction in a single foot of pipe, but friction in an average foot of pipe line having the normal number of fittings, which affect the friction in the straight pipe for some distance.

$$H_\lambda = 81 \frac{v^{1.84/d^{0.018}}}{d^{1.26}}$$

where H_λ = friction head per foot of pipe, milinches of water.
v = velocity, ft per sec.
d = actual internal pipe diameter, in.

The chart in Fig. 20 shows the friction head per average foot of steel or iron pipe based on the above formula.

The use of thin-walled *copper tubing* with sweated fittings has become common in hot-water heating systems. The capacities

[1] See "Hot Water Heating System Design," by F. E. Giesecke, *Heating, Piping and Air Conditioning*, August, 1941, p. 485.

of Type M copper tubing do not coincide well with steel and iron pipes of the same nominal size. (The nominal size of copper water tubing is ⅛ in. less than the actual outside diameter.) Giesecke[1]

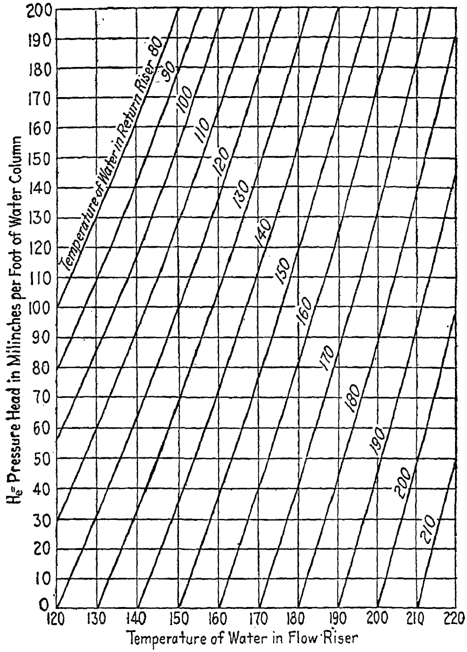


FIG. 19.—Pressure head for various flow and return temperatures.

determined that the loss of head in Type M copper tubes for 140 F water may be expressed by the formula

$$H_\lambda = 66.07 \frac{v^{1.7613 d^{0.01544}}}{d^{1.2398}}$$

where H_λ = loss of head, milinches per foot of tube.
v = velocity, ft per sec.
d = actual inside diameter of the tube, in.

The chart in Fig. 21 shows the friction head for 140 F water and Type M copper tubes based on the above formula.

[1] "Friction Heads Due to Water Flow in Copper, Brass, and Other Smooth Pipes," by F. E. Giesecke, *Jour. ASHVE*, November, 1942.

Giesecke[1] also demonstrated that the resistance of fittings varies as the square of the velocity and that resistances can be expressed

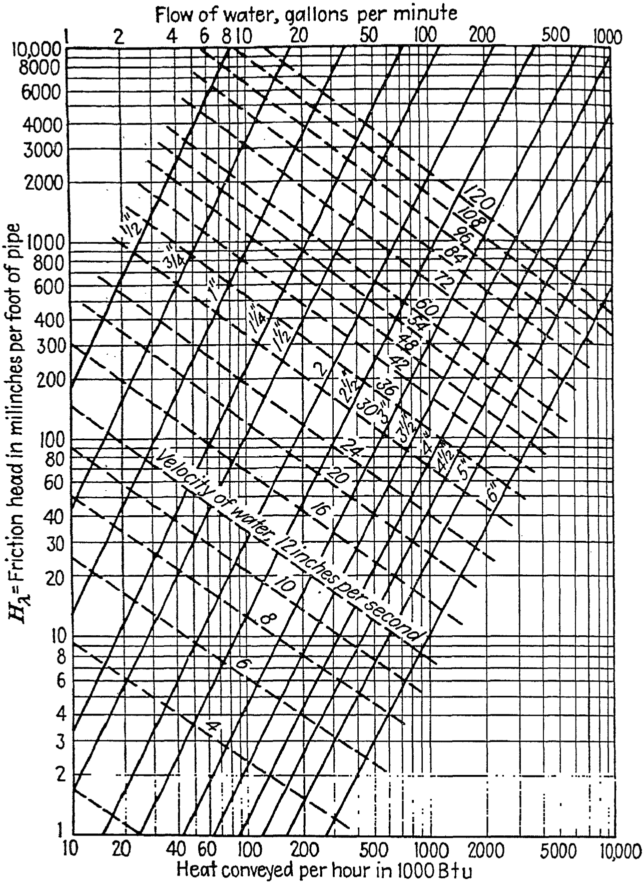


FIG. 20.—Heat conveyed and friction head per foot of standard-weight (Schedule 40) steel or iron pipe. The heat conveyed is for 20 F difference in water temperature between the supply and return mains. (From "Hot Water Heating System Design," by F. E. Giesecke, *Heating, Piping, and Air Conditioning*, August, 1941.)

[1] See "Supplementary Friction Heads in One-inch Cast-iron Tees," by F. E. Giesecke and W. H. Badgett, *Trans. ASHVE*, Vol. 38, pp. 111–120, 1932.

conveniently as equivalent to so many 90-deg screwed elbows (see Figs. 22 and 23). For this purpose he recommended the elbow

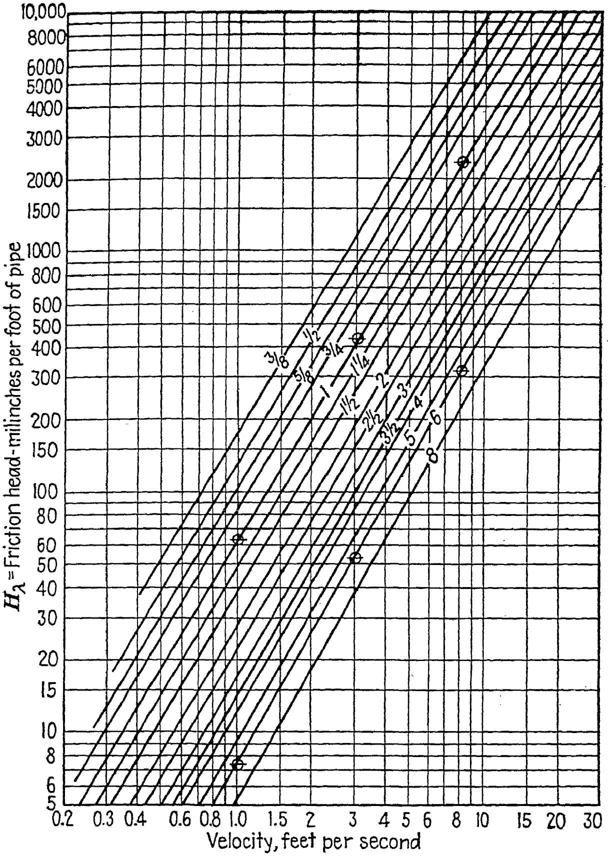


FIG. 21.—Friction head per foot of Type M copper tube, 140 F water. (*From "Friction Heads Due to Water Flow in Copper, Brass, and Other Smooth Pipes," by F. E. Giesecke, Jour. ASHVE, November,* 1942.)

equivalents given in Table IX for steel pipe and copper tubing. See also Table XIV on page 100 for equivalent resistances of bends, fittings, and valves.

Figure 22[1] shows the friction head for screwed 90-deg steel or iron elbows assuming a "no-length" elbow, *i.e.*, pipe lengths are

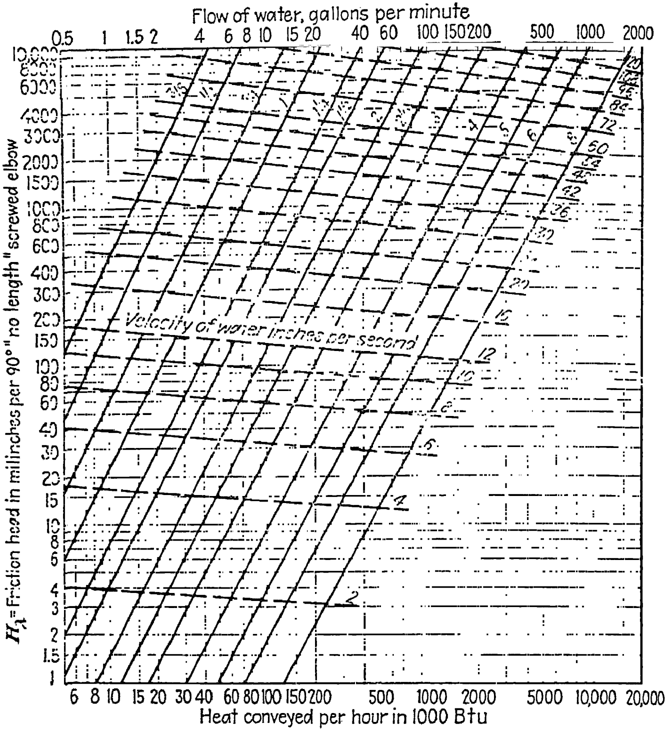


FIG. 22.—Heat conveyed and friction head in 90-day standard screwed "no length" elbows. The heat conveyed is for 20 F difference in water temperature between the supply and return mains. (*From "Hot Water Heating System Design," by F. E. Giesecke, Heating, Piping, and Air Conditioning, August,* 1941.)

figured to the intersection of center lines of the pipe rather than to the end of the pipe in the elbow.

Figure 23[2] shows the loss of head in 90-deg copper elbows. To determine the friction loss in a fitting, find the loss in head of a

[1] See "Hot Water Heating System Design," *op. cit.*, p. 486.

[2] See "Loss of Head in Copper Pipe and Fittings," by F. E. Giesecke and W. H. Badgett, *Trans. ASHVE*, 1932, pp. 529–538.

90-deg elbow from Fig. 22 or 23 and multiply it by the proper elbow equivalent of Table IX.

Selection of Pipe Sizes.—For a condition of steady flow, the available head H_t is equal to the friction head H_λ. Pipe sizes should be so chosen that this will be true with the temperature difference for which the system is designed. The charts in Figs.

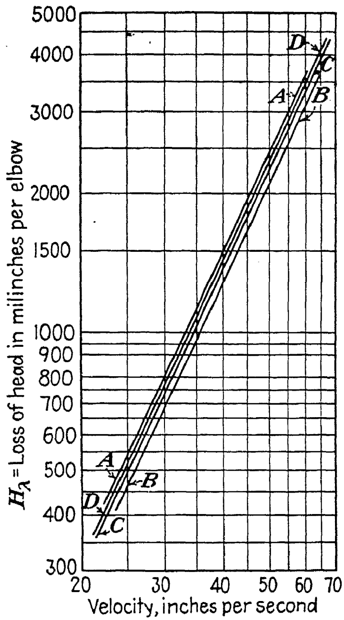


FIG. 23.—Loss of head in 90-deg copper elbows.
$A = \frac{3}{4}$ in., $B = 1$ in., $C = 1\frac{1}{4}$ in., $D = 1\frac{1}{2}$ in.

19 to 23 may be used for this purpose. The flow and return temperatures for maximum conditions must first be chosen. The best results are obtained when the temperature difference is about 20 F. The flow temperature at the boiler outlet for an open gravity system should be about 180 F.

The piping for the most unfavorably situated radiator should be considered first. The equivalent length of the circuit is computed and the average friction head per foot of pipe determined.

TABLE IX.—ELBOW EQUIVALENTS FOR FITTINGS USED WITH IRON OR STEEL PIPE AND COPPER TUBING IN HOT-WATER HEATING SYSTEMS[1]

Fitting	Number of elbow equivalents	
	Iron or steel pipe	Copper tubing
Elbow, 90 deg	1.0	1.0
Elbow, 45 deg	0.7	0.7
Elbow, 90 deg long turn	0.5	0.5
Elbow, 90 deg welded	0.5	0.5
Open return bend	1.0	1.0
Reducing coupling	0.4	0.4
Open gate valve	0.5	0.7
Open globe valve	12.0	17.0
Angle radiator valve	2.0	3.0
Radiator	3.0	4.0
Boiler or heater	3.0	4.0
Tee, equivalents for branch line flow, with following percentages diverted:		
100	1.8	1.2
50	4.0	4.0
25	16.0	20.0

[1] Reproduced by permission from "Heating, Ventilating and Air Conditioning Guide," ASHVE, 1944.

Tentative pipe sizes are selected and the calculations repeated until the friction head is equal to the available head.

Approximate Methods.—For a simple gravity system of small size, acceptable results can be obtained by the use of one of the following approximate methods:

The charts[1] (Fig. 24) show the pipe sizes required for temperature drops of 20, 30, and 40 F, using steel pipe. The vertical lines represent the total length of the circuit, divided by the effective height. The length of the circuit should be figured as if a separate pipe were carried from the heater to each individual radiator and back to the heater again. The effective height in simple circuits should be the distance between the center of the heater and the center of the radiator. In compound circuits, the mean effective height would be the Btu delivered by each separate heat-emitting unit, multiplied by its effective height, the sum being divided by the total Btu of the circuit. The horizontal lines in the charts represent the capacity of the circuit in Btu per hour and in square feet of radiation. It is better to work from the Btu side of the

[1] This method extracted, by permission, from "ASHVE Guide," 1924–1925.

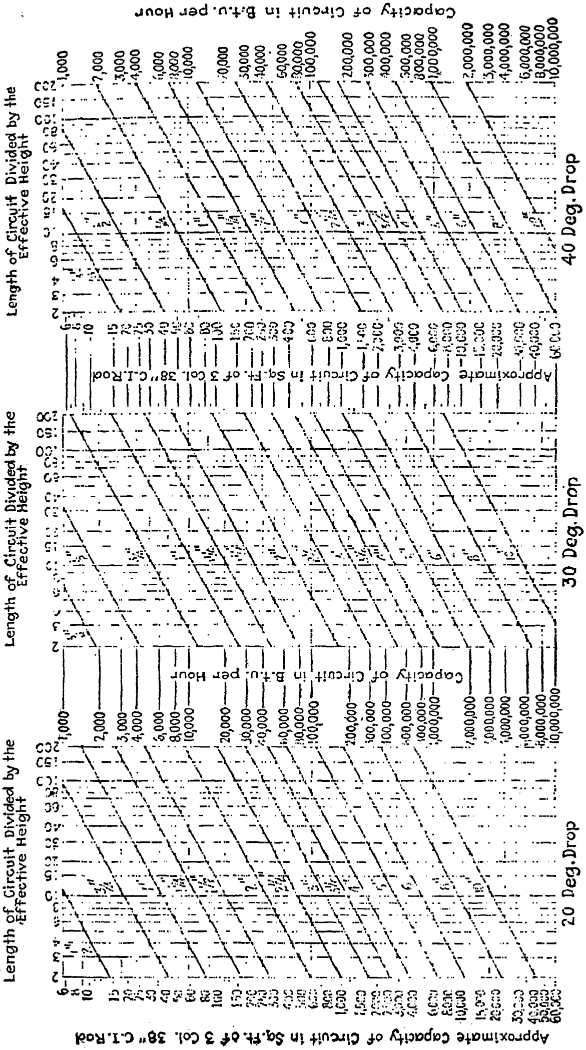


FIG. 24.—Heat carrying capacity versus pipe size, gravity hot-water heating using standard-weight (Schedule 40) steel or iron pipe.

chart. The chart for a 30-deg drop has been made opposite hand from the 20- and 40-deg charts to save space.

TABLE X.—SIZE OF MAINS, STEEL PIPE, GRAVITY FLOW HOT WATER

(Assumed length 100 ft, temperature drop in radiators 20 F)[1]

Nominal pipe size	Capacity, square feet of direct hot-water radiation		
	Two-pipe upfeed	One-pipe upfeed	Overhead
1¼	75	45	130
1½	110	65	190
2	200	121	340
2½	310	190	530
3	540	330	920
3½	780	470	1,330
4	1,100	650	1,800
5	1,900	1,100	3,200
6	3,000	1,800	5,000
8	5,900	3,500	9,900

[1] For other temperature drops, the pipe capacities may be changed accordingly. For example, with a temperature drop of 30 F the capacities shown in this table may be multiplied by 1.5.

TABLE XI.—SIZE OF RISERS, STEEL PIPE, GRAVITY FLOW HOT WATER

(Assumed temperature drop in radiators, 20 F, capacities in square feet of direct hot-water radiation)[1]

Nominal pipe size	Upfeed				Downfeed risers, not exceeding four floors
	First floor	Second floor	Third floor	Fourth floor	
1	33	46	57	64	48
1¼	71	104	124	142	112
1½	100	140	175	200	160
2	187	262	325	375	300
2½	292	410	492	580	471
3	500	755	875	1000	810

[1] For other temperature drops, the pipe capacities may be changed accordingly. For example, with a temperature drop of 30 F the capacities shown in this table may be multiplied by 1.5.

Knowing the number of Btu, or the square feet of radiation, follow the line horizontally to the vertical line representing the ratio of the effective height to the length of the circuit. The next

lower diagonal line, representing the size of pipe, gives the answer required.

For example, 10,000 Btu would be supplied to a circuit 100 ft long with 10 ft of effective height by a 1¼-in. pipe with a 20-deg drop, by a 1-in. pipe with a 30-deg drop and a 1-in. pipe with a 40-deg drop, but the total capacity in the case of the 40-deg drop would be about 17,000 heat units.

The pipe sizes, as given above, would be correct if a separate pipe were taken from the heater to each radiator, with a separate return from each radiator back to the heater. From 10 to 30 per cent would have to be added to the actual length of the circuit to allow for the additional resistance of fittings. It is customary to run all these different circuits as far as possible in a single main. This decreases the frictional resistance because of the additional capacity of the larger pipes.

These charts may be used for any type or form of gravity hot-water heating having from 6,000 to 60,000 sq ft of radiation. The principle involved is the same for residences, greenhouses, or industrial buildings. Hot-water circulation is positive and certain, provided that none of its basic principles is violated.

For a simpler and less refined method, Tables X and XI give, respectively, the capacities of mains and of risers of steel pipe size for gravity systems having a temperature drop of 20 F.

Radiator Tappings.—The schedule of tappings in Table XII is used for hot-water radiators in a gravity system.

TABLE XII.—RADIATOR TAPPINGS

Size of radiator, square feet	Supply and return tapping, inches
Up to 40	1
40 to 72	1¼
Over 72	1½

Copper Tubing.—Capacities of Type L copper tubing in thousands of Btu per hour assuming a 20 F temperature drop through the radiators are given in Table XIII. Velocities of the water in the lines may be estimated from Fig. 21. For approximate sizes using the chart of Fig. 24 and Tables X and XI, Type K copper tubing having the same nominal size as shown for steel pipe will give comparable results.

TABLE XIII.—HEAT-CARRYING CAPACITY (HOT WATER) OF TYPE L COPPER TUBING[1]
(Thousands of Btu per hour, 20 F temperature drop in radiators)[2]

Nominal tube size,[3] inches	Friction head, milinches per foot of tube											
	720	600	480	360	300	240	180	150	120	90	75	60
3/8	10	9	8	6.8	6.2	5.4	4.6	4	3.6	3	2.8	2.4
1/2	20	18	16	13.5	12	10.8	9	8	7	6	5.4	4.7
5/8	36	30	26	22.1	20	17.8	15	13.1	11.8	9.9	9	7.9
3/4	51	46	40	34	31	28	23.2	20.5	18.1	15.3	13.9	12.1
1	104	94	82	70	63	56	47	42	37	32	28	25
1 1/4	185	169	149	125	112	100	84	75	66	56	50	44
1 1/2	300	270	235	200	180	160	134	120	105	90	81	71
2	625	560	495	420	375	335	280	250	200	188	170	150
2 1/2	1,130	1,010	890	750	680	600	500	450	395	335	305	270
3	1,840	1,650	1,450	1,210	1,100	980	820	740	650	550	490	420
3 1/2	2,750	2,480	2,170	1,840	1,650	1,450	1,210	1,100	980	820	740	650
4	3,900	3,505	3,100	2,600	2,350	2,090	1,760	1,580	1,390	1,180	1,080	950

[1] Reproduced by permission from "Heating, Ventilating and Air Conditioning Guide," ASHVE, 1944.

[2] For other temperature drops the pipe capacities may be changed accordingly. For example, with a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 1.5.

[3] The outside diameter of copper water tubing is 1/8 in. larger than the nominal size. For wall thicknesses see p. 475.

CHAPTER X

PLUMBING SYSTEMS

In this chapter are presented the allied subjects of domestic water-supply piping, waste systems, and sewers and septic tanks. A considerable number of codes, standards, and recommended practices are available in these fields and are referenced throughout this chapter together with pertinent articles and textbooks. In order to avoid repetition, references have been numbered consecutively throughout the chapter and grouped together in a single bibliography at the end.

DOMESTIC WATER-SUPPLY PIPING

The fundamental considerations in laying out water-supply systems in buildings are (1) to serve all present and possible future fixtures with the use of a minimum amount of pipe and (2) to deliver an adequate supply of water at the proper pressure and temperature without undue noise.

In a small building having only a few fixtures using water, the pipes are run as directly as possible to the various outlets. In a large building, particularly where there may be future changes and additions in the plumbing, a comprehensive system of piping is required. The common arrangement for buildings such as office buildings is to use a downfeed distribution system for both hot and cold water with supply mains at the top of the building supplying the vertical downcomers. An upfeed system, however, is satisfactory.

Layout.—Figure 1 shows, diagrammatically, the layout of such a system. The water is brought in from the city mains or other source of supply to the supply or "house" pumps and flows thence to the overhead mains. To stabilize the flow a storage tank is provided, usually as a gravity tank located at the top of the building. An alternative is to provide a pressure tank in the basement. The pump is usually controlled by the water level in the tank, a float-operated switch being used, if the pump is

motor driven, to start and stop the pump as required. In the case of the basement tank, an air cushion is maintained above the water. For satisfactory operation the gravity tank, if used, must be placed so that the water level is at least 30 ft above the highest outlet. This is particularly true when there are automatic toilet-flush valves.

In buildings over about five stories, it is desirable to provide pressure-reducing valves on the lower floors to reduce the pressure at the fixtures. Otherwise, there will be objectionable noise

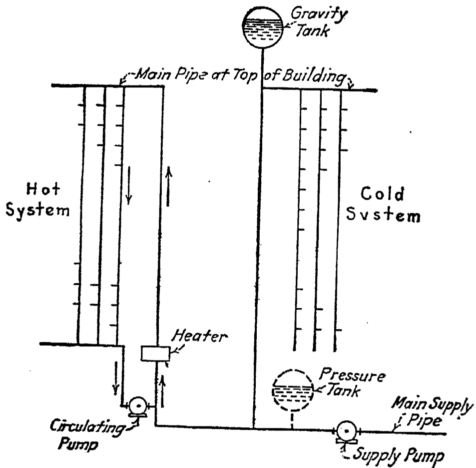


FIG. 1.—General arrangement of water-supply system.

when faucets and flush valves are opened. One reducing valve is used for a group of fixtures.

To supply the hot-water system, a connection is taken from the supply pump discharge through the heater and thence to the overhead hot-water main. For the sake of convenience and economy, the hot-water system should always be arranged for recirculation so that there will be a constant supply of hot water at each fixture. Recirculation will be produced, to a certain degree, by gravity, but in a large building a pump is desirable. It is installed in the circuit, as shown in Fig. 1, and need be designed only to overcome the frictional resistance which is seldom over 10 psi for a sufficient flow.

Cross Connections.[1]—A number of states and cities have laws providing that water and sewer installations be arranged so there is no possibility of contaminating domestic water supply by water of nonpotable quality. Although *cross connection* generally is understood to embrace any connection that permits the domestic hot- or cold-water system to become bacteriologically unsafe, chemically poisonous, or otherwise unfit to drink, the term is sometimes used specifically to denote a connection between a potable public water-supply system and a secondary or private water-supply system, the source of the latter being different from the public supply. An *interconnection* has been defined as any connection between the potable water-distributing piping and the drainage or waste pipes, *Backflow* through any connection is defined as the flow of water or other liquids into the distributing pipes of a potable supply of water from any source or sources other than its intended source.

The fluid conveyed by backflow may be sewage, used water, a dual water supply, or liquids other than water such as industrial wastes. The physical arrangements whereby such backflow may be effected include valves, overflows, drains, direct connections, check valves, extension of the domestic water system, devices operated by water, and plumbing fixture interconnections. Backflows may be induced by forces resulting from vacuums, pressures produced by pumps, drops in pressure, or independent pressure or vacuum sources.

Vacuums may be produced by dropping of a water column, by condensation of steam, or by draining a water system or main. Private piping for equipment such as filters or softeners, air conditioning, or industrial equipment operating at pressures higher than obtains in the public supply piping may cause backflow if connections between the two systems exist. Leaking check valves, or manually operated valves that are incorrectly operated, are usually responsible. Probably the most publicized type of backflow results from interconnection of potable water-supply systems and sewage systems. Plumbing fixtures may be considered to be interconnections since they are connected both to potable water and to the sewer.

Several methods are used to protect against backflow, such as use of double check-valves, tanks with overhead discharge, and swing connections, all of which are either quite expensive or do not give the necessary protection. The tendency to produce back-

flow can be minimized by maintaining as uniform a pressure at private taps as possible; relieving vacuum conditions as soon as possible, either in the street main or a private water system; developing suitable mechanical devices to relieve vacuums and provide necessary air gaps; installing proper plumbing fixtures that are protected on the discharge side of checks with vacuum breakers or their equivalent.

The safest way to protect against backflow through cross connections and interconnections is to provide a positive air gap but, where this is not feasible, a recommended type of backflow preventer should be used. Complete protection for plumbing systems can be secured only if each outlet or plumbing fixture that is subject to backflow is individually protected. Specifications for air gaps in water-distributing piping systems and plumbing fixtures and recommended types of backflow preventers have been developed by ASA Sectional Committee A40.[2] These American Standards are entitled "Air Gaps and Backflow Preventers in Plumbing Systems," ASA A40.4 and A40.6 respectively.[a] The Committee presents the standard as an effort to develop a specification for air gaps which can be policed in the field without complicated measurements or the necessity of applying involved mathematical formulas, and to formulate recommendations as to construction, installation, performance requirements, and tests of safe backflow preventers. The latter, in place of air gaps, are recommended for installation only where absolutely necessary.

The air gap in a water-supply system is defined in ASA A40.4 as the unobstructed vertical distance through the free atmosphere between the lowest opening from any pipe or faucet supplying water to a tank or plumbing fixture and the flood-level rim of the receptacle. The minimum required air gap for generally used plumbing fixtures shall be twice the diameter of the effective opening, but not less than specified in the accompanying tabulation. The effective opening is the minimum cross-sectional area at the point of water-supply discharge measured or expressed in terms of (1) the diameter of a circle, or (2) if the opening is not circular, the diameter of a circle of equivalent cross-sectional area. With ordinary plumbing fixtures, the minimum cross-sectional area usually occurs at the seat of the control valve, but may be at the spout.

[a] Copies may be obtained from The American Society of Mechanical Engineers, 29 West 39th St., New York 18, N.Y.

MINIMUM AIR GAPS FOR GENERALLY USED PLUMBING FIXTURES
(ASA A40.4)
(All dimensions in inches)

Fixtures	Minimum air gaps	
	When not affected by near wall[a]	When affected by near wall[b]
Lavatories with effective openings not greater than ½ in. diameter	1.0	1.50
Sinks, laundry trays, and gooseneck bath faucets with effective openings not greater than ¾ in. diameter	1.5	2.25
Over-rim bath fillers with effective openings not greater than 1 in. diameter	2.0	3.00
Effective openings greater than 1 in.[c]	2 × effective opening	3 × effective opening

[a] Side walls, ribs, or similar obstructions do not affect the air gaps when spaced from the inside edge of spout opening a distance greater than three times the diameter of the effective opening for a single wall, or a distance greater than four times the diameter of the effective opening for two intersecting walls.

[b] Vertical walls, ribs, or similar obstructions extending from the water surface to or above the horizontal plane of the spout opening require greater air gaps when spaced closer to the nearest inside edge of spout opening than specified in note (a). The effect of three or more such vertical walls or ribs has not been determined. In such cases, the air gap shall be measured from the top of the walls.

[c] Few tests have been made with effective openings over 2 in. and probably application of this standard will provide a greater factor of safety than for the smaller effective openings.

Two types of backflow preventers of the vacuum breaker type are covered in the American Standard for Backflow Preventers in Plumbing Systems for Plumbing Fixtures and All Water-Connected Devices, ASA A40.6, which is combined with the standard on air gaps. A backflow preventer is defined in this standard as a device for installation in a water-supply pipe to prevent backflow of water into the water-supply system from the connections on its outlet end. Type *A* backflow preventers depend on the automatic mechanical operation of one or more moving or movable parts, wholly within the device (see Fig. 2 for one type). Type *B* backflow preventers do not depend on operation of movable parts (see Fig. 3 for this type).

These types of backflow preventers must be installed between the control valve and the fixture so as not to be subjected to water pressure, except the back pressure incidental to water flowing to the fixture. The backflow preventer must be located a sufficient height above the flood-level rim of the fixture to which it is connected so that there is no possibility of backflow. In the case of direct flush valves for water closets, the elevation must be at least

in capacity and shorter life with hot-water lines. In using Table I, it should be noted that pressure drops higher than 30 psi per 100 ft are not recommended, and that a drop of 10 psi per 100 ft is considered preferable for domestic installations.

TABLE II.—BRANCH WATER SUPPLY SIZES[a] FOR FIXTURES AND MAXIMUM FLOW IN GALLONS PER MINUTE[b]

	Number of fixtures									
	1	2	4	8	12	16	24	32	40	
Water closets:										
Gallons per minute			24					128	150	Tanks
Pipe size				1¼						
Gallons per minute			80	120			200	250	300	Flush valves
Pipe size				2						
Urinals:										
Gallons per minute		12						90	120	Tanks
Pipe size										
Gallons per minute								150	175	Flush v
Pipe size										
Lavatories and wash sinks, based upon each faucet:										
Gallons per minute				24	30			64	75	
Pipe size										
Bathtubs:										
Gallons per minute					96	112	144	192	240	
Pipe size								2½	2½	
Shower baths:										
Gallons per minute					96	128	192	256	320	8 in rain.
Pipe size							2½	2½		
Acid and slop sinks, manufacturing, kitchen and laundry:										
Gallons per minute						96	120	150	200	per bibb
Pipe size										per bibb

[a] The above sizes are based upon a pressure drop of 30 lb per 100 ft, steel pipe in "fairly rough" condition.

In estimating risers and mains, the number of gallons for water closets and urinals where flush valves are used are to be as given for tanks.

The hot water faucets are to be disregarded when estimating cold-water risers and mains.

[b] Maximum flows based on an average pressure of 30 psi at the fixture. For water pressures of 90 psi, these flows would be increased from 50 to 60 per cent. For a pressure of 5 psi, the flow would be decreased by about 40 per cent.

In estimating the water requirements,[5] both the instantaneous demand of each fixture and the combined demand of all fixtures must be considered. The combined demand is obviously much less than the sum of the individual demands, because of diversity of use. Table II gives the simultaneous flow that may be assumed

for different fixtures and the pipe sizes for branch pipes for single fixtures or a group of fixtures based on a friction loss of 30 psi per 100 ft. The pipe sizes, for satisfactory operation, must be sufficient so that an ample quantity of water will flow when the faucet is opened, and with velocities of flow in the pipes that are not so high as to cause excessive noise. Where quietness is an important factor, the water velocity should not exceed 10 ft per sec. It may

TABLE III.—SIMULTANEOUS USE OF FIXTURES IN SMALL RESIDENTIAL BUILDINGS[6]
(Rate of flow, gpm)

Type of building	Fixtures	Total gal per min all fixtures	Gal per min for fixtures in simultaneous use
Small single family house	2 Sillcocks	10.0	5.0
	2 Laundry trays	20.0	10.0
	1 Kitchen sink	7.5	
	1 Lavatory	5.0	5.0
	1 Water closet	2.5	
	1 Bathtub	10.0	
		55.0	20.0
Large single family house	2 Sillcocks	10.0	5.0
	2 Laundry trays	20.0	10.0
	1 Kitchen sink	7.5	
	3 Lavatories	15.0	5.0
	3 Water closets	7.5	2.5
	2 Bathtubs	20.0	10.0
		80.0	32.5
Two-family flat	2 Sillcocks	10.0	5.0
	4 Laundry trays	40.0	20.0
	2 Kitchen sinks	15.0	7.5
	2 Lavatories	10.0	5.0
	2 Water closets	5.0	2.5
	2 Bathtubs	20.0	
		100.0	40.0
Four-family apartment	2 Sillcocks	10.0	5.0
	6 Laundry trays	60.0	30.0
	4 Kitchen sinks	30.0	15.0
	4 Lavatories	20.0	10.0
	4 Water closets	10.0	5.0
	4 Bathtubs	40.0	
		170.0	65.0
Six-family apartment	2 Sillcocks	10.0	5.0
	8 Laundry trays	80.0	30.0
	6 Kitchen sinks	45.0	21.5
	6 Lavatories	30.0	15.0
	6 Water closets	15.0	5.0
	6 Bathtubs	60.0	10.0
		240.0	86.5

NOTE.—All water closets assumed to be of the tank style.

be observed from Table II that the volumes of water assumed do not increase proportionately to the number of fixtures because of the probable diversity of use.

In computing the size of risers, an even greater diversity of use may be assumed. One method is to assume a flow to each floor of 60 per cent of the flow, as computed from Table II, with a further discount of from 10 to as much as 40 per cent for the portions serving several floors.

For residential buildings, Table III gives the simultaneous and individual flows to the various fixtures as determined by a recent study, and Table IV gives the minimum water pressures required for satisfactory service.

TABLE IV.—PRESSURE REQUIRED TO SECURE GOOD FLOW AT DIFFERENT FIXTURES[6]

Fixture	Pressure,	Proper flow gpm
$\frac{3}{8}$-in. combined sink faucet...	10	4.5
$\frac{1}{2}$-in. combined sink faucet...	5	4.5
Ordinary basin cock..........	8	3.0
Self-closing basin cock........	12	3.0
Bathtub comp. faucet........	5	6.0
$\frac{1}{2}$-in. laundry bib...........	3	6.0
Flush valve.................	15	40.0
Ballcock....................	15	3.0
50-ft garden hose and fixtures.	25	4.0

NOTE.—Pressure indicated means pressure at fixture ahead of fully open stop with water flowing at rate indicated.

There is considerable variation in practice in respect to pipe-size selection, and it is by no means an exact science. Fortunately, in all but the largest installations, a variation of one pipe size does not involve a serious increase in cost on the one hand or a deficiency of flow on the other.

The per capita consumption of an extensive city water system usually averages from 100 to 150 gal per day. This consumption for the system as a whole should not be confused with the consumption per person for domestic water supply in houses, apartments, hotels, etc., which ranges from 25 to 75 gal per day and averages about 40 when the large number of low-income families are included. This figure covers water supply both hot and cold for toilet fixtures, bath, kitchen, and laundry but does not include lawn sprinkling. The large difference between domestic consump-

tion and that for the entire city water-supply system can be attributed to use in public institutions, such as schools, factories, office buildings, industrial plants, and commercial establishments, and for lawn sprinkling, street flushing, etc.

Hot Water.—The flow rates and pipe sizes for hot-water-supply piping may be taken from Table II, counting, of course, only such fixtures as have hot-water connections.[5] The amount of storage capacity required and the recuperative rate needed in the water heater must be determined, however, from the hourly and daily hot-water consumption, as outlined in succeeding paragraphs, rather than from the instantaneous demand.

Domestic hot-water consumption generally ranges from 10 to 20 gal per day per person including kitchen, laundry, and bath, averaging about 15 gal, and may go as high as 30 gal per person on peak days. The regularity of demand varies noticeably with the water temperature, the number of persons using the service, and the coincidence of large loads, such as those for baths and wash-days. Three means are commonly employed for supplying hot-water demands: (1) the instantaneous heater, usually gas-fired, which makes hot water as required up to maximum demand and has no previously heated supply in reserve; (2) the off-peak storage-type electric heater with tank capacity sufficient for supplying all hot-water consumption throughout the day and recuperating slowly over a period of 8 to 10 hr, usually during the night with electricity at off-peak rates; and (3) a combination of moderate storage capacity with a heating rate capable of recuperating the contents of the tank to full temperature within 1 to 3 hr. The third style is common with storage-type gas- or coal-fired heaters, and with coil heaters either used with steam or placed below the water line on boilers.

Approximate daily hot-water requirements for different types of buildings, with peak hourly percentage, and the duration in hours over which the peak extends are given in Table V. The recuperative capacity of the heater plus tank storage should equal or exceed the number of gallons corresponding to the duration of peak demand. The daily and monthly hot-water consumptions are important also in determining the amount of fuel or steam required for water heating. Water temperatures required for various hot-water services are given in Table VI. For data on this subject further reference may be made to "Handbook of the National District Heating Association" and the "ASHVE Guide."

TABLE V.—APPROXIMATE DAILY HOT-WATER REQUIREMENTS IN GALLONS FOR DIFFERENT TYPES OF BUILDINGS TOGETHER WITH PEAK-HOUR PERCENTAGE AND THE LENGTH OF TIME WHICH PEAK HOUR EXTENDS[a]

APARTMENT HOUSES, PRIVATE RESIDENCES, ETC.[b]

Rooms per house or per apartment	Number of bathrooms				
	1	2	3	4	5
1	60				
2	70				
3	80				
4	90	120			
5	100	140			
6	120	160	200		
7	140	180	220		
8	160	200	240	250	
9	180	220	260	275	
10	200	240	280	300	
11		260	300	340	
12		280	325	380	450
13		300	350	420	500
14			375	460	550
15			400	500	600
16				540	650
17				580	700
18				620	750
19					800
20					850

Peak-hour demand, 10 per cent. Duration of demand, 4 hr

HOTELS

	Gal per 24 hr
Room with basin	10
Room with bath and/or shower:	
Transient	40
Resident (bachelor)	40
Resident (women)	70
Resident (mixed)	60
Two rooms, bath, and/or shower	80
Three rooms, bath, and/or shower	100
Public bath	150
Public shower	200
Public basin	150
Slop sinks for cleaning	30

Peak hour demand, 8 per cent. Duration of demand, 6 hr on, 4 hr off, and 6 hr on.

RESTAURANTS

Meals	Hand washing	Dish washer
Per meal:		
Average cost, $0.50	0.5	1.0
Average cost, 1.00	1.0	2.0
Average cost, 1.50	1.5	4.0

Peak demand . . . . 1 meal restaurants, 20 per cent; all-day restaurants, 7 per cent.
Duration of demand:
1-meal restaurants. . 3 hr; all-day restaurants . . . . 6 hr on, 4 hr off, and 6 hr on.

LOFTS AND OFFICES

Per person:	
Office help	2.0
Factory help	5.0
Cleaning per 10,000 sq ft	30.0

[a] From data developed by the Consolidated Gas Company of New York for the American Gas Association.

[b] These consumptions are deemed to be on the high side, representing peak days for liberal users of hot water, rather than usual average values. For design purposes, they serve for estimating maximum loads which the equipment will be expected to carry.

There are a few points in the piping of an indirect water heater which should be carefully observed for satisfactory results. A well-designed installation of this kind of heater is shown in Fig. 4. The tank should be installed horizontally and as high in the basement as possible to give the maximum circulating head. If the boiler is a cast-iron sectional boiler, as illustrated, all of the sections should preferably be connected to a manifold supplying the water heater. Blowout valves and shutoff valves should be installed for cleaning purposes.

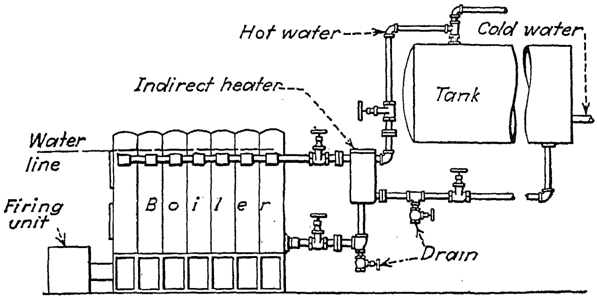


Fig. 4.—Piping for indirect water heater.

Relief Valves.—An important feature of a hot-water supply system is the relief valve, which is necessary to protect against excessive pressures. If the heater is fuel fired and the fire not properly controlled, extremely high pressures can be generated.

Table VI.—Water Temperatures Required for Various Purposes

Class of Service	Temperature of Water Used, Degrees Fahrenheit
Garages (for washing cars)	75 to 85
Hand washing	98 to 100
Shaving	125
Bathing	100 to 105
Dish washing: Hand washing	125
Machine washing	140 to 160
Washing: Silks and woolens	92 to 98
Linens and cottons	120 to 125
Swimming pools	70 to 80

Even if the heater is a steam heater or an indirect heater connected to the boiler, the thermal expansion of the water in the system can create sufficient overpressure to at least give trouble at the joints. There is often a check valve on the cold-water-supply pipe which prevents pressure relief in that direction; or, if not, the city water meter is usually so constructed as to prevent reverse flow.

The relief valve should be installed directly on the tank or heater, or at least between the heater and the first valve, so that it cannot

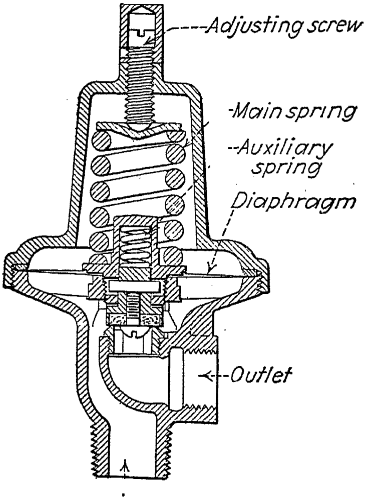


FIG. 5.—Relief valve operated by pressure.

be rendered inoperative by the closing of a valve. There should be no valve on the branch pipe leading to the relief valve.

Relief valves are sometimes constructed so as to relieve in case of excessive temperature. Figure 5 shows a pressure-relief valve of modern design, and Fig. 6 shows a combined pressure- and temperature-relief valve.

The valve in Fig. 5 has a diaphragm against which the pressure acts in opposition to a spring. The valve disk closes with the pressure, assisted by a small auxiliary spring, and is lifted from its seat when the pressure against the diaphragm overcomes the main spring.

The valve in Fig. 6 is a spring loaded valve with a cartridge at the end of the tube which contains a material with a melting point of 212 F or less as specified. The tube is inserted into the tank or heater. When the temperature is exceeded, the material in the cartridge melts, and the water pressure raises the valve disk from

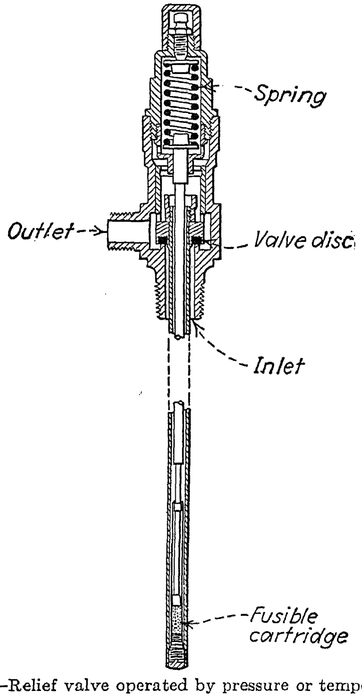


FIG. 6.—Relief valve operated by pressure or temperature.

its seat. This type of valve is particularly suitable for electric water heaters.

The regulation of water temperature is an important problem. When the use of water is small and infrequent, a storage tank may be used and the heater controlled by a thermostat from the tank temperature. In large installations, particularly where there are shower baths or other outlets which are used steadily, adequate storage is impracticable, and the heaters should be large enough to take care of the instantaneous demand.

Where shower baths are used, it is sometimes felt desirable to provide, in addition to the usual thermostatic control of the water heaters, a means for further controlling the water temperature so as to prevent accidental scalding in case the regular thermostat fails to function. One device for this purpose is a valve which admits water from the cold line into the hot line when required.

Thawing Frozen Water Pipes.—Pipes should be located so that they will not be exposed to freezing temperatures or, if this is impractical, they should be protected with insulation as discussed on page 720. Water pipes also can be kept from freezing by maintaining a sufficiently high water velocity (see pages 1070, 1071). Since ice occupies a volume about 10 per cent in excess of the same volume of water, pipes that are allowed to freeze will extend and may possibly rupture in accommodating the greater volume of ice. Because copper tubing will undergo distortion to better advantage than steel, copper tubing may withstand as many as three successive freezings without bursting; whereas pipe of other materials such as steel may fail on the first freezing.[7] If on inspection after freezing the pipe is thought to be intact, it probably is worth while to thaw it out; otherwise it may be better to remove and replace it with new pipe.

The most convenient means of thawing small frozen pipes is by electricity. The terminals of a source of power are connected as near as possible to the ends of the frozen pipe section. If the house service main between the meter and the street is frozen, connection may be made to the pipe on the street side of the meter and to a convenient fire hydrant. The power may be supplied by a transformer, or it is possible to use many standard types of arc-welding machines provided they have an output of at least 300 amp and an open circuit voltage of not to exceed 60 volts. After the extent of the pipe to be thawed has been determined, the voltage to be impressed may be approximated by allowing 5 volts for line loss and one volt for each 7 ft of pipe in the circuit.[7] Each screwed pipe fitting is counted as 1 ft of pipe. The current required varies with the size of the line and the pipe material as indicated in Table VII, which also gives the cross section of copper wire required to apply the necessary amperage and voltage.

On the smaller service lines, the period of time required for thawing is inversely proportional to the square of the current. For example, with a ½-in. service, if it takes 5 min to thaw at 250 amp, it will take 20 min to thaw at 125 amp. Since the time

required also depends on the size of the pipe, the length, pipe material, location, extent of freeze, type of pipe joints, etc., the time stated in Table VII is an approximation only. By impressing 100 to 300 amp in the case of steel pipes of average size, they usually can be thawed in 5 to 20 min, however. Larger sizes require proportionally larger currents and longer times as is evident from the table. For lead pipe, Hobart Brothers Company, manufacturers of electric welders, recommend a maximum current of 150 amp with as little as 75 amp preferred because of the low melting point and high resistance of lead. Although the lower rate of current would require a proportionately longer time to thaw the pipe, damage to the pipe would be avoided.[7]

TABLE VII.—DATA ON THAWING FROZEN WATER PIPES BY ELECTRICITY[a]

(Schedule 40 steel pipe and Class K copper tubing)

Nominal diameter of pipe or tubing, inches	Amperes required		Minutes required, minimum	Kilowatt-hours consumed		Wire size[b]	
	Wrought iron, steel, or lead pipe	Copper tubing		Wrought iron, steel, or lead pipe	Copper tubing	Wrought iron, steel, or lead pipe	Copper tubing
½	100	250	5	⅙	½	85	250
¾	150	375	5	¼	⅝	135	500
1	200	500	8	½	1¼	200	700
2	300	750	20	2	5	350	1,250
3	400	1,000	60	8	18	500	1,900
6	400	1,000	90	12	30	500	1,900

[a] Compiled principally from data contained in references 7(a), (b), and (c), p. 1001.
[b] Thousand circular mils, rubber insulation.

In any case, a complete circuit must be made, all electrical ground clamps must be removed such as those for radio, telephone, and the lighting service itself to prevent current passing to the building wiring through the ground connection. All convenient faucets should be opened to permit release of water pressure since the heating effect of the current turns the ice to hot water and eventually to steam. If the electrical circuit includes the water meter, the latter must be removed or by-passed by securely strapping a conductor around it of cross section equal to the feed connection. In starting the operation, the amperage supplied should be low and gradually increased as thawing proceeds. If the

machine, pipe, or pipe joints get too hot to touch with the bare hand, current must be reduced or switched off and on. If an ammeter in the circuit shows no current flowing, a ruptured pipe, bad joints, bad connections, or grounding is indicated. If the joints are at fault, it may be necessary to bridge them by copper strips or make secondary connections at the joints.[7] In no case should the pipe be overheated in an attempt to hasten thawing. During thawing, be prepared to shut off the water quickly in the event a pipe is ruptured to avoid water damage to building structure or contents. After thawing, water should be allowed to run full for at least a half hour to make certain that the line is free of ice.

Materials.—Both hot- and cold-water systems should be built of materials that are less subject to corrosion than ordinary black pipe. Galvanized pipe and fittings have been the common standard in the past, but in recent years use of copper tubing with sweated cast-brass or drawn-copper fittings has become quite common in the higher class of residences and in the better commercial buildings. Wrought-copper or cast-brass fittings for soldered joint construction of copper water tubes may be secured in accordance with the American Standard for Soldered Joint Fittings, ASA A40.3, dimensions for which are shown on page 590, Table LXXVII.

When the price of copper is high, its use may result in a considerably more expensive installation than with ferrous materials, but under most conditions the life is supposed to be considerably longer, especially in hot-water lines. Also, smaller size copper tubing can be used than is practicable for steel pipe because of the reduced tendency of copper and brass to corrode or form scale on the interior of the pipe, thus reducing the flow. This is particularly desirable for hot-water lines where the smaller sizes make hot water available in a shorter time after the faucet is opened, thus reducing heat losses from the pipe and causing less waste of hot water.[8] There are certain localities, however, where the character of the natural water supply is unfavorable for the use of copper. This usually can be ascertained by local inquiry. The combination of copper and steel in the same piping system, such as copper piping with a galvanized-steel hot-water tank, is not considered best practice although it frequently is done and, if water conditions are favorable, no unusual corrosion may be encountered (see page 1262).

Copper pipe and brass or copper fittings become rather costly in the larger sizes and are seldom used in plumbing work above about the 3-in. size.

Brass pipe of iron-pipe-size dimensions with threaded joints has been used in the past but has become largely superseded by copper tubing.

WASTE SYSTEMS

General Arrangement.—The disposal of storm and sanitary waste, so important from the standpoint of public health, is strictly regulated by local health authorities, whose rules should be carefully consulted in the design of any project. The suggestions and data which follow represent general practice.

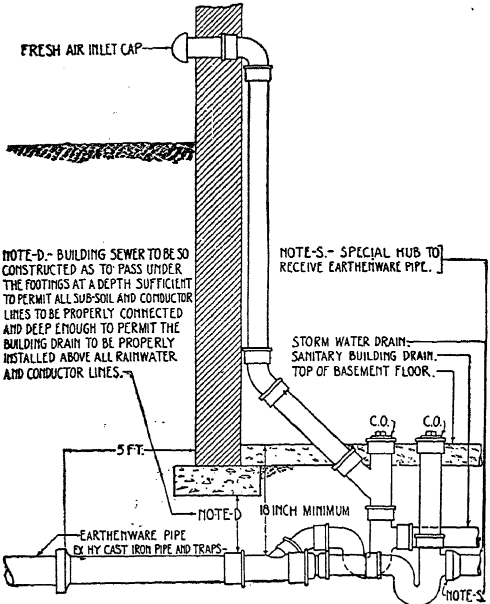
The requirements for a correct waste system are (1) adequate size and pitch of pipes to handle the maximum expected volumes without completely filling the pipe; (2) freedom from any sort of obstruction which might cause an accumulation of solid materials; (3) proper venting of gases; (4) proper sealing of inlets; (5) provision for cleaning out in case of clogging.

The ordinary building drainage system consists of one section devoted to collecting the wastes from sanitary fixtures and another for roof and floor drains. Each section has a U-shaped seal just before they tie together into a single line leading to the main street or alley sewer. There is a fresh-air inlet for the sanitary system on the building side of the trap, and each riser is vented to the roof. Figure 7 shows the usual arrangement of traps and fresh-air inlet, and Fig. 8 shows a typical bathroom layout. The riser has a parallel vent line, and each of the combination fittings is vented so that the water will flow freely down the main pipe. Without such a venting arrangement there would be air binding and danger of siphoning out the seals of the fixtures as water rushes down the vertical pipe from above.

Materials.—Vitrified-crock sewers are usually permitted in earth which is not made ground or backfill. In such cases cast-iron soil pipe with lead-calked joints is required. Cement-asbestos pipe sometimes is used to convey sewage. For a discussion of dimensions, type of joints, etc., see pages 422 to 471.

Waste pipes within buildings and not buried must be of cast iron, galvanized wrought iron or steel, lead, brass, or copper. The usual practice is to use either cast iron, with bell-and-spigot joints, or the recessed type of screwed cast-iron fitting with galvanized

pipe, the latter particularly for small work, which should be purchased in accordance with the American Standard for Screwed Cast-Iron Drainage Fittings, ASA B16.12. Dimensions and requirements for the latter are abstracted on page 457. Data on American Standard Cast-Iron Soil Pipe and Fittings, ASA A40.1, are contained on pages 450 to 453.



RELATIVE POSITION OF BUILDING AND STORM WATER TRAPS.

Fig. 7.—Sanitary and storm water traps.

Miscellaneous Requirements.—The pitch of house drainage pipes should never be less than $\frac{1}{8}$ in. per ft. All changes of direction should be gradual and not abrupt; 45-deg fittings should be used wherever possible and 90-deg fittings should be of the long-sweep pattern. All unnecessary turns or offsets should be carefully avoided and the drains run as directly as possible from the fixtures to the vertical stacks.

When there are two or more outlets on a vertical waste pipe, the lower fixtures should be protected against siphonage or air binding by venting, as shown in Fig. 8.

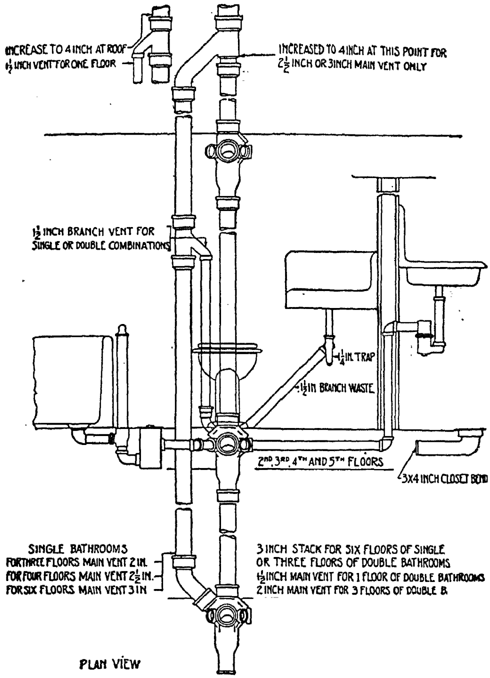


FIG. 8.—Typical bathroom waste arrangement.

TABLE VIII.—EQUIVALENTS OF VARIOUS KINDS OF FIXTURES FOR DETERMINING WASTE-PIPE SIZES

(From Par. 10.3.1, Proposed American Standard Plumbing Code, June 1944 issue)[12]

Fixture	Fixture unit value	Fixture	Fixture unit value
Lavatory with 1¼-in. trap	1	Combination laundry tub and sink with 1½-in. trap	2
Lavatory with 1½-in. trap	2	Combination laundry tub and sink with 2-in. trap	4
Bathtub with 1½-in. trap	2	Service sink (slop sink)	3
Bathtub with 2-in. trap	4	Service sink with flush valve	6
Shower stall with 1½-in. trap	2	Drinking fountain	½
Shower stall with 2-in. trap	4	Urinal—stall and washout	2
Shower stall with 3-in. trap	6	Urinal—pedestal siphon jet and blowout	4
Water closet	6	Floor drain with 3-in. trap	6
Kitchen sink with 1½-in. trap	2		
Kitchen sink with 2-in. trap	4		
Laundry tub with 1½-in. trap	2		

Each single toilet fixture must be separately trapped, except in the case of adjacent wash basins or laundry tubs, which may have a common trap.

TABLE IX.—CAPACITY OF WASTE AND SOIL PIPES IN TERMS OF EQUIVALENT CLASS *A* FIXTURES[a]
(Table 10.4.4, Proposed American Standard Plumbing Code, June 1944 issue)[12]

Pipe size, inches	Vertical soil and waste stack	Maximum fixture units that may be connected to					
		A horizontal branch or at one branch interval on stack			House drains and sewers or other drains receiving discharge from more than one branch interval		
		1/8-in. fall	1/4-in. fall	1/2-in. fall	1/8-in. fall	1/4-in. fall	1/2-in. fall
1¼	1	.....	1	1			
1½	3	.....	3	3			
2	15	.....	6	9	......	8	12
2½	35	.....	13	18	......	18	25
3	75[b]	.....	32[c]	45[c]	34[b]	46[b]	64[b]
4	400	110	150	225	150	200	300
5	1,000	250	350	490	360	500	700
6	2,200	460	700	950	625	950	1,300
8	6,000	1,400	2,000	2,800	1,950	2,800	3,900
10	12,000	3,600	5,000	6,500	5,000	7,000	9,000
12	18,000	6,300	8,400	10,500	9,000	12,000	15,000
15	......	.....	.....	......	14,000	20,000	28,000

[a] Capacities are based on Class *A* fixtures intended for the use of a family or individual, such as fixtures in residences, apartments, and fixtures in private wash or bathrooms in hotels, clubs, hospitals, or office buildings (see Table VIII for fixture unit values). For pipe sizes 2 in. and larger, the capacities for *Class B fixtures for general public use* are approximately one-half those shown in Table IX. Falls are in in. per ft of run.
[b] Not over two water closets in either Class *A* or *B*.
[c] Not over one water closet in either Class *A* or *B*.

Pipe Sizes.—The sizes of the soil or waste pipes in a building usually are determined by assigning to each kind of fixture an equivalent value and then selecting the pipe size corresponding to the total number of "equivalent" fixtures. These equivalent values are given in Table VIII. After the total number of equivalent fixtures has been determined, the proper pipe size can be chosen from Table IX. Recommended minimum sizes of traps and fixture drains for different fixtures are shown in Table X. The main building sewer should in no case be less than 6-in. size. No soil or waste stack shall be smaller than the largest horizontal branch connected thereto.

TABLE X.—RECOMMENDED MINIMUM SIZES FOR TRAPS AND FIXTURE DRAINS[9]

Fixture	Size of trap and fixture drain, inches	Fixture	Size of trap and fixture drain, inches
Bath tubs	1½	Shower stalls	2
Combination fixtures	1½	Sinks, kitchen, residence	1½
Drinking fountains	1¼	Sinks, hotel or public	2
Floor drains	2	Sinks, dishwasher	1½
Laundry trays	1½	Urinals, stall	2
Lavatories	1¼		

Storm Drains.—Roofs and paved areas, yards, etc., may be drained into a storm-sewer system, or to a combined storm-sanitary sewer system, but should not be drained into sewers intended for sanitary sewage only. When leaders or storm drains are connected to a combination sewer, they should be effectively trapped. When leaders are placed within a building, they should be made of cast iron, galvanized steel, wrought iron, or open-hearth iron, cement-lined steel, brass, copper, or lead. Outside leaders may be of sheet metal. The size of a vertical leader may be based on the maximum projected roof area as given in Table XI. The maximum pro-

TABLE XI.—MAXIMUM PROJECTED ROOF AREAS SERVED BY VERTICAL LEADERS AND STORM DRAINS OF SIZES SHOWN
(From Par. 11.3.1 and 11.3.2, Proposed American Standard Plumbing Code, June 1944 issue)[12]

Inside diameter, inches	Projected roof area, square feet			
	Vertical leaders	Horizontal storm drains		
		⅛-inch fall per foot	¼-inch fall per foot	½-inch fall per foot
2	500	350	500	720
3	1,500	1,030	1,490	2,120
4	3,100	2,230	3,320	4,610
5	5,400	5,510	7,950	11,400
6	8,400	6,480	9,300	13,320
8	17,400	13,700	19,800	28,200
10	......	24,780	35,700	50,900
12	......	40,000	57,600	72,300

NOTE.—This table is based upon a maximum rate of rainfall of 4 in. per hr. If in any state, city, or other political subdivision, the maximum rainfall is more or less than 4 in. per hr, then the above figures for roof areas shall be adjusted proportionately by multiplying the figures by the ratio to 4 in. of the maximum rate of rainfall in inches per hour.

jected roof area also determines the minimum size of building storm sewer, main storm drain, or branches in accordance with Table XI. The size of storm drains required for yard areas can be determined from the same table.

The sanitary and storm-drainage systems of a building should be entirely separate, except that both systems may be connected to a combined sanitary and storm street sewer, if one is available. In this case, it is preferable to make such connections downstream at least 10 ft from any stack connection. The size of combined drains or sewers may be computed by converting the fixture units of Table VIII to sq ft units in accordance with Table XII. This figure added to the sq ft of roof area for storm drains, is used to determine the size of the combined drain or sewer from Table XI.

TABLE XII.—FACTORS FOR CONVERTING CLASS *A* FIXTURE UNITS TO SQUARE FEET OF PROJECTED ROOF AREA
(Table 11.5.3, Proposed American Standard Plumbing Code, June 1944 issue)[12]

Total Number Connected Units	Square Feet of Area for Each Connected Fixture Unit
First 10 units, each	80
Next 10 units, each	40
Next 30 units, each	25
Next 50 units, each	20
Next 200 units, each	15
Next 500 units, each	10
Next 1,000 units, each	7
All units in excess of 1,800	5

Subsoil drains should be of open-jointed tile pipe not less than 4 in. in diameter. Underground storm drains should be of vitrified clay or concrete sewer crock or cast-iron soil pipe, not less than 4 in. in diameter. A backwater valve should be used to protect such drains where backwater is expected. Underground sanitary drains beneath buildings should be of cast-iron soil pipe, not less than 4-in. in diameter.

Minimum safety and health requirements for the design, installation, inspection, and performance of plumbing equipment and systems, including water supply, distribution, drainage, and venting systems have been formulated by ASA Sectional Committee

A40 for presentation as an American Standard Plumbing Code.[12] Dimensional standards and materials are not included therein since they are provided in other American Standards and ASTM Specifications. Reference should be made to the latest issue of the ASA code and to local plumbing codes of the various states and municipalities.

SEWERS AND SEPTIC TANKS

When 5 ft outside the building wall, the house drain becomes a sewer and may be constructed of vitrified crock or other suitable material. In order to avoid trouble with tree roots and similar obstructions entering at cemented joints, cast-iron soil pipe with leaded joints often is used between the building and the public sewer. The introduction of slip-seal and other bitumastic gaskets for use in making up the joints of crock sewers has largely overcome the difficulty with tree roots so that crock sewers can be used now with reasonable assurance of satisfactory performance. A house sewer may lead to either a public sewer, a septic tank, or a vault.

Public Sewers.—The design of public sewers, sewage-treatment plants, and associated equipment is a specialized field in which interest has been stimulated recently through the need for more adequate sewage-disposal facilities commensurate with modern standards for sanitation. Those interested in this subject should read the trade journals or consult one of the excellent textbooks available in this field.[10] See page 1063 for computing Earth Loads on Pipe in Trenches.

Carrying Capacity.—The carrying capacities of sewer pipes with *clean water* are given in Table XIII. Where the velocity is sufficiently high so that the flow is turbulent, the friction loss in pipes carrying sludges may be computed the same as for water at the corresponding temperature using the same flow formulas. For denser sludges involving laminar or plastic flow which occurs at velocities lower than for turbulent flow, a determination of the yield value and coefficient of rigidity of the sludge or sewage must be made.[11] Data on flow through corrugated metal and other types of culvert pipes will be found on pages 277 and 283.

The Kutter and Manning formulas[10] (see pages 283 to 288) are generally used for computing the carrying capacity of sewers since they are particularly applicable to gravity flow in open channels and in buried conduits that run only partly full.

TABLE XIII.—CARRYING CAPACITY OF SEWER PIPE
(Gallons per minute)

Size of pipe, inches	Fall per 100 ft.							
	1 in.	2 in.	3 in.	6 in.	9 in.	1 ft.	2 ft.	3 ft.
3	13	19	23	32	40	46	64	79
4	27	38	47	66	81	93	131	163
6	75	105	129	183	224	258	364	450
8	153	216	265	375	460	527	750	923
9	205	290	355	503	617	712	1,006	1,240
10	267	378	463	655	803	926	1,310	1,613
12	422	596	730	1,033	1,273	1,468	2,076	2,554
15	740	1,021	1,282	1,818	2,224	2,464	3,617	4,467
18	1,168	1,651	2,022	2,860	3,508	4,045	5,704	7,047
24	2,396	3,387	4,155	5,874	7,202	8,303	11,744	14,466
27	4,407	6,211	7,674	10,883	13,257	15,344	21,771	26,622
30	5,906	8,352	10,223	14,298	17,714	20,204	28,129	35,513
36	9,707	13,769	16,816	23,763	29,284	33,722	47,523	58,406

Septic Tanks.—In suburban and rural districts not served by public sanitary sewers, the proper disposal of liquid wastes from toilet fixtures is best accomplished through the use of septic tanks discharging to underground absorption systems. By this method the wastes are disposed of with a minimum danger of polluting nearby wells and in such a way that flies, vermin, fowls, and domestic animals cannot get in contact with infectious material. The purification of sewage by bacteria working in a septic tank, followed by distribution through tiles for absorption in the top soil is a natural process through which complex organic wastes are rapidly broken down and purified. The need for such sanitation, the design of septic tanks and disposal systems, and the principles on which they operate are discussed in numerous books, technical articles, and bulletins (see the following Bibliography).

In localities where there is considerable underlying gravel, septic tanks sometimes are dispensed with and sewage discharged, instead, into a vault where it seeps off through the gravel. This is a dangerous practice which may tend to pollute the water supply and should be avoided wherever possible.

BIBLIOGRAPHY

Specific References

[1] For detailed discussion of cross-connections, see (*a*) "Practical Aspects of Cross-connection, Interconnection, and Back-flow Protection," by R. F. Goudey, *Jour. AWWA*, March, 1941, pp. 391–394.

(b) "Cross-connections in Plumbing and Water-supply Systems," by A. A. Kalinske, Wisconsin State Board of Health (rev. 1941).

(c) "A Simple Method for Location of Cross-connections in Piping Systems," by R. C. Doke, *Jour. AWWA*, 1940, Vol. 32, pp. 1997–2005.

[2] See "New Plumbing Standards to Keep Drinking Water Pure," by A. A. Kalinske and F. M. Dawson, *Ind. Standardization*, August, 1943.

[3] See "Water-distributing Systems for Buildings," by Roy B. Hunter, U.S. Department of Commerce, Report on Building Materials and Structures, Report BMS79. For sale by Superintendent of Documents, Washington, D.C.

[4] For data on carrying capacity of copper water tubing, see "Hydraulic Service Characteristics of Small Metallic Pipes," by Fair, Whipple, and Hsiao, *J. New Eng. Water Works Assoc.*, December, 1940, Vol. XVIV, No. 4.

[5] For rates of discharge from faucets, see "Tests on the Hydraulics and Pneumatics of House Plumbing," by H. E. Babbitt, *Bull.* 178, University of Illinois, Engineering Experiment Station.

[6] From a report prepared by F. M. Dawson and A. A. Kalinske of the State University of Iowa for the National Association of Master Plumbers, 1936.

[7] See (a) "Requirements for Pipe-thawing by the Electrical Method," by P. C. Ziemke, *Coal Age*, September, 1940, p. 72.

(b) "Thawing Service Pipes," by F. C. Amsbary, Jr., *Jour. AWWA*, July, 1936, pp. 856–867.

(c) "Thawing Frozen Pipes by Electricity," by L. A. Ferney, *Water and Water Works Engineering* (British), November, 1942, Vol. 43, p. 310; also abstracted in *Jour. AWWA*, November, 1942, Vol. 34, p. 1714.

(d) For resistance to bursting due to freezing, see "Copper and Brass Pipes and Tubes," by Wm. C. Schneider, *Jour. AWWA*, vol. 23, no. 7, pp. 984–985, July, 1931.

[8] See "Improved Hot-water Supply Piping," by J. M. Krappe, *Research Series*, 64, Engineering Experiment Station, Purdue Univ., January, 1939. Also subcommittee report on "Copper Tubing for Hot-water Piping Systems," prepared by the Water Heating Subcommittee on Copper Tubing for Hot-water Piping Systems, *AGA Proc.*, 1935, pp. 345–51.

[9] From "Plumbing Manual," Report BMS66, Building Materials and Structures, U.S. Department of Commerce, available from Superintendent of Documents, Washington, D.C.

[10] See, for instance, "American Sewerage Practice," by Leonard Metcalf and Harrison P. Eddy, Vol. 1, "Design of Sewers," McGraw-Hill Book Company, Inc., New York, 1928.

[11] See "Flow of Solids in Piping," by H. E. Babbitt and D. H. Caldwell, *Heating, Piping and Air Conditioning*, July, 1942, pp. 423–427, and August, 1942, pp. 491–494. Also, "Laminar Flow of Sludge in Pipes with Special Reference to Sewage Sludge," *Bull.* 319, University of Illinois, Engineering Experiment Station, and "Turbulent Flow of Sludges in Pipes," by the same authors, *Bull.* 323, University of Illinois Engineering Experiment Station.

[12] See "Proposed American Standard Plumbing Code ASA A40 (Minimum Requirements for Plumbing)." Copies may be obtained from the American Standards Association, 29 West 39th St., New York 18, N.Y.

General References

"Plumbers' Handbook," by Samual Edward Dibble, McGraw-Hill Book Company, Inc., New York, 1922.

"Plumbing Engineering," by Walter S. L. Cleverdon, Pitman Publishing Corporation, New York, 1937.

"Sewage Disposal for Single Houses and Small Institutions," *Eng. Bull.* 2, Michigan Department of Health, Lansing, Mich.

"The Septic Tank System for Home Sewage Disposal," *Eng. Bull.* 18, University of Kansas, Lawrence, Kan.

"Heating, Ventilating and Air Conditioning Guide," ASHVE, 51 Madison Ave., New York 10, N.Y.

"Handbook," National District Heating Association, 827 N. Euclid Ave., Pittsburgh 6, Pa.

"American Standard Cast-iron Soil Pipe and Fittings," ASA A40.1 (abstracted pp. 450–453).

"American Standard Cast-iron Screwed Drainage Fittings," abstracted, pp. 457–460.

"General Engineering Handbook," by C. E. O'Rourke, McGraw-Hill Book Company, Inc., New York, 1932. Particularly with reference to sewers.

"Some Structural Requirements for Sanitary Wells," *The Driller*, January and February, 1943. Also, abstracted in *Jour. AWWA*, July, 1943.

"American Standard for Drinking Fountains," ASA Z4.2-1935.

CHAPTER XI

UNDERGROUND STEAM PIPING

By J. H. Walker[1]

This chapter deals with underground steam piping as used in district-heating systems and for groups of buildings such as institutions and industrial plants. Assuming that normal construction will involve the use of steel pipe with welded joints in so far as practicable, seven major items remain to be settled in the design of district heating and underground steam piping, *viz.:* (1) pipe size, (2) wall thickness and materials, (3) proper insulation, (4) protection of the pipe and its insulation from water and from mechanical damage, (5) drainage of condensation, (6) provision for thermal expansion, with controlling anchorage, and (7) safety precautions.

The treatment given these subjects in this chapter, plus information at other places throughout this Handbook, will be found as follows:

1. For selection of pipe sizes with respect to pressure drop, see formulas and charts on pages 81 to 137 and 246 to 264; for reasonable steam velocities see pages 864 to 866.
2. For formulas for determining the required wall thickness and material with respect to steam pressure and other conditions, see pages 31 to 49.
3. For insulating materials, see pages 710 to 718 with a table of Typical Selection of Insulation for Underground Steam Distribution Piping on page 719. Owing to special conditions affecting heat loss from buried steam pipes, this subject is dealt with on pages 1016–1018 of this chapter.
4. For subsoil drainage and protection from mechanical damage, see pages 1009 to 1014.
5. For draining condensation and sizing condensate lines, see pages 1014 and 1015.

[1] Honorary Member, National District Heating Association; Member, ASHVE, ASME; Superintendent of Central Heating, The Detroit Edison Company.

6. For general data on thermal elongation, expansion fittings, etc., see Chap. VII, pages 754 to 860. Further comments pertaining to underground steam piping will be found on page 1015.

7. Safety requirements abstracted from Sec. 4 on District Heating Systems of the American Standard Code for Pressure Piping are given on page 1005 and elsewhere throughout this chapter. Attention is directed to the fact that Sec. 4 does not cover piping at pressures of 15 psi or lower.

Pressure.—The pressure at which steam is supplied to the distribution network will depend on the boiler pressure at which the steam is generated and whether it is first passed through electrical generating equipment to make use of available energy. Low-pressure distribution up to about 30 psi is desirable from the standpoint of lower investment cost through using lighter weight pipe and fittings, lower heat loss from the pipes, less expensive pressure-reducing valves, and general reduction in maintenance costs through less trouble with traps, valves, etc. The lesser hazard involved in the lower pressures is an added inducement. The advantages of high-pressure distribution (100 to 175 psi) are smaller pipe sizes and fittings, and the greater adaptability of steam to various uses other than building heating.

Pipe Size.—In determining size of pipe for a specific installation, several conditions must be either known or assumed, *i.e.*, the initial pressure and the minimum permissible terminal pressure are specified, the steam quantity necessary is known, and the length of line including equivalent lengths for elbows, tees, valves, etc., is obtainable. The pipe size required to meet these conditions may be calculated readily then by means of one of the pressure-drop formulas (see pages 246–254).

Within the limits imposed by acoustic velocity (see page 254) this method practically disregards steam velocity, which may reach 50,000 to 75,000 ft per min if the allowable pressure drop permits. High velocities are not objectionable in underground piping. Noise usually is not a factor and there is little danger of erosion. By thus taking full advantage of the available pressure drop the maximum economy of pipe sizes is obtained.

In most district-heating systems, there are one or more trunk mains of relatively large diameter, with branch mains connected at intervals. If the area served is large, the pipes often form a grid or network. In some cases, in addition to the original trunk mains, high-velocity feeder pipes are installed, as and where required, to

carry steam directly from the boiler plant to strategic points in the distribution network.

In systems serving institutions where there are no city streets to fix the route of the pipes, they can be laid out so as to reach the buildings by the most economical route. A loop system is a good idea where practicable.

Where grid or network systems are employed, computation of pipe size, pressure loss, and quantities of steam flowing becomes complex. It is customary to use a cut-and-try method in which a point of zero flow, or balance point, is assumed, and flow quantities, pressure drops, etc., are computed on this basis for a given system. If this does not work out, a different point is then assumed and the problem recomputed as often as necessary to secure the correct answer. For further information on this procedure as applied to flow in networks, see pages 90–95, 196–198.

Pipe Thickness.—Where the pressure does not exceed 250 psi, Schedule 40 pipe in accordance with the American Standard for Wrought Iron and Wrought Steel Pipe, ASA B36.10,[1] will be adequate for any service, using screwed, flanged, or welded joints. Where pipe is threaded and used for steam pressures of 250 psi or greater, or for water pressures in excess of 100 psi at temperatures of 220 F or over, Schedule 80 pipe is required by the American Standard Code for Pressure Piping, ASA B31,[1] in order to furnish added mechanical strength. Where welded joints are used and corrosion is not an important consideration, Schedule 40 (standard-weight) low-carbon steel pipe may be used for steam pressures up to approximately 400 psi. For lower pressures under the same conditions, Schedule 30 or even Schedule 20 pipe may be acceptable. Schedule 80 (extra-strong) low-carbon steel pipe may be used for saturated steam pressures up to about 800 psi, either screwed or welded.

For an accurate determination of the proper pipe-wall thickness, however, the formulas prescribed in Sec. 4 on District Heating Piping of the Code for Pressure Piping should be used. The formulas are the same as for power piping given on pages 42–47.

Dimensional Standards and Materials.—In general, dimensional standards and materials should conform to American Standards and ASTM materials specifications for the service conditions obtaining in the line. Appropriate choice of materials and

[1] Copies can be purchased from the American Standards Association, 29 West 39th St., New York 18, N.Y.

standards from a safety standpoint is furnished in the District Heating Section of the American Standard Code for Pressure Piping, ASA B31.1. Dimensional standards and material specifications referred to are abstracted in Chap. IV, pages 348 to 693.

The specific requirements for underground steam piping will vary depending on the pressure and temperature of the steam conveyed, on individual preferences of the designer, and on the type of service involved. For instance, the several types of steam piping for underground district heating may be segregated into the following classes: tunnel mains, tunnel feeders, surface mains, surface feeders, and customer's services. Typical selections of materials and dimensional standards of one utility for these classes as applied to a system where saturated steam is conveyed at a pressure of less than 125 psi are given in Table I. In all cases, compliance with the ASA Code for Pressure Piping requirements obtains. General requirements for piping are as follows:

(a) *Pipe.*—Steel pipe, either seamless or welded, ordinarily is used although the Piping Code permits a variety of materials and several types of welded pipe such as lap-welded, electric-resistance welded, etc. Where pipe is desired for bending, low-carbon seamless steel pipe is recommended.

(b) *Screwed and Welded Joints.*—As in other piping fields, screwed and flanged joints have been displaced to a large extent by welded construction, both in main steam lines and customer's service connections. Steam and condensate lines 3-in. nominal pipe size and smaller sometimes are screwed although sizes 4 in. and larger nearly always are welded. For pressures 400 to 600 psi, screwed fittings in sizes larger than 2 in. are not permitted by the Code for Pressure Piping. In customer's service connections, mitered bends may be used instead of ready-made elbows where the extra pressure drop of the former is of little significance. Both gas and arc welding methods are employed depending on the conditions of use, and both produce satisfactory welds under qualified procedure and with qualified operators. Rules for the qualification of procedures and operators are contained in the American Standard Code for Pressure Piping, Section 6, on Fabrication Details, which is abstracted on pages 494–496.

(c) *Flanged Joints.*—While welding has largely displaced flanged joints, some flanged connections are required, particularly in making connections to flanged valves, expansion joints, or fittings where space limitations do not permit welding, or where easy removal of a fitting or valve is desired. Cast-iron, malleable-iron, or bronze flanged fittings are acceptable for the lower pressures and temperatures, whereas cast or forged steel flanged fittings are required for pressures in excess of 250 psi and temperatures in excess of 450 to 500 F.

(d) *Valves.*—Valves in accordance with the manufacturer's standard for 125 psi are considered acceptable by the Piping Code for 125 psi service and should have cast-iron, malleable-iron, steel, or bronze bodies, discs, bonnets, and yokes. Solid-wedge gate valves with inside screw and non-rising stems are used by one utility. For pressures in excess of 250 psi, all valves larger than 3 in. nominal size must have flanged or welding ends. Welding of steel valves directly

TABLE I.—TYPICAL SELECTION OF DIMENSIONAL STANDARDS AND MATERIALS FOR UNDERGROUND STEAM PIPING[1]

	Operating pressure, psi	Joints	Pipe	Flanges	Fittings[6]	Valves
Customer's services	28 to 35	3—[2] screwed 4+ welded	Schedule 40 ASTM A53	3— screwed 125 lb CI 4+ 150-lb steel welding-neck	4— 250-lb CI screwed 6+[5] 125-lb CI flanged	2— all brass 2½+ 125-lb CI flanged
Tunnel mains	28 to 35	Welded[3,4] or lapped	Schedule 40 ASTM A53	300-lb steel lapped or welding-neck	8— 250-lb CI flanged 10+ 300-lb steel flanged	8— 250-lb CI flanged 10+ 300-lb steel flanged
Tunnel feeders	125 max	Welded[3,4] or lapped	Schedule 40 ASTM A53	300-lb steel lapped or welding-neck	8— 250-lb CI flanged 10+ 300-lb steel flanged	8— 250-lb CI flanged 10+ 300-lb flanged steel
Surface mains	28 to 35	Welded[3]	Schedule 40 ASTM A53	150-lb steel welding-neck	125-lb CI flanged	125-lb CI flanged
Surface feeders	125 max	Welded[3]	Schedule 40 ASTM A53	300-lb steel welding-neck	250-lb CI flanged	250-lb CI flanged

BOLTING: Hexagon-head carbon-steel bolts and hexagon nuts, Class I of ASTM A194 used with cast-iron flanges; bolt studs, Class A of ASTM A96, nuts, Class I of ASTM A194 used with steel flanges. Carbon-steel bolt heads and nuts are in accordance with the heavy series of American Standard ASA B18.2.

GASKETS: 1/16-in. cross-laminated asbestos composition gaskets used.

[1] These selections conform to the requirements of the American Standard Code for Pressure Piping, ASA B31.1-1942.

[2] 3— is used to indicate sizes 3 in. and smaller, 4+ to indicate sizes 4 in. and larger, same code for other sizes as noted.

[3] Welding-neck flanges used to connect to flanged fittings or valves.

[4] Welded construction acceptable, provided backing rings are used.

[5] Welding fittings may be used sizes 4 in. and larger.

[6] For welded construction, welding fittings may be used.

into the lines as is common practice for power plant piping is not considered so favorably in underground piping because: (1) of space limitations when it is necessary to remove the valve, and (2) removal of a welding-end valve might incapacitate a line or system until the valve could be repaired or replaced. Special requirements regarding stop valves are provided in the District Heating Section of the Piping Code to insure free drainage of condensate along the bottom of the pipe. These permit the use of globe and plug valves as well as gate valves provided they are constructed and installed so as not to obstruct the flow of condensate, and angle valves may be used in positions for which they are suitable. Special requirements pertaining to reducing and relief valves are included in a subsequent paragraph of this chapter.

(*e*) *Fittings.*—Screwed or flanged fittings of cast-iron, malleable-iron, or steel are acceptable and should be used in accordance with the requirements designated in the Piping Code. For welded lines, welding fittings in accordance with the American Standard for Steel Butt-welding Fittings, ASA B16.9, or the proposed American Standard for steel socket-welding fittings are used. Special fittings, headers, or welded assemblies are acceptable if made in accordance with the requirements of Section 6 of the Code for Pressure Piping.

(*f*) *Bolting.*—To avoid breakage of cast-iron flanges, flanged fittings, or flanges on equipment, bolts in cast-iron flanged joints, or in case a cast-iron flange is matched with a steel flange, must be carbon steel as specified by the Piping Code. Threads in accordance with the coarse-thread series of the American Standard for Screw Threads, ASA B1.1, are ordinarily used for carbon-steel bolts. With steel flanges, alloy steel bolting conforming to ASTM Specifition A96 should be used to insure maintaining a tight joint. Regular-series square nuts or heavy-series hexagon nuts in accordance with the American Standard for Wrench Head Bolts and Nuts and Wrench Openings, ASA B18.2, are required by the Code for Pressure Piping. Materials for nuts should conform to ASTM Specification A194. For alloy-steel bolting, or carbon-steel bolting for high-temperature service, threads should be in accordance with the American Standard for Screw Threads for High-strength Bolting, ASA B1.4.

(*g*) *Gaskets.*—For medium- to low-pressure steam service, asbestos gaskets have been found to give good service, although metal or other material which will not burn, char, or change so that it will not perform the service intended may be used. Rubber gaskets may be used for hot-water service or condensate lines at temperatures not in excess of 250 F.

Testing after Installation.—Where underground piping is to be buried or otherwise made inaccessible, it is customary practice to test the lines hydrostatically to ascertain if any leaks exist. The Code for Pressure Piping requires, where practicable, a test of one and one-half times the maximum allowable service pressure to be held for a period of at least 2 hr without evidence of leakage. When a hydrostatic test is impractical, it is required that the piping be tested with steam at a pressure at least equal to the pressure at which the piping is to be operated. These tests may be made on sections, or on the whole of the piping system, but the connections between the sections must be similarly tested.

Pipe Protection.—The pipe should be enclosed in a *conduit* from which it is separated so as to allow free longitudinal expansion and which is strong enough to protect against earth pressure. The insulation may be attached to the pipe or to the inner surface of the conduit. The most suitable depth below the surface depends upon the nature of the installation. In industrial work with level ground, 2 or 3 ft is sufficient, but in district-heating work the average depth is usually about 6 ft to the center line of the pipe. Fewer obstructions are encountered at the lower levels. The pipe should consist of a series of perfectly straight sections between anchors and expansion joints.

The design of the conduit will depend upon the nature of the service and upon soil conditions. For a temporary installation or where cheapness is essential, a rough box of planks can be used as a conduit, but this construction is very short lived.

Wood casing construction, in which wood is the insulating material, has been widely used in the past. The casing is of segmental construction, tin lined, bound with wire, and coated with asphaltum. In a very well drained soil this is a satisfactory conduit, and has a life averaging, in some cases, 15 to 20 years. In any but the best drained soils, however, the wood rots rather quickly and allows moisture to reach and attack the pipe itself. Wood construction also has a tendency to char with steam pressures above about 75 psi.

Several designs of underground conduit construction are shown in Fig. 1. Design (*a*) is particularly useful for wet soil. If properly installed it is practically watertight. It is not, however, a common design.

Design (*b*) is used by several district-heating companies with various slight modifications. It is constructed in two stages, the bottom being poured first. After the pipe is installed, the sides and top are poured over the arched, corrugated steel form. This design is rugged, simple, and has the advantage of being constructed of common materials.

Design (*c*) is a proprietary article, featured by the use of "cell concrete," *i.e.*, a concrete which is made porous by the addition of a gas-forming chemical. The cell concrete is the insulator and is poured around the pipe into a steel form which is later enclosed in ordinary concrete.

Design (*d*) is quite similar to design (*b*). Multicell tile is used for the side walls. Many different designs of masonry, tile, and

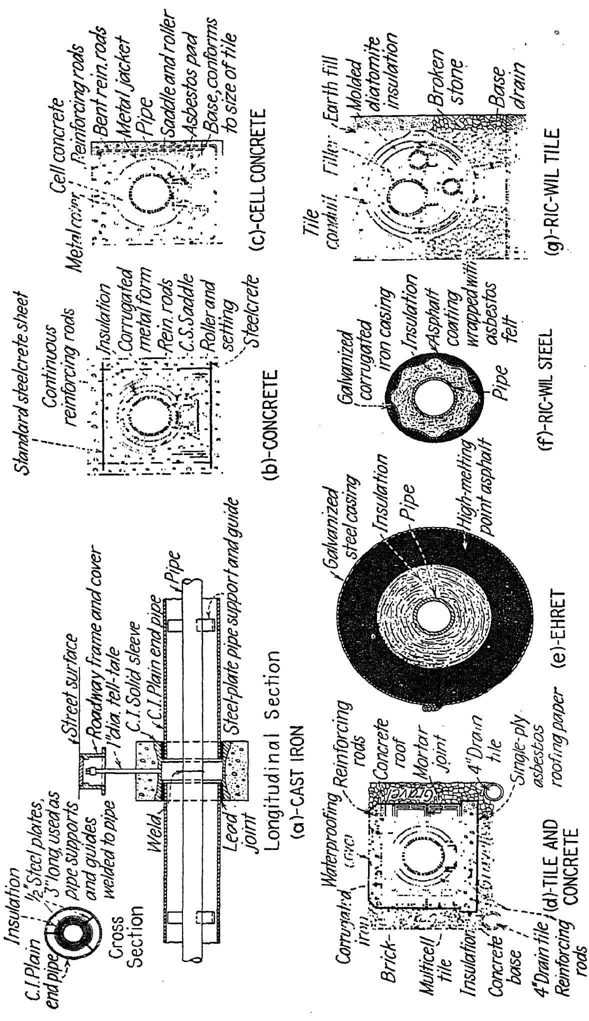


FIG. 1.—Various types of underground steam-conduit construction.

concrete conduits have been used in past years but most of them follow the general pattern of (*b*) and (*d*).

Designs (*e*), (*f*), and (*g*) are proprietary products. Design (*c*) consists of a layer of insulation surrounded by a thick layer of asphalt enclosed in a steel casing. The conduit is factory-assembled and the lengths are joined by welding the pipe and then applying the insulation and asphalt coating at the short section previously left bare for the welding.

Design (*f*) is also factory-assembled. It consists of a corrugated galvanized iron casing with a loose fill of insulating material around the pipe. The casing is asphalt-coated and wrapped.

Design (*g*) is a split ceramic tile which can be used for more than one pipe. A loose-fill insulation is packed around the pipes.

None of these forms of conduit could be called absolutely water-tight when actually submerged, even when laid with the greatest care. They are, however, suitable for wet soils, when properly underdrained. Since the effectiveness of insulation is reduced materially when it becomes wet, it is important that the pipe and its insulation be enclosed in a waterproof jacket. Tile or concrete conduits are usually designed to provide an internal drainage area to carry away any water that may collect on the inside of the conduit from a leaky pipe or joint, or from seepage through the conduit. The water may be carried to a manhole or sump having an automatic pump, or directly to a sewer. When a steam line is to be actually submerged at intervals, the only practicable conduit is one built of cast iron with calked joints. Even that construction is difficult to make tight.

Where conditions are favorable, pipe may be inserted in a hole bored through the earth without digging a conduit trench, as described on pages 1184–1186 of the Handbook. Because of the size of pipes used and the possibility of damaging other buried piping or conduits in business areas, this method is employed infrequently for underground steam piping.

Tunnels.—Steam-distribution and condensate pipes sometimes are housed in a tunnel. The tunnel system has many advantages particularly where cost can be kept down by cut-and-cover construction rather than actual tunneling. Several steam lines, condensate lines, perhaps telephone and electric cables, and other pipes may be installed in the same tunnel where all are readily available for inspection and repair. Figure 2 shows one of the more expensive types of tunnels, designed to withstand heavy earth pressure

at depths from 25 to 60 ft below the surface. In general, the use of tunnels is warranted only if two or more pipes are to be installed in

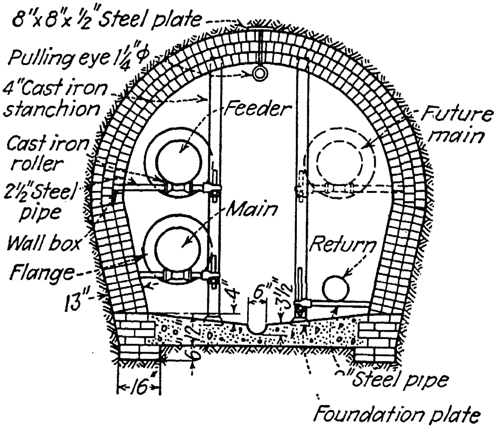


FIG. 2.—Typical cross section of a deep tunnel where heavy earth loads have to be supported.

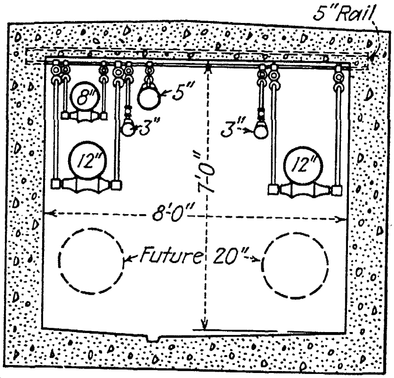


FIG. 3.—Typical cross section of a shallow tunnel—usually built by cut-and-fill methods.

the same street or where subsurface conditions or obstructions are not favorable for the conduit type of construction.

For tunnels that are located less than 25 ft below the surface, a less expensive type of construction is employed, such as shown in

Fig. 3. Whereas it is customary to reinforce the top of the tunnel to withstand external loads as well as to provide adequate support for the pipe lines, the sides and bottom of the tunnel may be of ordinary concrete construction. This type of tunnel is used extensively to serve groups of institutional buildings.

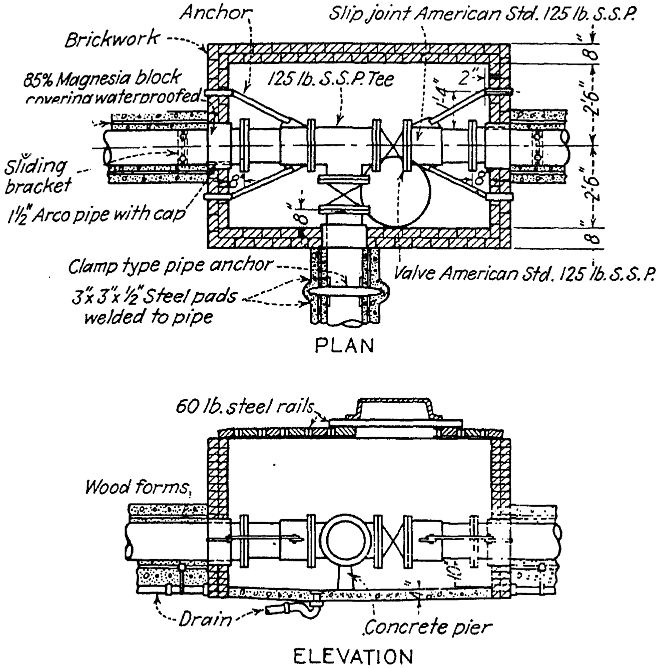


Fig. 4.—Typical manhole for buried construction.

Manholes.—With buried construction, manholes are required to house valves, traps, and some kinds of expansion joints. They are built of brick or concrete and should be at least 4 by 4 ft with a depth depending upon the depth of the pipe. They should be drained into the conduit drain tile. Typical manhole construction is shown in Fig. 4.

Underdrainage.—With any form of conduit, unless it is actually below water level, provision should be made to carry away ground water and prevent it from collecting around the conduit. A layer

of coarse gravel or stone is placed below and partly surrounding the conduit and below this are laid one or two lines of tile, with open joints, which serve to carry off the water to a sewer or other outlet. Underdrainage with an adequate outlet is a very necessary provision.

Drainage of Condensation.—Provision must be made for removing condensed steam from the mains. Pipe should be carefully graded during erection and the pitch should be not less than 1 in. in 50 ft, and preferably in the direction of steam flow. The condensate should be drained off through traps, at intervals not greater than 300 to 400 ft. Since it usually is impracticable to provide a continuous slope, steam traps are installed at all low points where a water pocket exists. The contour of the ground often will dictate the pitch. In some cases, traps can be discharged directly to the sewer. In other instances, it may be necessary to run discharge lines to sump pits in manholes where the water is removed by float-controlled electric pumps and discharged into the city sewer.

Where feasible, it is desirable to return steam condensation to the boiler plant to reclaim the heat in the condensate, to save the equivalent fresh water, and to avoid the necessity of chemically treating the latter. In general, it is not economical to return condensate in district-heating systems because of the high cost of the return piping and difficulties in pumping but, in systems serving groups of institutional buildings, condensate return often is desirable. Where the condensation can be returned by gravity, sizes of the return pipes may be obtained from Table II.

Extensive requirements regarding drains, drips, and steam traps for underground steam distribution lines operating at pressures in excess of 15 psi, are contained in Par. 418 of the Code for Pressure Piping, the more important of which are as follows:

418(*a*) Drains or drips shall be provided to drain the condensate from the steam piping and equipment wherever it may collect. Blowoff outlets for air or condensate open to atmosphere, or connected to sewers, sumps, or receivers, shall be provided at all low points and elsewhere when necessary for the proper operation of the pipe line and equipment. Each drip, drain, and blowoff shall be controllable by at least one stop valve, located as close as practicable to the point of drainage.

(*b*) Drip lines from steam headers, mains, separators, and other equipment shall be properly trapped with the traps installed in accessible locations. By-passes shall be provided around steam traps unless the traps may be replaced by a spare, or drainage continued by means of an open drip to the atmosphere or elsewhere at times when the trap is serviced or found inoperative.

Table II.—Capacity of Gravity Returns for Underground Distribution Systems in Pounds of Condensate per Hour

Nominal pipe size,[a] inches	Pitch of pipe per 100 ft						
	6 inches	1 foot	2 feet	3 feet	5 feet	10 feet	20 feet
1	448	998	1,890	2,240	3,490	5,490	7,490
1¼	1,740	2,490	3,990	4,880	6,480	9,480	13,500
1½	2,700	4,190	5,740	7,480	9,480	14,500	20,900
2	4,980	7,380	10,700	13,900	16,900	24,900	36,900
3	13,900	22,500	30,900	37,400	50,400	74,800	105,000
4	30,900	44,800	64,800	79,700	105,000	154,000	229,000
5	54,800	79,800	120,000	144,800	195,000	294,000	418,000
6	90,000	138,000	187,000	237,000	312,000	449,000	
8	190,000	277,000	404,000	508,000	660,000	938,000	
10	344,000	498,000	724,000	900,000	1,190,000		
12	555,000	798,000	1,148,000	1,499,000	1,990,000		

[a] Size of pipe should be increased if it carries any steam.
Reproduced by permission from "Heating, Ventilating and Air Conditioning Guide," 1943.

(c) Drip lines from steam headers, mains, separators, or other equipment operating at different pressures, shall not be connected to discharge through the same trap. Where several traps discharge into one header under pressure, or which may be under pressure, a stop valve and a check valve shall be placed in the discharge line from each trap.

(e) The thickness of trap discharge piping shall be the same as the inlet piping unless the former is vented to atmosphere or operated under low pressure and has no stop valves. The trap discharge piping in all cases shall be of a weight suitable for the maximum discharge pressure to which it may be subjected. Trap discharge piping, if vented to atmosphere, shall be properly run to facilitate its self-discharge and its outlet shall be so located, or proper protection shall be provided, to prevent human injury caused by escaping steam or hot water. All trap discharge piping shall be protected against freezing, where necessary.

Expansion Fittings.—Linear expansion in underground work occasionally can be taken care of by the bending of parts of the pipe line, as is the case in power-plant piping. This is particularly true where the pipe runs from one building into the basement of another not more than 200 or 300 ft away. The line may then be anchored in the middle and allowed to expand in both directions. Usually, however, the movement must be absorbed by some kind of expansion fitting.

The various kinds of expansion fittings are described in Chap. VII, pages 764 to 771. For underground work, the slip joint is very frequently used. It must be placed in a manhole or otherwise made accessible, however, and occasionally requires repacking. The diaphragm or corrugated types of expansion fittings do not

have these objections and can be buried, but they must be placed at more frequent intervals.

Anchorage.—It is always necessary to anchor the pipe midway between the expansion fittings to control the movement. This is done by a clamp of some sort around the pipe or by means of a fitting with an anchor base. Branch connections preferably are made at or near the anchor points. If it is necessary to connect a branch at a point where there is considerable expansion movement, ample clearance must be provided around the branch pipe.

Heat Loss.[1]—The phenomenon of heat loss from a pipe buried in soil is somewhat different from the case of a pipe in air because of the insulating effect of the soil. The conductivity of the soil varies considerably, depending upon its composition and dryness, particularly the latter. The heat loss in wet ground will be greater (sometimes by as much as 200 per cent) than that from the same conduit in dry soil.

The heat loss from a pipe buried in the ground is not proportional to the external surface of the pipe. The heat loss per square foot of surface is much larger for small pipes than for large pipes.

The depth to which a covered pipe is buried in the soil makes very little difference in the heat loss, provided the center of the pipe is 2 ft or more below the surface of the ground. Beyond 2 ft in depth, unless pipes are very large, the heat loss remains substantially the same at all depths.

The economical thickness of insulation is difficult to determine exactly, except by actual test in existing soil conditions. The first layers of insulation are much more effective than succeeding layers and the economical thickness easily may be exceeded. If the increasing of the insulation requires enlarging the conduit and widening the trench which must be dug to bury it, then the thicker covering will almost certainly not be economical.

The results of several tests, given in Table III, show the actual measured condensation loss from several installations of underground pipes and the corresponding computed heat losses.

Heat loss from steam pipes located in tunnels may be computed essentially the same as for steam pipes in air (see pages 694 to 709), the important variable being the temperature of the air in the tunnel. The latter will depend on the extent to which the tunnel is ventilated, but in most cases it will be at least 85 F and some-

[1] For detailed treatment of this subject, see "Theory of Heat Losses from Pipes Buried in the Ground," by John R. Allen, *Proc. ASHVE*, 1920.

times as high as 120 F. Since the heat loss will vary considerably depending on the velocity of air over the pipes, this factor should be considered. Figure 9 on page 708 shows the effect of increase in air velocity on the heat loss from both bare and properly insulated surfaces, the bare surface being maintained at a temperature of 400 F. It may be observed that the effect of air velocity is much less pronounced with insulated surfaces.

TABLE III.—SUMMARY OF TESTS OF HEAT LOSS FROM UNDERGROUND PIPES

Test	*A*	*A*	*B*	*B*	*C*	*D*
Size of pipe	6 to 12 in.	6 to 10 in.	12 in.	8 in.	8 in.	8 in.
Length of pipe, ft[1]	....	....	301.5	681.25	576	224
Duration of test	8 months	8 months	2,376 hr	2,376 hr	8 months	250 hr
Average steam pressure, lb gauge	5	25	38.6	38.6	26.4	35
Character of soil	Dry clay	Dry clay	Dry clay	Dry clay	Dry clay	Dry clay
Type of conduit	Wood casing 81 per cent Concrete[2] 19 per cent	Wood casing 90 per cent Concrete[2] 10 per cent	Concrete[2]	Concrete[2]	Concrete[2]	Concrete[3]
Thickness of insulation, in.	1[4]	1[4]	1½	1	1	2
Condensation loss, lb per sq ft of pipe surface per hr	0.0511	0.0588	0.0755	0.0816	0.0429	0.0715
Btu loss per sq ft of pipe surface per hr	49.1	54.9	69.5	75.1	42.9	66.1

A Test made by The Detroit Edison Company 1913 to 1914.
B Test made by The Detroit Edison Company 1923.
C Test made by The Detroit Edison Company 1923 to 1924.
D Test made by The Detroit Edison Company 1931.
[1] [illegible] the branch service connections to buildings.
[2] [illegible] 1*b*.
[3] [illegible] and 4 in. of concrete buried directly in ground.
[4] [illegible] —no other insulation in wood casing.

A rough rule for the thickness of insulation for underground piping is as follows:

Low-pressure steam (up to 50 psi) in conduit using amosite preformed or laminated asbestos:

Pipe sizes 4 to 6 in., thickness 1⅛ in.

Pipe sizes 8 and 10 in., thickness 1¼ in.

Pipe sizes 12 to 16 in., thickness 1½ in.

High-pressure steam in tunnels 50 to 200 psi, using 85 per cent magnesia:

Pipe sizes 4 to 10 in., thickness 1½ in.

Pipe sizes 12 to 16 in., thickness 3 in.

For more specific data on insulation for underground piping, see Table VII, page 719.

Service Connections.—A service connection as customarily made on the consumer's premises is shown in Fig. 5. It is desirable to remove the heat from the condensation before it is wasted to the sewer. The best way to do this is in preheating the hot-water supply used in lavatories and thus reducing the amount of steam required for water heating. An approved method of installing a water-heating economizer in a gravity heating system is shown in

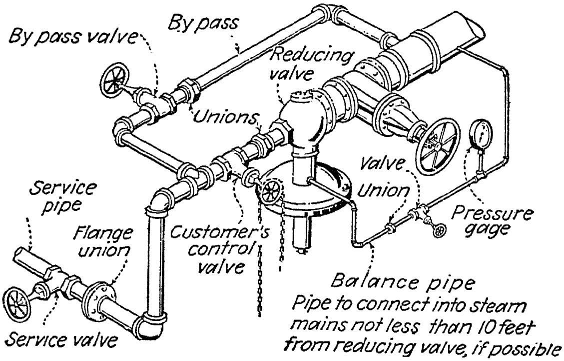


FIG. 5.—Connections for reducing valves of size 4 in. and larger, and for expanded valves.

Fig. 6. A simplified sketch illustrating a complete heating system served by district steam is shown in Fig. 7.

Reducing and Relief Valves.—Pressure-reducing and relief valves constitute an important item in district-heating piping since it is customary practice to distribute steam at street pressure which is reduced on entering the customer's premises for use in heating equipment designed, in general, for low-pressure service. Commercial heating equipment is now available, however, to withstand steam pressure up to 100 psi or above. Requirements regarding pressure-reducing and relief valves in consumer's premises for pressures in excess of 15 psi as contained in the Code for Pressure Piping are as follows:

408(*a*) Where the street pressure exceeds the safe working pressure of the building-heating apparatus, a pressure-regulating valve shall be provided near the point of supply to regulate the pressure on building heating equipment

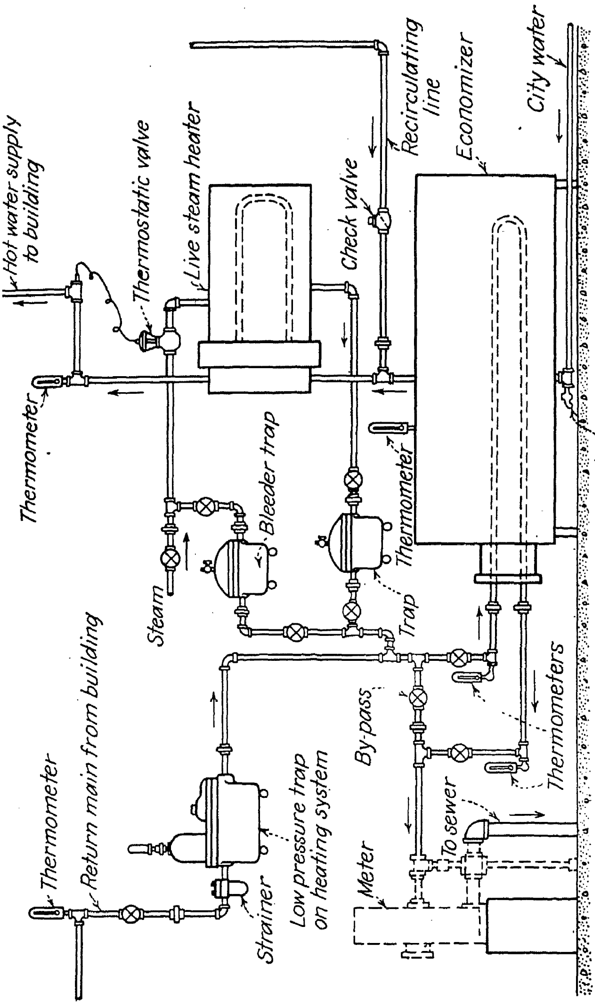


FIG. 6.—An approved method of installing a water-heating economizer in a gravity heating system.

within safe limits. In the case of cast-iron radiation this pressure shall not exceed 50 psi.

Where the street pressure exceeds 50 psi and is above the safe working pressure of the building steam-using apparatus, a relief valve or valves set at the safe

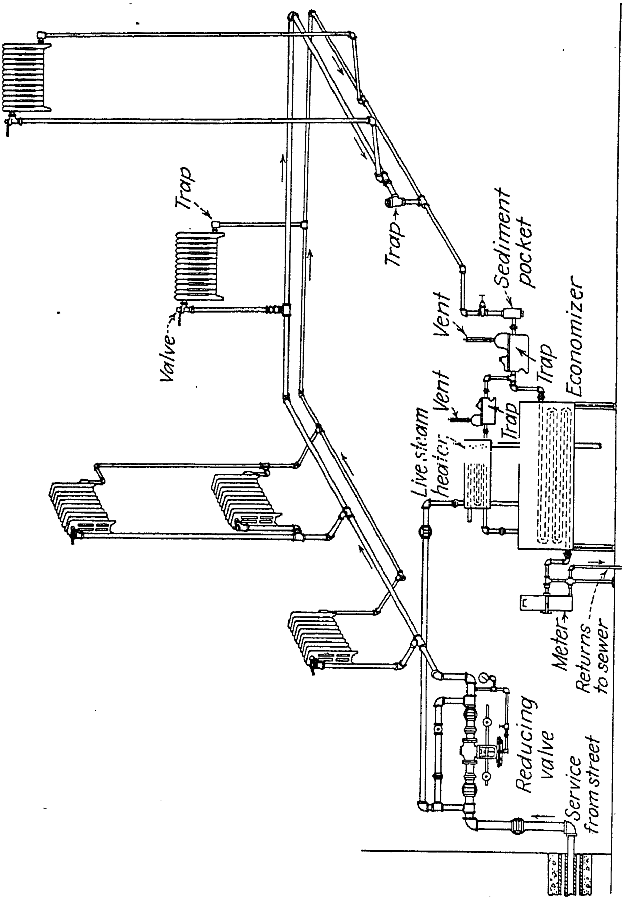


FIG. 7.—Building heating system using district steam.

working pressure of the building steam apparatus shall be provided, except that where two pressure-reducing valves are installed in series, both set at or below the safe working pressure of equipment served, no relief valve is required. If the installation of a relief valve is not feasible, a trip stop valve set to close at

the maximum safe working pressure shall be installed. When the building is continuously attended, an alarm valve or signal may be installed in lieu of a relief or trip stop valve.

(*b*) The capacity of relief valves shall be such that the pressure rating of the lower pressure piping and equipment shall not be exceeded with full steam flow from the relief valves. Relief valves shall be vented to the atmosphere, and proper protection shall be provided to prevent injury or damage caused by escaping steam.

(*c*) The use of a hand-controlled by-pass around a reducing valve is permissible. The by-pass shall not be greater in capacity than the reducing valve, unless the piping on the low-pressure side is adequately protected by relief valves or is of a construction which can withstand full street pressure.

(*d*) A pressure gage shall be installed on the low-pressure side of a reducing valve. Where two reducing valves are installed in series, the pressure gage shall be installed on the low-pressure side of the second reducing valve.

It should be noted that where two pressure-reducing valves are installed in series, both set at or below the safe working pressure of equipment served, no relief valve is required. In view of the difficulty of installing relief valves in the skyscraper type of building, the double reducing valves are considered to provide an acceptably safe installation. With distribution pressures not exceeding 40 to 50 psi, there is little hazard to building heating equipment even if full line pressure should build up in the radiators through faulty operation of a relief valve. For higher pressures, however, a second reducing valve or a relief valve on the low-pressure system is necessary.

Customers' Premises.—The installation rules of utility companies supplying steam differ widely, and before the piping in a building is laid out the designer should familiarize himself with local requirements. Also, the economical operation of the heating system is dependent upon proper piping design.

Costs.—Table IV gives the costs per foot for pipes in concrete conduit under paved city streets. These figures are for actual construction costs not including overhead or profit, and for labor rates and material costs prevailing in 1931–1934, where the work is performed under city conditions as to obstructions, paving, etc. For simpler conditions, these figures should be scaled down considerably, or for higher labor rates and materials costs, an upward adjustment should be made. The length of the line, its location, traffic conditions, and conditions as regards obstructions all are important items which will affect significantly the cost of the line.

The cost each of manholes and accompanying slip-expansion joints, fitting, insulation, etc., are given separately so that the

necessary manholes with trimmings can be computed as required. Manholes usually are installed for each 300 to 400 ft of underground pipe, or more frequently if special conditions require them. A good average figure for cost of manholes is $1.50 per cu ft.

TABLE IV.—COST PER FOOT OF UNDERGROUND STEAM PIPING[1]

	Pipe size, inches			
	8	10	12	16
Pipe, fittings, and insulation	2.19	3.18	4.10	4.51
Installing pipe and applying insulation	0.77	1.04	1.34	1.44
Excavation, backfill, and conduit construction	6.37	6.43	8.20	9.73
Inspection and paving	1.68	1.61	1.75	1.88
Engineering and superintendence	1.12	1.43	1.47	1.51
Total cost per foot	$ 12.13	$ 13.69	$ 16.86	$ 19.07
Cost each of manholes including expansion joint, fittings, covering, etc.	$575.00	$660.00	$743.00	$827.00

[1] 1931–1934 costs for pipes laid under city streets, using concrete conduit, such as illustrated in Fig. 1[illegible], and steel pipe and fittings for 150 psi welded construction.

UNIT PRICES, LABOR AND MATERIAL

Common labor	$0.60 per hr
Bricklayers	1.25 per hr
Pipe fitters	1.00 per hr
Welders	1.00 per hr
Cement	1.80 per bbl
Stone	1.84 per yd
Sand	1.60 per yd
Gravel	1.60 per yd
Pebbles	2.40 per yd

CHAPTER XII

WATER-SUPPLY PIPING

BY GEORGE H. FENKELL[1]

This chapter deals with water-supply piping for municipal, industrial, and irrigation purposes. Much of the information presented here and elsewhere in this book is applicable also to piping used in connection with hydraulic power and mining developments, and to sewage systems. Plumbing and fire-protection piping are treated in Chaps. X and XIII, respectively. The chemical composition and physical properties of water are given on pages 264 to 269.

HYDRAULICS

General flow formulas applicable to all fluids and suitable for use with friction factors based on Reynolds numbers (the *rational formula* system) will be found on pages 107 to 137. Empirical flow formulas commonly used by water-supply engineers and others interested in hydraulics are available on pages 269 to 289, together with tables and charts giving the carrying capacity and pressure drops for various diameters of pipe by the respective formulas. A discussion of these formulas as applied to the different branches of water supply piping will be found in the next section.

The following standard texts on hydraulics are listed here for the convenience of those wishing to refer to them: "Handbook of Hydraulics," by Horace Williams King; "Hydraulic Tables," by Gardiner S. Williams and Allen Hazen; "Hydraulics," by Joseph N. LeConte; "Hydraulics and Its Applications," by A. H. Gibson; "Hydraulics of Pipe Lines," by W. F. Durand; "Hydraulics," by E. W. Schoder and F. M. Dawson; "Treatise on Hydraulics," by Mansfield Merriman; "Hydraulics," by R. L. Dougherty; "Hydraulics," by H. J. Hughes and A. T. Safford; "Hydraulics,"

[1] Honorary Member ASCE and AWWA, formerly General Manager and Chief Engineer, Department of Water Supply, Detroit, Mich.

by H. W. King, C. O. Wisler, and J. G. Woodburn; "Applied Fluid Mechanics," by M. P. O'Brien and G. H. Hickox.

The economics of pipe size and water velocity as related to fixed charges and pumping costs have been analyzed in technical society papers and magazine articles.[1] Formulas for determining economical diameters of pipes are of value in estimating the proper size of pipe to use for long single lines, whereas required pressures usually dictate capacities of distribution networks.

Flow Formulas.—Where the properties of a fluid always are substantially the same, as is the case with water flowing at atmospheric temperature, satisfactory flow computations can be made with empirical formulas. The particular formula chosen usually depends on custom and differs among industries. For instance, waterworks engineers and some others are inclined to use the Williams and Hazen formula, whereas mechanical engineers often use the Saph and Schoder formulas, sewage engineers the Kutter or Manning formulas, and irrigation engineers the Scobey formulas. These formulas are well established in their respective fields where each is associated with friction factors which are mentioned frequently in the literature. Thus, in technical articles on waterworks practice, the flow characteristic of each different kind of pipe or surface condition nearly always is defined as "the value of the (Williams and Hazen) coefficient C" is so and so.[2]

For the convenience of those practicing in different branches of water-supply work, as well as for purposes of comparison, all the aforesaid formulas have been included in this handbook on pages 269 to 289, together with tables and charts giving carrying capacities and pressure drops for different diameters of pipe. References to original sources are given there for the benefit of

[1] See (*a*) "The Economical Diameter of Pipe," by J. W. Ledoux, *Eng. News-Record*, Vol. 96, p. 250, 1926.

(*b*) Article by D. H. Maury, *Eng. News-Record*, Vol. 88, p. 779, 1922.

(*c*) Article by R. W. Powell, *Eng. News-Record*, Vol. 98, p. 499, 1927.

(*d*) "Economic Size for Water Distribution Systems," by Thos. R. Camp, *Proc. ASCE*, December, 1937, p. 1837.

(*e*) "Outline of Design Factors for Steel Water Pipelines," by Russell E. Barnard, *Jour. AWWA*, Vol. 36, No. 1, pp. 23–31, January, 1944.

[2] For data on roughness coefficients for use in the *rational formula* see, "Friction Factors for Pipe Flow," by Lewis F. Moody, *Trans. ASME*, Vol. 66, No. 8, pp. 671–684, November, 1944. This paper contains a bibliography of 16 references and valuable discussion was contributed by several outstanding authorities in this field.

those needing more complete data than space limitations permit presenting in this handbook.

In the case of *hot* water where temperature has appreciably changed the density and viscosity from the atmospheric values on which the empirical formulas are based, it is advisable to use the *rational flow formula* given on pages 107 to 137 if accurate results are desired.

In using the Saph and Schoder charts of Figs. 39*a* and 39*b* and similar data in Table XLII on pages 272 to 275 it should be noted that these apply to wrought pipe in which nominal *inside* diameters are used in sizes up to and including 12 in., and actual *outside* diameters for sizes 14 in. and larger. This conforms to standard nominal sizes for wrought-iron and wrought-steel pipe. Cast-iron pipe, however, is classified according to its nominal *inside* diameter throughout the size range. So this difference in diameter of the two kinds of pipe affects considerably the quantity of water that will be delivered by pipe lines larger than 12 in. in diameter, other conditions remaining the same. The charts and table can be adapted readily to the large sizes of cast-iron pipe by interpolating between the diameter lines. It may be noted that the Saph and Schoder friction head values for wrought pipe, as shown on pages 270 to 275, are in approximate agreement with the results obtained through use of the Williams and Hazen C value of 130 for what they termed "average new cast-iron pipe." The corresponding Saph and Schoder values are said to represent "average conditions for clean cast-iron and wrought-iron or wrought-steel pipe."

The Williams and Hazen flow quantities given for actual inside diameters in Table XLIV on page 280 are listed for $C = 100$ instead of the $C = 130$ which would correspond to the basis for the Saph and Schoder chart. Referring to the Williams and Hazen formula on page 276 it is evident that for the same pressure drop, decreasing C from 130 to 100 corresponds to about a 23 per cent loss in carrying capacity. Or, considering the decrease from smooth new pipe having a C value of 140, the drop to $C = 100$ would amount to a 28½ per cent loss in carrying capacity. With water that causes tuberculation, such losses in capacity are to be expected with either cast-iron or steel pipe over a period of some years (see also values for C given in Table XLIII on pages 277–278). In order to extend the use of Table XLIV to conditions other than those represented by $C = 100$, a set of conversion factors is shown

in connection with the table which makes it possible to adjust to C values either above or below $C = 100$.

Loss in Capacity with Age.—Unfortunately the decrease in carrying capacity of pipes with age is an uncertain quantity and depends on the characteristics both of the water and of the protective coating given the pipe. Cement-asbestos, concrete, and cement-lined iron or steel pipe are relatively immune to the turberculation which, in some waters, so seriously impairs the carrying capacity of steel or iron pipes having only the ordinary tar-dip coating.

According to Williams and Hazen,[1] in making estimates for pipe lines where the carrying capacity after a series of years is the controlling factor, rather than the capacity of new pipe, a considerably depreciated value of C must be used, depending upon the amount of deterioration to be expected. As fair values for general computation they recommended $C = 100$ for cast-iron pipe and $C = 95$ for riveted-steel pipe, with somewhat lower values for small diameters where tuberculations affect a greater percentage of the cross section. They went on to say further that over a period of years that is not long compared with the total life of the pipe, the roughening of the surface and the reduction of cross-sectional area through rusting and tuberculation may reach a point where twice as much head is lost in friction as was the case when the pipe was new. In other words, if average new pipe had a C value of 130, this might be reduced to 65 with time.

Williams and Hazen concluded further that steel pipes tuberculate and corrode in much the same manner as cast-iron pipes, but that riveted pipe, owing to the laps and rivets, will carry only the same quantity of water as cast-iron pipe of the same size and 10 years older. The effectiveness of the protective coating applied to the interior of the pipe has more influence on tuberculation and corrosion than does the quality of the iron or steel of which the pipe is made. Hard waters and lake waters sometimes attack pipe only half as fast as soft and clear but unfiltered river waters. In one case $C = 100$ might be a safe design figure to use, whereas $C = 65$ might be called for under the worse conditions. Muddy waters sometimes deposit sediment in pipes through which there is but little flow and which thus lose their carrying capacity with a rapidity for which no rule can be suggested. A few waters have been noted capable of forming calcium carbonate deposits on the

[1] "Hydraulic Tables," by Gardiner S. Williams and Allen Hazen, John Wiley & Sons, Inc., New York, N.Y., 3d ed., 1920.

interiors of the pipes, reducing their sections and making them rougher.

The decrease in carrying capacity of riveted steel and analogous pipes with age was reported by Scobey[1] as part of his investigation of the relative carrying capacity of different varieties of steel pipe. He concluded that all iron and steel pipes lose capacity progressively when in use, and he set up a chart to show the trend with age, subject to whatever immunity may be afforded by protective coatings. In addition to investigating the effect of corrosion and tuberculation, Scobey studied the effect of silt deposits which often are a factor in irrigation unless scouring velocities are available.

More recently the age-coefficient relation for flow in tar-coated cast-iron pipe and the effect of water quality upon rate of capacity loss were investigated by a committee of the New England Water Works Association.[2] The committee's report also deals with remedial measures for reducing and preventing loss of capacity, including discussion of cement and bitumastic-enamel linings, methods of cleaning and lining pipe in place, and corrective treatment of the water carried. Coefficient values for steel and concrete pipes are discussed in appendixes. Among the outstanding conclusions of the report are (1) that the pH value of the water has a marked effect on the carrying capacity trend, high pH values being conducive to maintaining good carrying capacity and (2) that the carrying capacities under adverse conditions fall off to a greater extent than corresponds to a Williams and Hazen C value of 100. The committee's composite trend for all waters and pipe sizes investigated shows a falling off in capacity over a period of 30 years to a point which corresponds to a C value of 65, whereas Scobey's investigation showed an average reduction after 30 years corresponding to a C of about 110. This discrepancy bears out the committee's conclusion that the quality of the water and its effect upon the rate of capacity loss with age are of sufficient importance to warrent a determination of capacity trends for each city or source of supply.

[1] "The Flow of Water in Riveted Steel and Analogous Pipes," by Fred C. Scobey, *Tech. Bull.* 150, January, 1930, U.S. Department of Agriculture. See pp. 279 to 284 of this handbook for an account of the relative carrying capacities assigned to different kinds of pipe by Scobey.

[2] See "Report of Committee on Pipe Line Friction Coefficients and Effect of Age Thereon," *Jour. NEWWA*, Vol. XLIX, No. 3, pp. 235–310, September, 1935.

Recommendations of the National Board of Fire Underwriters and the National Fire Protection Association concerning decrease in Williams and Hazen C coefficient with age will be found in Table XLIII-*B* on page 278. A statement by the Cast Iron Pipe Research Association is quoted under Change of Interior Condition with Age on page 271. See also page 1261 in Chap. XVII on Corrosion.

Useful Life of Buried Pipes.—The durability of buried water-supply pipes depends chiefly on ability to resist internal and external corrosion (see Chap. XVII on Corrosion). Prior to the development of effective protective coatings for steel pipe, most underground lines were constructed of cast iron for three reasons: (1) the thicker wall of cast-iron pipe provides more material to waste away before failure can take place; (2) cast iron as a material is somewhat more resistant to corrosion than iron or wrought steel; and (3) the original hot-tar-dip coating process is more effective when applied to cast iron than when applied to steel or wrought iron.

The life of cast-iron pipes placed underground may reach or exceed 100 years, and replacement is apt to be caused more by obsolescence or by loss in carrying capacity than by deterioration through corrosion. Hence it is desirable to install cast-iron pipe in liberal sizes, and this result may be obtained by the designer in part through using a low value for the coefficient C in sizing the pipe. This is particularly true in the smaller diameters where encrustations cause a relatively greater restriction to flow and where entry into the pipe for cleaning purposes may be impossible. Where accessible, the interior of medium-sized pipes may be cleaned with a "go-devil" propelled through the pipe by water flow. If long life is desired for cast-iron pipe where the inside is not accessible for cleaning, it is advisable to design for a C coefficient of only 65 for bad water conditions, and not to exceed 100 under favorable water conditions.

Although buried cast-iron pipe resists corrosion remarkably well, it is expensive in large sizes and serious breaks may occur at any time. Hence steel pipes have been used to an increasing extent, particularly in large sizes and on the West Coast where transportation of excess weight is an obstacle to the use of cast-iron pipe. The introduction of new varieties of steel pipe with welded or coupled field joints and better methods of coating has influenced this trend.

Other materials finding wide acceptance for underground water-supply systems are cement-asbestos and precast reinforced con-

crete. Among the advantages of these products are immunity to external corrosion from the soil or internal corrosion from the water, and very little reduction in flow capacity with age, *viz.*, the C coefficient holds up well for the life of the pipe. In common with other pipe materials used underground, but perhaps to a greater degree with cement and concrete pipes, consideration must be given to proper placing in the trench, careful backfilling, and sufficient cover for protection against frost and damage from trucks or other passing equipment (see section on Laying Pipe).

Pressure Drop through Fittings, Valves, and Meters.—General information about pressure drops through fittings and valves will be found on pages 95 to 102, and in Table XIV on page 100. General information about entrance and exit losses for liquids will be found on pages 53 and 72, and about losses due to sudden enlargement or contraction on pages 102 to 103. Losses through fittings used in water service pipes will be found in Table XII on page 1082. In addition, the following specific data applying to water-supply piping may be of interest.

The additional loss of head caused by the presence in a pipe line of certain resistances such as fittings, valves, or meters, as well as entrance and exit losses, may be expressed conveniently in terms of velocity head as $h_\lambda = kv^2/2g$ (see pages 95 to 102) where k is the fractional part or number of velocity heads lost at the point of resistance. Although the values of k given in Table I are approximate, they are sufficiently accurate for many of the problems encountered in waterworks practice.

TABLE I.—FRICTIONAL LOSSES IN WATER-SUPPLY PIPING EXPRESSED IN TERMS OF VELOCITY HEAD
(See text)

Type of resistance	k	Type of resistance	k
Entrance.......	0.5	Venturi meters:	
Exit...........	1.0	Throat ratio 1:2.	2.5
Gate valves:		Throat ratio 1:3.	11.0
Open.......	0.1	Curves:	
Half open....	3.0	180 deg.........	1.1
Tees or crosses:		90 deg.........	0.8
Straight flow.	0.1	45 deg.........	0.4
Angle flow...	1.5	Increaser..........	0.4
		Reducer...........	0.4

For further information on the loss of head through bends, fittings, and valves, references may be made to the published test

TABLE II.—PRESSURE LOSSES THROUGH TYPICAL NUTATING-PISTON (DISK) TYPE WATER METERS
(Meters conform to AWWA Specifications 7M.1 for cold-water meters)

Rate of flow, gpm	Approximate pressure loss[1] through meter, psi							
	Nominal pipe size in.							
	5/8	3/4	1	1½	2	3	4	6
5	0.5	0.4						
10	2.0	1.3	1.0					
15	5.0	3.0	2.0					
20	10	5.0	2.5	1.4				
25	...	8.0	3.5	1.8				
30	...	11.5	4.8	2.2	1.0			
35	...	....	6.0	2.8	1.2			
40	...	....	8.5	3.3	1.6			
45	...	....	11.0	4.0	1.8			
50	...	....	13.5	4.7	2.0			
75	...	....	....	9.5	4.5			
100	...	....	....	16	7.5	1.2		
125	...	....	....	...	12	2.0		
150	...	....	....	...	18	2.5	1.2	
175	...	....	....	...	...	3.5	2.0	
200	...	....	....	...	...	5 0	2.5	0.5
250	...	....	....	...	...	8.0	4.0	1.0
300	...	....	....	...	...	11.0	6.0	1.2
350	...	....	....	...	...	....	8.0	1.6
400	...	....	....	...	...	....	10.5	2.0
500	...	....	....	...	...	....	16.5	3.5
600	...	....	....	...	...	....	....	5.0
800	...	....	....	...	...	....	....	9.0
1,000	...	....	....	...	...	....	....	14.5

[1] These are approximate losses for representative types of disk meters. For actual loss in specific makes of disk or other type meters, consult the manufacturer.

results of Freeman[1] and of Williams, Hubbell, and Fenkell.[2] One of the important conclusions reached in the latter reference is that the fric nal losses due to a single 90-deg turn are at a minimum (measured over a total travel of 80 pipe diameters) when the radius of curvature of the bend is between 2 and 3 pipe diameters. The probable reason for there being an optimum point, instead of a

[1] "Experiments upon the Flow of Water in Pipes and Pipe Fittings," by John R. Freeman, C.E., published by the ASME, 1941.

[2] "Experiments at Detroit, Mich., on the Effect of Curvature upon the Flow of Water in Pipes," by Gardiner S. Williams, Clarence W. Hubbell, and George H. Fenkell, *Trans. ASCE*, Vol. 47, No. 911, pp. 1–369, 1902.

progressive reduction in loss with increased radius of curvature, is that the disturbance produced by the optimum curvature persists over a shorter length of travel than it does with a longer radius bend.

Although the losses through *Venturi* meters are small and can be estimated with considerable accuracy, the losses through *nutating-piston* (disk) type meters commonly used in sizes up to 6 in. may be considerable. Pressure losses through typical nutating-piston meters are given in Table II for their usual flow ranges. Since there is a difference in pressure loss through meters of different makes, the manufacturer should be consulted for actual pressure loss through a specific meter. For requirements as to maximum capacities and corresponding permissible losses of head for various sizes of displacement meters as established in AWWA Specification 7M.1, see Table IV on page 1049.

Water Hammer.—The theory and intensity of water hammer are treated in some detail on pages 291 to 300. The three common causes of water hammer as encountered in water-supply systems are reviewed in the present section with brief accounts of the usual means for combatting the trouble. Since space limitations do not permit covering the subject fully here, it is suggested that reference be made to the papers listed in connection with the respective topics and in the footnotes on page 291.

Water hammer usually is associated with *valves* in one way or another since either a stop valve or a check valve may be the direct cause of the trouble or at least a contributing factor. Where too rapid closure of a stop valve is responsible, the obvious remedy, of course, is to slow down closure so as to take longer than the critical time. Extended discussion of this subject will be found in the various texts and symposiums referenced in the footnote on page 291.[1] Closing a valve completely within less than the critical period for the line may result in nearly all the velocity head of the water going into shock pressure. For each reduction in velocity of 1 ft per sec within the critical period, a shock pressure of the order of 40 to 60 psi can be set up in the line (see Table L, page 296). For water velocities of around 5 ft per sec pressures[2] of

[1] See also "Water Hammer Studies in Long Pipe Lines," by Laurance E. Goit, *Jour. AWWA*, Vol. 31, No. 11, pp. 1893–1908, November, 1939.

[2] See "Reduction of Shock Pressure in Solvent Delivery Lines," by Howard S. Gardner and John H. Folwell, *Ind. Eng. Chem.*, Vol. 31, No. 4, pp. 446–450, April, 1939.

200 to 300 psi are not impossible in small lines, although friction and other factors tend to hold them down. Conditions can be improved through providing a surge relief or air chamber[1] near the valve. Sometimes a combination mechanical-pneumatic type of arrester[2] can be used to advantage on pipes up to about 3 in. in diameter and 50 to 400 ft long.

Water hammer may be transmitted for long distances through water-supply piping so as to affect remote parts of a network, including dead ends and services leading into customer's premises. The intensity is worse under conditions where complete reversal of flow is possible, or where water can surge back and forth in the pipe. Fortunately, since the effect of friction is to dampen vibrations and to make each reversal of diminished intensity, the disturbance gradually subsides. The effect of accelerating a water column by *opening* a valve is analogous to the effect of retardation in closing, except that the pressure variations have the opposite sign. The period of pulsation is the same. The chief difference is that the wave effects are damped out more rapidly with opening than with closing.

Data on the *minimum times* consumed in *closing gate valves* of various sizes given in Table III are of interest for checking against the critical times for closure as computed by the methods given on pages 291 to 297. These are the minimum times expected for manually closing gate valves of usual design; longer times can be taken, of course, if the operator is willing to do so. Whereas this may be feasible in normal operation, it may not be practicable under emergency conditions such, for instance, as arresting flow through a line that has been ruptured farther on.

With *gate valves* the period of effective closing probably is confined to about the *last* 20 *per cent of the valve travel* which is effective in shutting off approximately 80 *per cent of the flow*. Hence the first 80 per cent of valve travel can be executed as quickly as convenient, but the last 20 per cent should be done as deliberately as

[1] See (*a*) "Air Tanks on Pipe Lines," by Minton M. Warren, *Trans. ASCE*, Vol. 82, pp. 250–277, 1918.

(*b*) "Air Chambers for Discharge Lines," by Lorenzo Allievi, *Trans. ASME*, Vol. 59, No. 8, HYD-59-7, pp. 651–659, 1937.

(*c*) "Air Chambers and Valves in Relation to Water Hammer," by R. W. Angus, *Trans. ASME*, Vol. 59, No. 8, HYD-59-8, pp. 661–668.

[2] See "Relief from Water Hammer Pressure," by Lewis H. Kessler, *Jour. AWWA*, Vol. 30, No. 1, pp. 15–37, January, 1938. Also contains a good bibliography on air chambers.

possible. This not only tends to minimize water hammer but is expedient owing to the greater resistance to closure offered as shut-off is approached. Where power-driven operating devices are used, similar precautions should be taken. For geared gate valves the initial period of closure may have to be considerably slower than without gears, but the mechanical advantage available is of great assistance in effecting the last 20 per cent of closure, particularly with large gate valves at high rates of flow. By-passes are a help in closing large valves and should be closed last. The closing characteristics of globe valves and cone valves differ from those of gates as described in the AWWA paper by Goit referenced on page 1031.

With *fire hydrants* care should be exercised to avoid setting up excessive water hammer through too rapid closure. According to AWWA Standard Specifications for Fire Hydrants for Ordinary Water Works Service (abstracted on pages 1043 to 1048) the operating mechanism, particularly the pitch of thread on the stem, shall be so designed that when the operating nut is turned at a proper rate for shutting off the flow of water, the static pressure plus water hammer shall not exceed twice the static if the static averages 60 psi or greater. If the static pressure averages less than 60 psi, the pressure shall not be raised more than 60 psi above static.

Where *branch* lines are involved, or pipes of *different diameters*, the water-hammer problem becomes more involved. The effect of such complications has been discussed in technical papers[1] but is too involved for consideration here.

The second common cause of water hammer in water-supply systems is the *surge* that takes place when the *power supply* to a *centrifugal pump* is suddenly shut off for some reason or fails unexpectedly. Under these circumstances severe water hammer may be set up in either the pump discharge or suction piping, or both, depending on the layout. The fundamental cause of the trouble is the momentum of the column of water flowing through the pipe which tries to continue on toward its destination after the power interruption.

[1] See (*a*) "Methods of Calculating Water-hammer Pressures," by F. M. Dawson and A. A. Kalinske, *Jour. AWWA*, Vol. 31, No. 11, pp 1835–1864, November, 1939.

(*b*) "Computation of Water Hammer Pressures in Compound Pipes," by Robert E. Glover, 1933 *Symposium on Water Hammer*, ASME Hydraulic and ASCE Power Division.

What happens in *pump discharge* lines after a power interruption is somewhat as follows: Immediately after interruption the impeller slows down and the column of water coasts along the discharge pipe away from the pump with an ensuing drop in pressure at the pump. Next, the column slows down and reverses its direction of flow so as to come back toward the point of low pressure at the pump. If there is a check valve at the pump, no continued reversal of flow is possible and a back pressure builds up against the check which, in general, will be about equal to the preceding pressure drop below normal. Under these circumstances, the shock pressure may reach twice the normal head if there has been no breaking of the water column or, if the column has broken, the pressure may rise to a much higher value than twice the normal head. Several of the devices in common use for cushioning shock are mentioned in succeeding paragraphs.

TABLE III.—MINIMUM TIME CONSUMED IN MANUALLY CLOSING GATE VALVES OF DIFFERENT SIZES[1]

Nominal diameter, inches	Number of turns to close	Minimum time of closure, seconds	Nominal diameter, inches	Number of turns to close	Minimum time of closure, seconds
4	9	9	14	45	91
6	13	18	16	52	105
8	27	42	18	58	117
10	32½	58	20	64	158
12	38½	69	24	76	188

[1] Reproduced, by permission, from the "Handbook of Water Control" published by the California Corrugated Culvert Co.

In some cases a *surge relief* is called for, at or near the *outlet end of the line*. Depending on the size of the line and the pressures involved, this may be an air chamber, a relief valve, or an open overflow which lets the oncoming water spill out of the pipe line above some predetermined elevation. Although provision for surge at the outlet end of the line will not necessarily prevent all reversal of flow, it does tend to cushion shock there and at the same time decreases the magnitude of the reversal that is thrown back toward the pump.

Two devices can be provided for *cushioning shock* in the *discharge line at a pump*. The first concerns the check or other form of nonreturn valve at the pump. The ordinary swing-check valve tends to slam shut on reversal of flow, thus causing unnecessarily

severe shock pressure. This trouble can be reduced by using a nonslam tilting-disk check valve (see Fig. 1) or some form of power operated valve (see Fig. 2) which is controlled by a relay actuated from the power circuit or discharge pressure at the pump. Immediately on power interruption the relay acts to start closing the valve whose operating mechanism can be timed to complete closure before reversal of flow can take place. In some cases a spring closing device can be used successfully on a swing-check valve to ensure having the flap close before back surge can start.

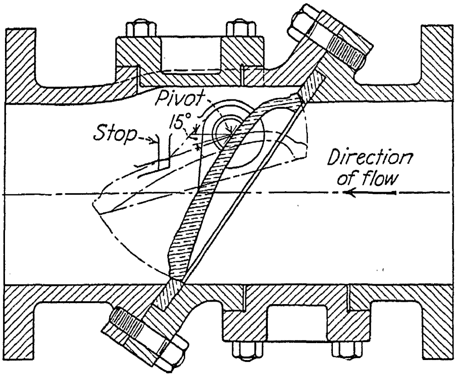


FIG. 1.—Nonslam tilting-disk check valve for reducing shock pressure on reversal of flow in water pipes. (*Courtesy of the Chapman Valve Manufacturing Company.*)

Although the aforesaid devices obviate the slamming to be expected with an ordinary swing check, they cannot prevent a considerable rise in pressure when reversal of flow is stopped at the closed valve. Hence, a second device in the nature of a *suppressor* is required, which may take the form of an air chamber[1] or a relief valve.[2] Such devices must be of ample size in order to accomplish

[1] See (*a*) "Pressure Air Chambers in Centrifugal Pumping," by W. L. Boerendans, *Jour. AWWA*, Vol. 31, No. 11, pp. 1865–1892, November, 1939.

(*b*) "Relief from Water Hammer," by Lewis H. Kessler, *Jour. AWWA*, Vol. 30, No. 1, pp. 15–37, January, 1938.

(*c*) "Methods of Calculating Water Hammer Pressure," by F. M. Dawson and A. A. Kalinske, *Jour. AWWA*, Vol. 31, No. 11, pp. 1835–1864, November, 1939.

[2] See "Eliminating Surge Troubles on Pump Discharge Lines of Washington Water Supply," by D. M. Radcliffe, *Water Works & Sewer.*, May, 1934, pp. 145–147.

the desired result. Air chambers sometimes perform more effectively when damped with orifices or check valves in the connecting pipe.

Where a *relief valve* is *power operated* and *actuated* by a *relay* so as to open before reversal of flow takes place, overpressure can be held

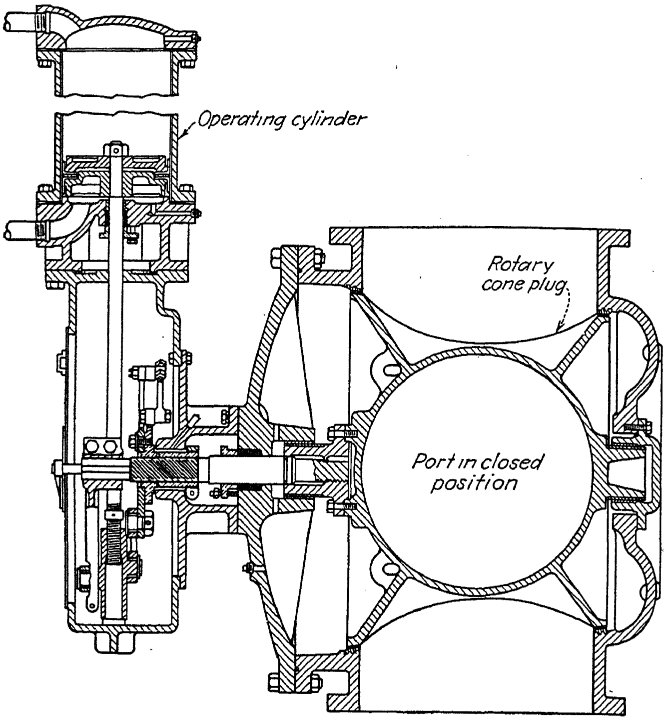


FIG. 2.—Hydraulically-operated pilot-controlled cone valve for use as a check valve or a relief valve for reducing shock pressure due to water hammer. (*Courtesy of the Chapman Valve Manufacturing Company.*)

down so as not to rise more than 10 to 20 per cent above normal operating pressure, although there may be an initial drop in pressure at the pump down to atmosphere or below. With the surge relief valve open, however, all succeeding reversals are dissipated through the open valve. The pipe line then assumes a penstock

condition and the surge relief valve must be closed very slowly to prevent penstock surges. With large diameter lines for low-pressure water service the economic justification for rather elaborate protective devices is obvious since without them the lines would have to be designed for shock pressures considerably in excess of the normal working pressure. This is particularly true in the case of concrete pipes, or of thin-walled steel pipes as used in the West. With thin-walled steel pipe where the pressure may fall below atmospheric under shock conditions, it may be necessary to provide vacuum breakers to prevent collapsing the pipe (see pages 1059 to 1063).

What happens in a *pump suction* line after a *power interruption* depends, of course, on the arrangement of the suction piping. Nothing of much consequence will occur where the suction line is short, and considerable suction lift has to be developed by the pump in order to get water to flow to it. In the case of a *booster* pump, however, where water flows to the pump suction through a long line under pressure or by gravity, the result of a power interruption is much like what takes place in a discharge line and the measures taken to cushion shock are similar. To be most effective a suppressor in such a booster pump suction should be placed close to the pump. The booster-pump problem frequently is encountered in connection with the intermittent filling of standpipes such as overhead sprinkler tanks, pressure tanks in tall buildings, and locomotive filling tanks in railroad yards. In cases where frequent fillings are required, the shock-pressure problem may be most annoying and corrective measures clearly called for, especially if the pump suction is taken from a branch line off a distribution network which may be adversely affected for miles back from the pump. As an alternative to installing suppressors in such cases, consideration should be given to providing automatic means for slowly closing a valve in the pump discharge before power is cut off.

Another pump-suction problem involving surge on a large scale is encountered in *waterworks intakes* where the pumps may be fed through a conduit extending for miles from some lake or a reservoir in the mountains. In order to look after surge in the case of a sudden power interruption it may be necessary to provide ample relief valves or a gravity overflow discharging to a receiving basin of generous proportions, for instance the raw-water reservoir. A

the desired result. Air chambers sometimes perform more effectively when damped with orifices or check valves in the connecting pipe.

Where a *relief valve* is *power operated* and *actuated* by a *relay* so as to open before reversal of flow takes place, overpressure can be held

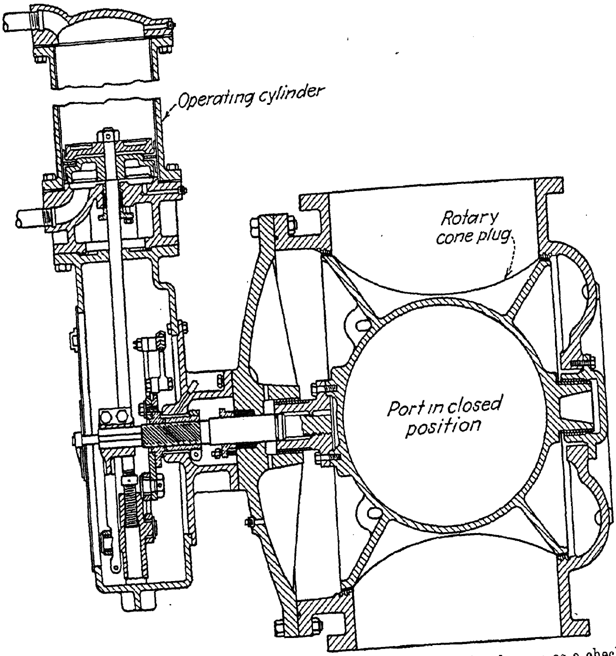


FIG. 2.—Hydraulically-operated pilot-controlled cone valve for use as a check valve or a relief valve for reducing shock pressure due to water hammer. (*Courtesy of the Chapman Valve Manufacturing Company.*)

down so as not to rise more than 10 to 20 per cent above normal operating pressure, although there may be an initial drop in pressure at the pump down to atmosphere or below. With the surge relief valve open, however, all succeeding reversals are dissipated through the open valve. The pipe line then assumes a penstock

condition and the surge relief valve must be closed very slowly to prevent penstock surges. With large diameter lines for low-pressure water service the economic justification for rather elaborate protective devices is obvious since without them the lines would have to be designed for shock pressures considerably in excess of the normal working pressure. This is particularly true in the case of concrete pipes, or of thin-walled steel pipes as used in the West. With thin-walled steel pipe where the pressure may fall below atmospheric under shock conditions, it may be necessary to provide vacuum breakers to prevent collapsing the pipe (see pages 1059 to 1063).

What happens in a *pump suction* line after a *power interruption* depends, of course, on the arrangement of the suction piping. Nothing of much consequence will occur where the suction line is short, and considerable suction lift has to be developed by the pump in order to get water to flow to it. In the case of a *booster* pump, however, where water flows to the pump suction through a long line under pressure or by gravity, the result of a power interruption is much like what takes place in a discharge line and the measures taken to cushion shock are similar. To be most effective a suppressor in such a booster pump suction should be placed close to the pump. The booster-pump problem frequently is encountered in connection with the intermittent filling of standpipes such as overhead sprinkler tanks, pressure tanks in tall buildings, and locomotive filling tanks in railroad yards. In cases where frequent fillings are required, the shock-pressure problem may be most annoying and corrective measures clearly called for, especially if the pump suction is taken from a branch line off a distribution network which may be adversely affected for miles back from the pump. As an alternative to installing suppressors in such cases, consideration should be given to providing automatic means for slowly closing a valve in the pump discharge before power is cut off.

Another pump-suction problem involving surge on a large scale is encountered in *waterworks intakes* where the pumps may be fed through a conduit extending for miles from some lake or a reservoir in the mountains. In order to look after surge in the case of a sudden power interruption it may be necessary to provide ample relief valves or a gravity overflow discharging to a receiving basin of generous proportions, for instance the raw-water reservoir. A

good example of such a case is the surge well of Springwells Pumping Station[1] at Detroit, Mich.

The third common cause of shock pressure in water-supply systems is experienced with *reciprocating pumps* which cause pulsation problems not encountered with the continuous action of centrifugal pumps. Owing to the irregularity of flow through a reciprocating pump, more or less water hammer develops in the suction and discharge lines and cannot be suppressed entirely with vacuum or air chambers. For this reason it is advisable to design the suction and discharge lines of reciprocating pumps for something like 50 per cent in excess of the normal working pressure and to provide ample air chambers at the pumps. Shock conditions obtaining with *hydraulic rams* are decidedly worse than with reciprocating pumps, and generous provision should be made in the design of their piping. An allowance of at least 300 psi extra beyond the working pressure would seem called for with rams.

Flow through Networks.—The general subject of flow through *complex pipe lines* is covered on pages 90 to 95 where references are given to the *Hardy Cross method* and other systems for computing flow through distribution systems or networks. It is a well-understood fundamental that if flow is divided between two pipes of *equal* size, length, and smoothness, then the same quantity will flow through each and the friction loss in each circuit will be equal. Another fundamental is that if two pipes are of *different* diameter, length, or smoothness, then such a quantity will flow through each circuit as will produce the same total friction loss in each. Determination of the respective quantities needed for equal friction losses formerly was difficult and often required a tedious amount of cut-and-try calculation. Many methods of computation have been devised, most of which are either highly technical, or too cumbersome or tedious to be generally adopted. Since the Hardy Cross method was introduced in 1936, however, its relative simplicity has led to wide acceptance of this method and the systems based on it in the waterworks industry.

Charts for reducing parts of a water-supply system to an equivalent pipe size for use in the Hardy Cross solution will be found in an article by H. W. Clark.[2]

[1] See "Design Features of a 90-Ft Circular Low-lift-pumping Plant," *Eng. News-Record*, Mar. 19, 1931, pp. 468–472.

[2] See "Charts for Determining Equivalent Pipes and Loop Flow Distribution," by H. W. Clark, *Water Works & Sewerage*, pp. 313–316, September, 1944.

MATERIALS AND SPECIFICATIONS

The general subject of specifications and standards for pipe, valves, and fittings is covered in Chap. IV. Many new kinds of material and types of joints which have come into use comparatively recently are described there together with the older forms of construction. Certain additional information directly applicable to water-supply piping is presented here.

Pipe and Joints.—A great variety of *piping materials and joints* are used for water mains to suit diverse operating conditions and available natural resources or transportation conditions. For instance cast-iron, steel, reinforced-concrete, and cement-asbestos pipe all are in common use east of the Rockies where these materials are abundant and easily transported to the job. On the Pacific Coast and in the mountains, however, wood-stave pipe is used to some extent and light weight steel pipe finds favor owing to less troublesome water conditions and the long freight haul for the heavier materials. Likewise for services and small branch pipes, galvanized iron or steel, brass, copper, or lead may be used depending on local soil and water conditions and the life expectancy of the line. Descriptions of these materials and specifications for their purchase will be found on pages 406 to 478. Ferrous pipe used underground in water systems usually is given a *protective coating* inside and out for which specifications are given also. For a discussion of *corrosion* in aboveground or underground water-supply piping reference may be made to Chap. XVII on Corrosion.

Steel pipe is furnished in *lengths* of from 15 to 40 ft depending on the process by which it is made. Longer lengths are obtainable by some processes, but transportation would be a problem in lengths over 40 ft no matter how light the pipe. Cast-iron pipe usually is furnished in lengths of 6, 12, or 16 ft depending on the type of pipe and process of manufacture. Precast-concrete pipe usually is furnished in 12- or 16-ft lengths and cement-asbestos pipe in 13-ft lengths. Small brass and copper pipe of iron-pipe-size dimensions comes in 15- to 18-ft lengths, and copper and lead tubing in coils of 50 to 100 ft.

For customers' *services* the use of threaded galvanized-iron or steel pipe or threaded brass or copper pipe of iron-pipe-size dimensions may be justified in some communities, but copper tubing or lead pipe are more suited for this purpose in the smaller sizes, and cast-iron, coated-steel, or cement-asbestos pipe for larger services.

Type *K* copper tubing with soldered joints or flared-tube connectors is the most extensively used material underground and is highly regarded because of its corrosion-resistant properties, flexibility, ease of installation, and low resistance to flow throughout its useful life. Lead possesses many of the advantages of copper tubing but is open to the objections that it is unsuited to high pressures and is dissolved by some waters to an extent that may produce an occasional case of lead poisoning. Further discussion of materials and construction for underground services will be found on pages 1175–1179 in Chap. XV on Gas Piping,[1] and in Chap. XVII on Corrosion where susceptibility to soil corrosion is considered as well as internal corrosion.

Joints of various sorts (see pages 422 to 471) are used with water-supply piping depending on the pipe material and whether the line is installed above- or underground. *Cast-iron* pipe installed underground usually has bell-and-spigot joints made up with lead and oakum, although cement or sulphur compounds sometimes are used for jointing. Bell-and-spigot joints in high-pressure fire lines may be reinforced with bolting lugs. Bell-and-spigot cast-iron pipe with mechanical joints made up with bolts, follower rings, and rubber gaskets is used to some extent for water-supply piping. Flanged cast-iron pipe, if used, generally is aboveground or in manholes or pits. Couplings of the Dresser or Victaulic or similar types (see pages 426 to 429) may be used either above- or underground and with either cast-iron or steel pipe.[2] Where Victaulic couplings are used for jointing thin-walled pipe, grooved adapter bands must be welded to the pipe ends to provide a seat for the coupling. Welded joints are common with *steel* pipe in either location, while riveted joints are used to only a limited extent.

With steel *irrigation* pipe, joints may be made up with various clamping devices intended to facilitate assembling and disassembling the line for moving it around. One of the simplest devices of this sort used for low pressure is the "drive" joint where the lengths insert inside one another like stovepipe, are held together with clips attached to the pipe, and are bolted or wired to one another. In another type of portable pipe the joints are made up

[1] See also committee report by J. E. Gibson on "Service Pipe Materials and Practice in the U. S. and Canada," *Jour. AWWA*, Vol. 23, p. 1935, October, 1931.

[2] See "Mechanical Joints for Water Lines," by Elson T. Killam, *Jour. AWWA*, Vol. 35, No. 11, pp. 1457–1471, November, 1943.

with rubber gaskets and bales similar to the arrangement used on one style of Mason fruit jar.

Several designs of joints suitable for water-supply piping are described in considerable detail on pages 422 to 471, including those used with cement-asbestos and reinforced-concrete pipe. Other joints to be used with water-supply piping are described in Chap. XV on Gas Piping.

Valves and Hydrants.—In order to avoid excessive maintenance troubles only high-class valves and hydrants should be installed in municipal water-supply systems. The operating mechanisms of all valves and hydrants in the same system should turn the same way to open. The usual standard is clockwise to close as one looks down at the stem, and counterclockwise to open, although the opposite holds true in some large systems. Large valves should be provided with gearing to facilitate opening and closing. Where the gearing introduces any departures from the usual direction of rotation, an arrow should be provided to show the direction for opening or closing.

Gate Valves.—Straightway gate valves are used almost entirely for municipal water distribution and, where installed underground, they usually are of the inside screw type with nonrising stem. Valves equipped with street boxes are provided with square-stem nuts for extension keys instead of handwheels. The latter may be used aboveground and in pits or manholes. Valve gates may be of either the solid-wedge or the double-disk type. A typical gate valve for underground water service equipped with stem nut for extension wrench is shown in Fig. 3.

Where desired, quality gate valves may be purchased to the AWWA Standard Specifications for Gate Valves for Ordinary Water Works Service, 7F1[1] which have been approved also by the New England Water Works Association.. These specifications set forth the requirements for valve construction in full detail and should be consulted if and as required. The sizes covered range from 3 to 48 in., and the valves are suitable for pressures not exceeding 150 psi. By-passes shall be provided on sizes 16 in. and larger. Each valve shall be plainly marked with the manufacturer's name or symbol, the year in which it was made, the size, and the designation "150 W" indicating the water-working pressure, all cast on the bonnet or body. Each valve shall be sub-

[1] Copies may be obtained from the American Water Works Association, 500 Fifth Ave., New York 18, N.Y.

jected to a hydrostatic test of 300 psi without showing leakage through the metal or permanent deformation of any casting. Under 100 psi hydrostatic test, all joints and seats shall be perfectly watertight. All ferrous parts of valves, except finished or bearing surfaces, shall be given two coats of asphaltum varnish or pipe dip. After the values are assembled and tested, a third coat shall be applied on the exterior.

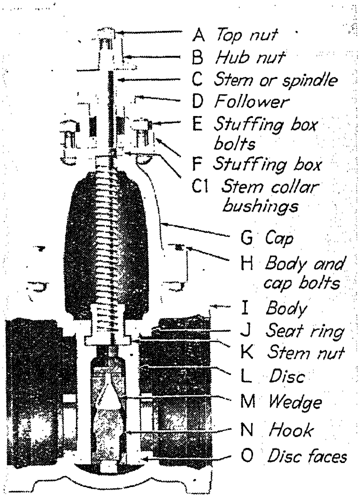


FIG. 3.—A typical AWWA double-disk gate valve for underground water service.

Sluice Gates.—In the control of water flowing by gravity in one direction through pipes or other conduits, it is necessary to use a regulating gate at the entrance to each main conduit and branch line from the beginning of the system to the end. Since flow is in one direction only, a sluice gate such as that illustrated in Fig. 4 can be used instead of a more complicated and expensive gate valve. Unless otherwise specified, sluice gates are designed only for face pressures tending to seat the valve. Ordinarily this type of construction is equal to supporting back pressures which are only a fraction of the rated face pressure. When sluice gates are used with pipe or other conduits and under considerable head, air vents should be provided immediately back of the gates as shown in Fig. 4.

Among the uses for sluice gates are: intakes for penstocks on hydraulic power developments; waste gates for dams; control gates for tunnels, canals, filter plants, reservoirs, and sewage-disposal plants; head, sluice, and distributing gates for irrigation and reclamation projects. Sluice gates suitable for waterworks practice may be ordered to the AWWA Tentative Specifications for Sluice Gates.[1] Up to and including the 36-in. size the AWWA gates are suitable for heads not in excess of 75 ft and for sizes from 36 to 96 in. for heads not in excess of 50 ft. The gates may be round or rectangular, or of other shape. For irrigation purposes and other applications where first cost rather than durability may be of paramount importance, lower heads and less expensive standards of construction may be called for than in municipal water-supply systems or hydro-power developments. Sluice gates for lesser applications may or may not, as the case may be, need to be up to the AWWA specifications. Hence there are cases where it is advisable to follow the manufacturer's standard practice or where an individual specification is warranted.

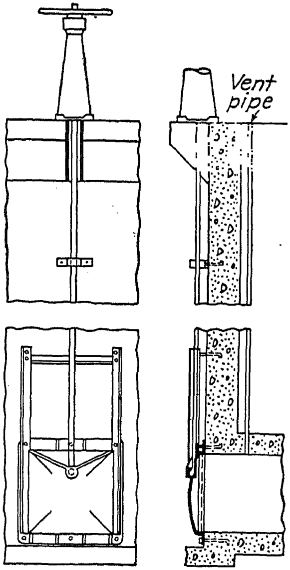


FIG. 4.—Sluice gate installation for gravity water conduit.

Drainage Gates.—There are several applications where a nonreturn valve is required for preventing backflow from a reservoir or through the outlet of a gravity drainage system. Such devices are known under various names such as drainage gate, flap gate, flood gate, and tidal or flume gate. A typical drainage gate or flap valve is shown in Fig. 5. Flap gates are used extensively on irrigation and drainage systems to prevent backflow under flood conditions.

Fire Hydrants.—In purchasing hydrants, reference can be made to AWWA Standard Specifications 7F.3 for Fire Hydrants for

[1] See footnote, p. 1041.

Ordinary Water Works Service,[1] which have been approved also by the New England Water Works Association. This specification covers post hydrants having either the compression type (opening either with or against the pressure) and the gate type of shutoff. A typical post hydrant of the compression type is shown in Fig. 6 and installation details for same in Fig. 7. High-pressure, independently gated, and special hydrants are not included.

The size of the hydrant shall be designated in terms of the minimum opening through the seat ring of the main valve and shall be stated in inches of diameter. The valve opening shall be at least 4 in. for hydrants having two 2½-in. nozzles, 5 in. for three 2½-in. nozzles, and 6 in. for four 2½-in. nozzles. For other combinations of nozzles the main valve opening shall have a capacity equivalent

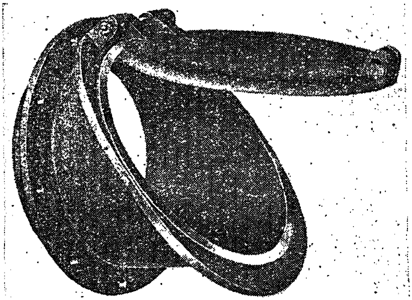


FIG. 5.—A typical drainage gate or flap valve.

to the foregoing list. Since it is understood that hose nozzles and pumper nozzles are not in action at the same time, one pumper nozzle may be added to any of the foregoing hose nozzle combinations, or substituted for any one or two hose nozzles, provided the area of the pumper nozzle does not exceed the area of the 2½-in. hose nozzles or that of the main valve of the hydrant.

The AWWA standard specifications for Laying Cast-iron Pipe, 7D.1, require that each hydrant shall be connected to the main pipe with a 6-in. cast-iron branch controlled by an independent 6-in. gate valve, except as otherwise directed. According to the regulations of the National Board of Fire Underwriters (see pages 1100 to 1103) no pipe smaller than 6 in. should be used for mains or hydrant branches. On the other hand a fire line should not be so large as to cause undue loss of pressure on the whole adjacent water system in case the line is ruptured during a fire. Unless otherwise accessible, the independent shutoff gate should be provided with a wrench-nut and valve box so that the valve can be closed with an extension wrench. The purpose of this gate valve is to permit taking the hydrant apart for repairs, or replacing it, if necessary, without having

[1] Such specifications are revised from time to time and reference should be made to the latest issue. See footnote, p. 1041.

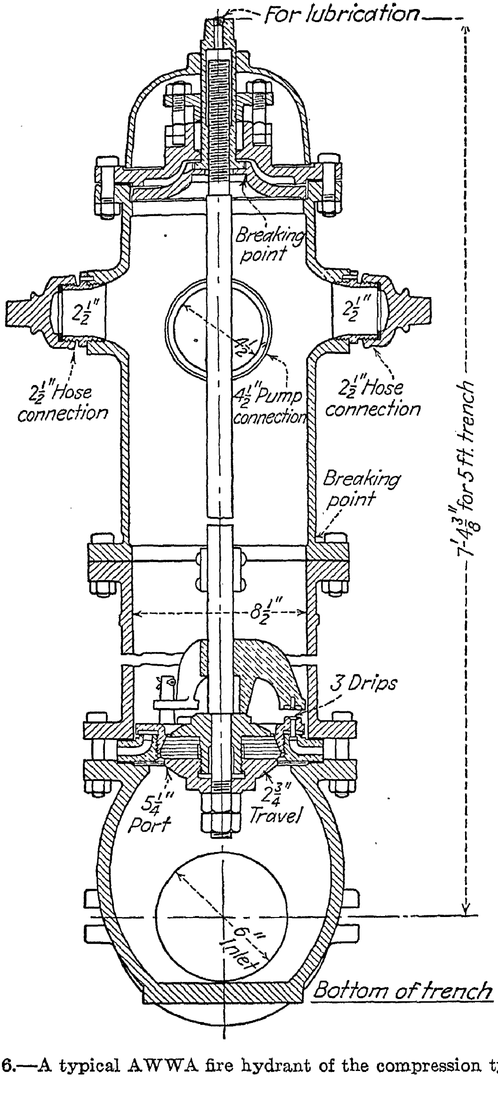


Fig. 6.—A typical AWWA fire hydrant of the compression type.

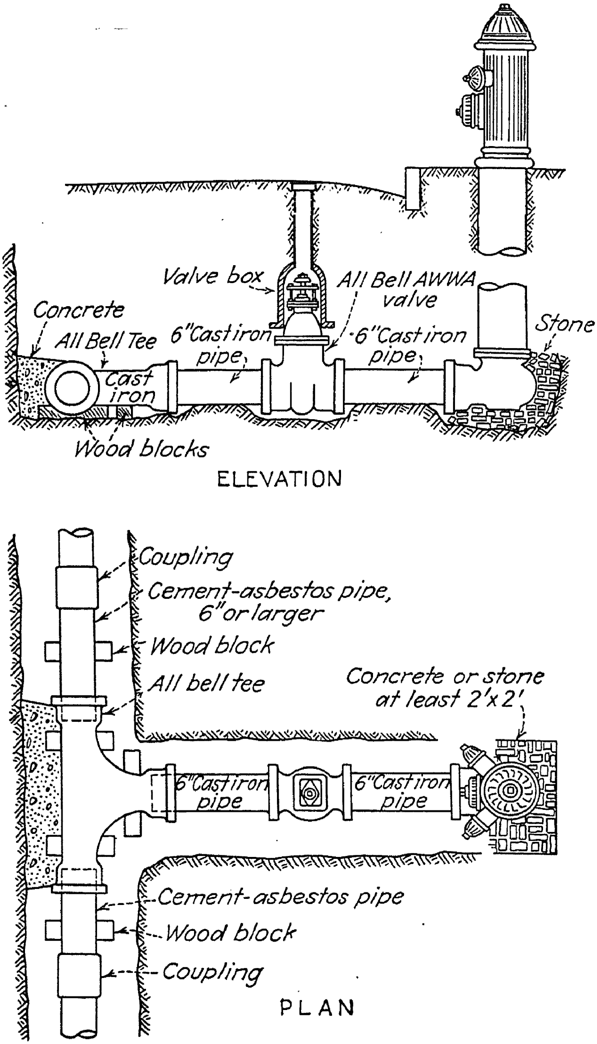


FIG. 7.—Typical installation of an AWWA fire hydrant.

to shut down the main. Details of a fire-hydrant installation are shown in Fig. 7.

The length of hydrant shall be expressed in feet (to the nearest half foot) to correspond to the depth below finished or established grade of the trench in which it is to be installed. Expressed in other terms, the length is the distance from the bottom of the connecting pipe to the ground line at the hydrant. With a hydrant 5 ft in length discharging 250 gpm through each 2½-in. hose outlet, the *friction loss* shall not exceed 1 psi times the number of hose outlets in use.

Unless otherwise specified, hose coupling threads on hydrants shall conform to the American Standard for Fire Hose Coupling Screw Threads, ASA B26, for all connections having nominal inside diameters of 2½, 3, 3½, and 4½ in. (see pages 486 to 488). Hose nozzles shall be of composition metal and fastened into the barrel of the hydrant by a fine thread or by leading, and suitably safeguarded against coming loose. Nozzle caps shall be provided for all hose outlets with threads conforming to those of the nozzle. Dimensions of the cap nut shall be similar to those of the operating nut which usually is pentagonal in shape and of dimensions specified by the AWWA. Unless otherwise ordered, caps shall be securely chained to the barrel. Caps shall have a gasket recess at the inner end of the threads and, when so specified, shall be provided with a suitable gasket for securing a tight seal with the nozzle.

Hydrants shall be so designed that, when in place, no excavation will be required to remove the main valve and the movable parts of the drain valve. The design of the barrel and operating mechanism shall be such that in case of drainage to the hydrant above or near grade level, the main valve will remain closed reasonably tight against leakage or flooding.

A positive operating noncorrodible drain valve or valves shall be provided to drain the hydrant properly by opening as soon as the main valve is closed, and to close tightly when the main valve is opened. Wherever hydrants are set in impervious soil, a drainage pit 2 ft in diameter and 2 ft deep shall be excavated below each hydrant and filled compactly with coarse gravel or broken stone mixed with coarse sand, under and around the bowl of the hydrant and to a level 6 in. above the drain outlet. No hydrant drainage pit shall be connected to a sewer.

The thickness of the wall of the barrel shall not be less than the thickness specified for Class 250 pit-cast cast-iron pipe of like diameter produced in accordance with American Standard Specifications ASA A21.2 and Table 3 thereof (see abstract on pages 432 to 437). Hydrants for a working pressure of 150 psi or less shall be subjected after assembly to two shop tests under a hydrostatic pressure of 300 psi. One test shall be made with the whole interior of the hydrant under pressure; the second with the main valve closed and the foot piece under pressure from the inlet side. Unsatisfactory conditions shall be corrected before the hydrant is accepted.

Rules for anchoring or bracing hydrants and other fittings will be found in the AWWA Standard Specifications for Laying Cast-iron Pipe (abstracted on pages 1050 to 1055). Metal harness of tie rods and pipe clamps of adequate strength, or other suitable support, may be used instead of concrete backing as directed by the engineer. When so ordered by the purchaser, *lugs* for harnessing the hydrant to the connecting pipe from the street main shall be provided on the bell of the inlet elbow.

All hydrants shall have permanent *markings* identifying the manufacturer, the size of main valve opening, and the year of manufacture. Rules are given for cleaning and painting in the manufacturer's shop. An optional system for the

uniform marking of fire hydrants has been adopted by the AWWA and other organizations, but is not generally followed. In this uniform marking system public fire hydrants have their barrels painted chrome yellow while the tops and nozzle caps are green, orange, or red depending on their capacity. (NOTE.—This scheme has not made much progress in the face of the well-established convention of painting fire hydrants red.)

As a matter of economy in small communities where freezing is not a problem, hydrants sometimes consist of a riser of 3-in. or 4-in. pipe carrying at the top a special angle valve having a hose nipple with cap and a pentagonal operating nut. Although such an arrangement may not comply with the underwriters' requirements, it is better than no fire protection.

Meters.—Water meters are classified primarily into two types: positive displacement meters available in sizes ⅝ to 6 in., and current or velocity meters in sizes 1½ to 20 in. *Displacement* meters may be of the reciprocating, rotating, oscillating, or nutating piston (formerly known as disk) types, all of which measure water by actual displacement. The displacement meter is used almost exclusively for measuring domestic and manufacturing water consumption where a reasonably accurate measure is desired involving unsteady flow of a relatively low rate. *Current* or *velocity* meters measure the velocity of the stream flow past a cross section of known area from which the volume of water passed may be estimated. They are adapted to measuring large volumes of water under steady-flow conditions where a low friction loss is desirable, such as for fire lines, and filling standpipes or tanks. Velocity meters are lower in first cost and require little maintenance. Turbine and Venturi meters are examples of the velocity type in common use. Water meters for *irrigation* purposes are described on pages 1085 to 1088.

Compound meters usually are a combination of a displacement meter in parallel with a velocity meter in which means are provided for deflecting small flows through the displacement meter while large flows go through the current meter. Compound meters measure unsteady flows and at the same time permit high velocities with low frictional losses. They are particularly adapted where the rate of flow varies to such an extent that a single meter of either type cannot be used with accuracy.

Tentative Specifications for Cold Water Meters, Displacement Type, 7M.1-T, were issued by the American Water Works Association under date of November, 1941. These specifications cover the various types of positive displacement meters previously

enumerated. The nutating-piston (disk type) displacement meter is the most commonly used variety for measuring water quantities up to 1,000 gpm, being used extensively for residences. Typical frictional losses through representative makes of this type of meter are shown in Table II on page 1030.

Displacement meters are practically positive in action and the pistons displace or carry over a fixed quantity of water for each stroke, revolution, oscillation, or nutation. Requirements are given in Table IV for maximum permissible losses of head for the corresponding safe maximum operating capactities which represent the peak rates of flow at which water can be passed through the meters for *short periods of time* without destructive wear. For continuous 24-hr service meters of the displacement type should not be operated on flows greater than one-fifth the peak capacity of the meter. Table IV also sets the minimum test flows and normal test-flow limits, and the speed of piston, revolutions, or nutations per cubic foot of water metered. Registration on the meter dial shall indicate the quantity recorded to be not less than 98.5 nor more than 101.5 per cent of the water actually passed through the meter while it is being tested at any rate of *normal flow* within the limits given in Table IV.

TABLE IV.—MAXIMUM AND MINIMUM CAPACITIES AND NORMAL FLOW RANGES FOR DISPLACEMENT WATER METERS
(Table 1 of AWWA Specification 7M.1-T)

Meter size, inches	Safe maximum operating capacity, gallons per minute	Maximum loss of head, lb per sq in.	Minimum test flow, gallons per minute	Normal test flow limits, gallons per minute	Speed of piston, revolutions or nutations per cubic foot
5/8	20	15	1/4	1 to 20	435
3/4	30	15	1/2	2 to 30	250
1	50	15	3/4	3 to 50	115
1 1/2	100	20	1 1/2	5 to 100	50
2	160	20	2	8 to 160	30
3	300	20	4	16 to 300	15
4	500	20	7	28 to 500	7
6	1,000	20	12	48 to 1,000	3

AWWA Specification 7M.1-T contains detail requirements for building, testing, and calibrating such meters, with a comprehensive set of notes of an advisory nature which are included for information. Over-all dimensions and register indications for the various sizes of meters are given in Table V.

TABLE V.—OVER-ALL DIMENSIONS AND REGISTER INDICATIONS FOR DISPLACEMENT WATER METERS
(Table 2 of AWWA Specification 7M.1-T)

Meter size, inches	Meter length		Tail piece length, inches	Maximum indication of initial dial		Minimum capacity of register	
	Screw ends, inches	Flange ends, inches		Cubic feet	Gallons	Cubic feet in millions	Gallons in millions
5/8	7½	....	2⅜	1	10	0.1	1
¾	9	....	2½	1	10	1	10
1	10¾	....	2⅝	10	100	1	10
1½	12⅝	13	...	10	100	10	100
2	15¼	17	...	10	100	10	100
3	....	24	...	10	100	10	100
4	....	29	...	100	1,000	100	1,000
6	....	36½	...	100	1,000	100	1,000

Elevated Storage Tanks.—Standard Specifications for Elevated Steel Water Tanks, Standpipes and Reservoirs, 7H.1,[1] have been adopted by the AWWA in collaboration with the American Welding Society. These specifications provide a guide for minimum requirements for the design, fabrication, and erection of elevated steel water tanks of either welded or riveted construction. Requirements for welding are taken from the AWS Rules for Field Welding of Steel Storage Tanks. A full list of ASTM specifications for acceptable materials is included, and safe unit stresses are set for design loadings which embrace dead load, live load, snow load, wind load, earthquake load, and balcony and ladder load. In localities where freezing temperatures are experienced, the steel riser pipe shall be not less than 36 in. inside diameter unless the riser is heated to prevent freezing. In extremely cold climates a minimum diameter of 72 in. is recommended unless the riser is heated. The tank shall be equipped with an overflow which shall be fitted, where specified, with a drain pipe of black steel pipe. Rules are included for water testing of tank seams and for painting.

Laying Water Mains.—Good practice in laying underground water mains is described in full detail in various standard specifications, recommended practice manuals, and manufacturers' bulletins of which the following are listed here for reference:

1. AWWA Standard Specifications for Laying Cast-iron pipe, 7D.1.

[1] See footnote, p. 1041.

2. American Recommended Practice Manual for the Computation of Strength and Thickness of Cast Iron Pipe, ASA A21.1.

3. Specification for Laying Transite (Cement-asbestos) Pipe, Johns-Manville bulletin.

4. Installation Manual for Transite Pressure Pipe, Johns-Manville bulletin.

Several important requirements of the foregoing references are abstracted below. Descriptions of different types of pipe joints for underground service will be found on pages 422 to 471, together with estimates of the amount of jointing materials required for bell-and-spigot joints and recommended practice for making them up.

Methods of Laying Pipe.—The various methods used for preparing a trench and backfilling around pipe as described in American Recommended Practice Manual, ASA A21.1, are listed on page 431. Of these, the common methods for laying water pipe are: (*a*) on a flat-bottom trench and (*b*) on wood blocks. In either of these methods the backfill may be tamped, or not tamped, according to which practice is specified. Properly tamped backfill was found by ASA Committee A-21 to have a large influence on the required metal thickness of pipe and, unless required otherwise, cast-iron pipe should be laid on a flat-bottom trench with well-tamped backfill. Where blocks are used, the AWWA Standard Specifications for Laying Cast-iron Pipe provide that two blocks shall be placed under 12-ft lengths, three blocks under 16- or 18-ft lengths, and four blocks under 20-ft lengths. In each case the end blocks shall be placed with their centers 30 in. from the joints. Wood wedges placed on top of the blocks and transversely to the line of pipe shall be used to hold it in alignment.

In the case of *cement-asbestos* pipe, Johns-Manville recommends that wood blocks be placed at two points under each length and that the backfill be tamped uniformly under the whole length of pipe and up to the horizontal diameter. Both laying specifications agree that selected backfill free from rocks or other unsuitable substances shall be used around the pipe and for a distance of 1 ft or more above the top of the pipe. Succeeding layers of backfill may contain coarser materials, but no rock larger than 8 in. in its greatest dimensions shall be used in backfilling.

According to the AWWA laying specification, ledge rock, boulders, and large stones shall be removed to provide a *minimum clearance* of 6 in. below and on both sides of the line for pipes 16

in. or less in diameter, and a minimum clearance of 9 in. for pipes over 16 in. in diameter. Adequate clearance for properly joining pipe laid in rock trenches shall be provided at bell holes. The extra depth of excavation shall be refilled to proper grade with suitable material. Likewise when the bottom uncovered at subgrade is too soft to support the pipe properly, a further depth and/or width shall be excavated and refilled with suitable material, or other approved means shall be adopted to assure a firm foundation.

The minimum *width* of *unsheeted trench* shall be 18 in., and for pipe 10 in. or larger at least 1 ft greater than the nominal diameter of the pipe, except by consent of the engineer; the maximum clear width of trench shall not be more than 2 ft greater than the pipe diameter. Where *sheeting* and *bracing* are used, the trench width shall be increased accordingly. Sheeting shall remain in place until the pipe has been tested and the backfill compacted to a depth of 2 ft over the top of the pipe.

TABLE VI.—MAXIMUM PERMISSIBLE LEAKAGE IN U.S. GALLONS PER DAY PER MILE PER INCH DIAMETER FOR PIPE LENGTHS AND PRESSURES STIPULATED[1]

Working pressure, pounds per square inch	Leakage, U. S. gallons				Leakage ratio to 150 pounds per square inch pressure
	Pipe lengths, feet				
	12	16	18	20	
150	100	75	66.6	60	1.0
125	91.4	68.6	60.9	54.8	0.914
100	81.7	61.3	54.4	49.0	0 817
75	70.0	53	47.1	42.4	0.707
50	57.7	43.3	38.4	34.6	0.577

[1] From AWWA Standard Specifications for Laying Cast-iron Pipe, 7D.1-1938.

Testing and Flushing Mains.—The AWWA specifications for laying pipe require that all foreign matter or dirt shall be removed from the inside of the pipe before it is lowered into position in the trench, and that it shall be kept clean during and after laying. This is intended to preclude damage to meters and to the seats of valves and hydrants as well as to preserve the purity of the water supply. All newly laid sections after bracing and partial backfilling around the pipe shall be subjected to and successfully with-

stand a hydrostatic test pressure 50 per cent above normal working pressure for a period of at least 30 min. No pipe installation will be accepted unless leakage under normal working pressure is less than the amount scheduled for its respective diameter, length, and pressure in Table VI. The evaluation of the actual leakage to the leakage under the standard pressure of 150 psi is calculated from the ratios of the square roots of the respective pressures, other factors being equal.

Before being placed in service all new mains for distributing potable water should be thoroughly *flushed* through the hydrants or other outlets and *chlorinated* as prescribed in the AWWA Standard Specifications for Laying Cast-iron Pipe. Blowoff valves and waste lines sometimes are installed at low points in the system to permit flushing out accumulations of mud from time to time.

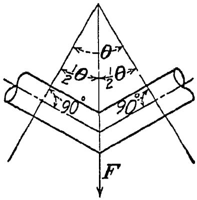
Anchors and Braces.—Wherever there is a dead end or a change in the direction of a pipe line made up with bell-and-spigot and similar pressure-packed joints, there is a pressure thrust tending to blow the joint apart. Since the centrifugal thrust due to water velocity usually is low with respect to pressure thrust, it often is neglected. Where desired, the centrifugal thrust can be computed from the formula

$$F = \frac{2Ayv^2}{g} \sin \frac{\theta}{2}$$

where F = the thrust, lb.
θ = the change in direction, deg.
A = the *inside* area of the pipe, sq ft.
y = the density of the fluid, lb per cu ft.
v = the velocity, ft per sec.
g = the acceleration due to gravity, *viz.*, 32.2 ft per sec.^{2}

The maximum amount of pressure thrust for different sizes of pipe and for various degrees of deflection for each 10 psi of internal pressure is shown in Table VII. The *actual* thrust will be decreased somewhat below the *maximum* computed thrust, depending on the holding power of the joint. In the case of lines carrying water or other liquids, however, allowance should be made for *shock* pressures due to water hammer (see pages 291 to 300) which may considerably increase the total applied pressure. In considering shock pressures, allowance should be made for the fact that, usually, they do not represent the continuous sort of loading associated with progressive settlement on earth support.

TABLE VII.—MAXIMUM SIDE AND END THRUSTS FROM PACKED PIPE JOINTS AT CURVES, FITTINGS, HYDRANTS, DEAD ENDS, AND THE LIKE[1]



Side thrust, lb, for each 10 psi internal pressure[2]

Outside diameter of pipe, inches[3]	180° ½ curve	90° ¼ curve	plug or 60° curve	45° ⅛ curve	30° $\frac{1}{12}$ curve	22½° $\frac{1}{16}$	11¼° $\frac{1}{32}$ curve		Per degree deflection[4] 0° to 20°
$F = F'$	×2.0	×1.414	×1.0	×0.764	×0.518	×0.390	×0.196	×0.098	×0.0174
3	142	101	71	54	37	28	14	7	1.2
4	252	178	126	96	65	49	25	12	2.2
6	566	400	283	216	146	110	55	28	4.9
8	1,006	710	503	385	261	196	98	49	8.8
10	1,570	1,110	785	600	407	306	154	77	13.7
12	2,260	1,600	1,130	863	585	440	221	111	19.6
14	3,078	2,173	1,539	1,174	796	600	301	151	26.7
16	4,022	2,842	2,011	1,537	1,042	784	394	197	35.2
18	5,090	3,600	2,545	1,947	1,320	994	499	254	44.3
20	6,284	4,445	3,142	2,400	1,630	1,227	616	308	54.7
24	9,048	6,400	4,524	3,460	2,345	1,760	886	444	78.8
30	14,138	10,000	7,069	5,400	3,665	2,750	1,387	694	122
36	20,340	14,400	10,170	7,770	5,270	3,970	1,980	990	177
42	27,708	19,600	13,854	10,600	7,170	5,400	2,715	1,358	241
48	36,192	25,580	18,096	13,820	9,370	7,050	3,544	1,772	315
54	45,804	32,390	22,902	17,500	11,850	8,930	4,485	2,243	398
60	56,548	40,000	28,274	21,600	14,640	11,000	5,540	2,770	488
72	81,360	57,500	40,680	31,100	21,100	15,880	7,970	3,985	708
84	110,836	68,250	55,418	42,300	28,700	21,600	10,850	5,425	964
96	144,764	102,300	72,382	55,250	37,500	28,200	14,176	7,090	1,260

[1] Computed by the formula $F = 2ap \sin \frac{1}{2}\theta$

where F = thrust, lb.

F' = thrust at dead ends, caps, and plugs, lb.

a = external cross-sect onal area of pipe, sq in., based on outside diameter.[3]

p = internal fluid pressure, psi.

θ = deflection angle, deg.

Side thrust from caps, plugs, short hydrant connections, and the like should be braced on both sides of the main. Caps or plugs on dead ends cause a corresponding end thrust.

[2] For any other internal pressure, multiply these side thrusts by the ratio of that pressure to 10 psi.

[3] The thrust in packed joints depends on *outside* diameter instead of on nominal *inside* diameter, of pipe.

[4] [illegible]sts per degree deflections for the range from 0° to 20° are practically constant.

Buried pipes receive considerable support from well-compacted earth in addition to what holding power the joints may have. Whereas this may suffice for low pressures and small degrees of curvature, additional support must be provided for packed joints in high-pressure mains and at elbows, dead ends, gates, hydrants, etc. Such support may be obtained through using bolted lugs, tie rods and clamps, or bracing in the form of wooden blocks, stone, or concrete which serve to spread the thrust against a larger area of earth. An example of the use of concrete and stone thrust blocks in connection with fire hydrants is given in Fig. 7.

According to the AWWA Standard Specifications for Laying Cast-iron Pipe reaction or thrust backing shall be applied on all pipe lines 8 in. in diameter or larger at all tees, plugs, caps, and at curves deflecting 22½ deg or more, or movement shall be prevented by attaching suitable metal rods or straps. The following instructions for bracing and anchoring cement asbestos pipe are quoted from the Johns-Manville installation manual:

Practically all thrusts exerted by pipe lines are horizontal or vertical. The soil at the bottom of the trench has greater unit bearing capacity to resist vertical thrusts downward than the soil at the side of the trench has to resist horizontal thrusts. This is because of the tendency of the side of the trench to heave or break out when sufficient horizontal pressure is exerted against it, while the bottom of the trench has less tendency to be displaced when downward pressure is exerted against it. The safe bearing loads given in the following table are for horizontal thrusts when the depth of cover over the pipe exceeds 2 ft.

Soil	Safe bearing load, lb per sq ft
Muck, peat, etc.	0
Soft clay	500
Sand	1,000
Sand and gravel	1,500
Sand and gravel cemented with clay	2,000
Shale	5,000

In muck or peat all thrusts should be resisted by piles or tie rods to solid foundations or by removal of muck or peat and replacement with ballast of sufficient stability to resist thrusts.

For vertical thrusts acting *downward*, the safe bearing loads of the various soils may be taken as twice those for horizontal thrusts.

For vertical thrusts acting *upward*, calculations are made to ascertain whether or not there is sufficient weight of pipe, water in pipe, and weight of projected volume of soil over the pipe line to resist upward movement. Where the soil will not compact over the pipe or where there is not sufficient weight of soil, concrete is cast around the pipe in sufficient weight and volume to counteract the thrust. Muck, peat, or water-saturated soft clay will not offer resistance to

movement, especially after being disturbed during the process of excavation for the pipe line and then backfilled.

An unbalanced thrust at a reducer should be resisted by a concrete thrust block around the reducer.

Where indicated to suit pressure conditions given in the following table, valves in Transite Pipe lines require anchorage against thrust as shown in Fig. 8.

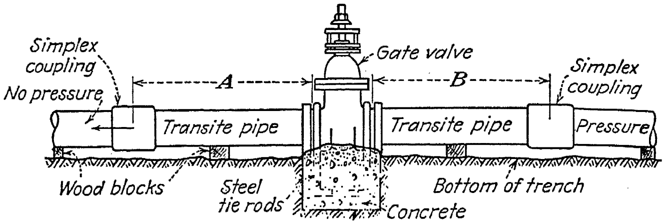


Fig. 8.—Anchor for bell-end gate valve in cement-asbestos pipe line.

Otherwise they may be left unanchored. Large valves inclosed in valve chambers can utilize the walls of the valve chamber as thrust block.

Working pressure, psi	Size of valve requiring anchorage
0 to 50	24 to 36″, inclusive
50 to 100	12 to 36″, inclusive
100 to 150	8 to 36″, inclusive
150 to 200	2 to 36″, inclusive

Note.—The above table is based on pipe well tamped in sandy clay with a minimum of 2 ft of cover over the pipe. Pipe in soft material should receive special consideration.

The safe bearing loads for different soils recommended by Johns-Manville include factors of safety of 2 to 4 from the allowable bearing loads sometimes given in structural handbooks, the values for soft clay having a factor of 4 and hard shale 2, with the other soil types intermediate.

In the case of a vertical concave curve, pressure thrust usually is resisted by firm soil which presumably remained undisturbed below the trench bottom and against which the pipe can push safely either with or without blocking. If the curve is convex, however, the thrust is resisted by backfilled earth and, particularly if the fill is shallow, some further anchorage may be needed. Under such conditions, heavy blocks of concrete or stone should be placed underneath the pipe and attached to it by suitable anchor bolts or clamps.

Pushing Services.—Considerable expense of excavation for installing services can be saved by using a pipe-pushing machine for inserting pipe between pits located some distance apart. The equipment and methods used are described under Gas Piping on pages 1184 to 1186.

WALL THICKNESS OF PIPE

The wall thickness of *under*ground water-supply piping must be sufficient to withstand internal working pressure, shock pressure due to water hammer, external loadings due to support and earth pressure, and jars from passing trucks and like disturbances. The wall thickness of pipes installed *above*ground must be sufficient to withstand internal working and shock pressures, plus such external loadings as may be imposed by expansion thrusts, the arrangement of supports, etc., as discussed elsewhere in this handbook. Thin-walled pipes that may be subjected to vacuum must be designed also to resist this form of collapsing pressure whether installed aboveground or underground. The design rules that have been evolved for meeting these conditions with pipe of different materials are presented in succeeding paragraphs.

Cast-iron Pipe.—Rules for the design of *underground* cast-iron pipe will be found in the American Recommended Practice Manual for the Computation of Strength and Thickness of Cast Iron Pipe, ASA A21.1, which is abstracted on pages 430 to 431. See page 1063 for computing Earth Loads on Pipe in Trenches. For computing the wall thickness of flanged, coupled, or bell-and-spigot cast-iron pipe used *aboveground,* safe rules will be found in the American Standard Code for Pressure Piping, ASA B31 (see pages 42 to 47). Considerable difference exists between the allowable stresses for the several varieties of cast-iron pipe such as pit-cast, horizontally cast, and centrifugally cast. In the design of cast iron it is customary to use the Dexter Brackett allowances for water hammer as given in the aforesaid manual and code. These allowances for water hammer plus loading from backfill and passing trucks were provided for in establishing the wall thicknesses for the standard ratings given in American Standard Specifications for Cast Iron Pit Cast Pipe for Water and Other Liquids, ASA A21.2. Specifications for protective coatings for cast-iron pipe will be found on pages 1270 and 1271.

Cement-asbestos Pipe.—This variety of pipe may be purchased to manufacturer's standards or to Federal Specification SS-P-351

(see pages 463 to 466) for pressure classes of 100, 150, and 200 psi water working pressure; there also is a manufacturer's Class 50. In addition to the usual hydrostatic pressure tests, the Federal specification prescribes flexure and crushing tests on sections of pipe which are intended to ensure satisfactory behavior under earth and truck loadings which can be computed as described on pages 1063 to 1067. The usual hydrostatic test pressures are set at $2\frac{1}{2}$ to 4 times the normal working pressure which ensures having the pipe designed strong enough to support overpressure due to water hammer.

Reinforced Concrete Pipe.—There is a tentative AWWA specification for reinforced-concrete pressure pipe (abstracted on pages 466 to 471) in addition to which there are individual manufacturer's specifications defining their own products in some detail and setting minimum wall thicknesses required for different diameters and types of reinforcement. Reasonable allowances for water hammer should be added to the working head and the sum of the two kept within the rated head for the particular class of pipe. Where large-diameter reinforced-concrete pipe is used at relatively low head, consideration should be given to providing water-hammer suppressors as discussed on page 1035 as a means of holding down costs.

Steel Pipe.—The wall thickness of steel pipe installed *above*ground may be determined according to the rules of the American Standard Code for Pressure Piping, ASA B31 (see pages 42 to 47). Under these rules the minimum wall thickness required is computed to suit the normal working pressure with an arbitrary thickness allowance for mechanical strength and/or corrosion. Owing to a liberal factor of safety between working pressure and bursting pressure and to the elastic nature of the material, no specific allowance for water hammer is required by the Code. No distinction is made between bare pipe and that which is coated in some way for resisting corrosion.

Pipe installed on piers or other supports *above*ground, or on bridges, or hung from above, must be capable of acting as a beam to support the weight of the pipe and its contents. In the case of thin-walled pipe, reinforcing rings or other forms of stiffners may be required at supports. Sometimes for long spans, additional wall thickness must be provided above that required to withstand internal pressure. Information on supporting pipes will be found on pages 744 to 753.

Wall thicknesses of steel pipe for *under*ground water service are listed in AWWA Standard Specifications for Electric Fusion Welded Steel Water Pipe, 7A.3 and 7A.4 (abstracted on pages 406 to 416) with corresponding maximum working pressures for a factor of safety of 4 with plate having a tensile strength of 50,000 psi. In determining actual working pressures the Dexter Brackett (see page 299) or other appropriate allowances for water hammer should be deducted from the maximum working pressures. No corrosion allowance is stipulated in these AWWA specifications but, unless otherwise ordered, a protective coating is required in accordance with AWWA Specification 7A.6 (see pages 1264 to 1270). See pages 1067 to 1070 for computing Earth Loads on Pipe in Trenches.

A study of the depth of corrosion pits in buried steel pipes has been made with a view to determining how this affects the life of the pipe and what amount of extra metal should be allowed, if any.[1] Authorities differ on this score, some holding that an arbitrary corrosion allowance of $\frac{1}{16}$ in. should be added to the wall thickness in all cases, whereas others contend that no allowance is needed if the best protective coatings are used. Where long life is desired, the authors of this handbook feel that some corrosion allowance is desirable until more experience is had with the coatings.

Collapsing Pressure.—Thin-walled steel pipe which is, or may be, subjected to *vacuum* should be designed to resist collapsing pressure as described on pages 35 to 37. With true syphons and suction pipes for pumps, a condition of partial vacuum will exist in normal operation, while pressure pipe installed in rolling country may be subjected to vacuum collapsing pressure if flow is interrupted for any reason. Where the diameter D is too great for the wall thickness t, collapse under the aforesaid conditions can take place on a grand scale as occurred in the classic case of the Los Angeles aqueduct where miles of 10-ft-diameter pipe having a wall thickness of $\frac{1}{4}$ to $\frac{5}{16}$ in. collapsed owing to a local washout, while another section $\frac{3}{8}$ in. thick withstood the vacuum successfully. With pipe of this diameter-thickness ratio a vacuum of only $\frac{1}{2}$ to 1 psi could produce collapse. When the damaged sections at the washout were replaced and water pressure turned

[1] See "A Method of Determining Wall Thicknesses of Steel Pipe for Underground Water Service," by Russel E. Barnard, *Jour. AWWA*, Vol. 29, pp. 791–807, 1937.

on again, the collapsed sections resumed their former cylindrical form.

Collapse from atmosphric pressure is a possibility to be considered where the D/t ratio of the pipe is greater than about 100. The amount of initial ellipticity present in the pipe enters into the problem, and it is recommended that a factor of safety of 2 to 4 be allowed beyond the usual supposition of 1 per cent ellipticity. Referring to the tables on pages 35 to 37, this can be done by making the pipe wall thick enough to give a D/t ratio that can support two to four times the maximum collapsing pressure (vacuum) that may be applied to the line. Maximum practical suction lifts in feet with corresponding atmospheric pressure differentials tending to create collapse of the pipe in pump suctions and true syphons are given in Table VIII.

TABLE VIII.—MAXIMUM PRACTICAL SUCTION LIFTS AND CORRESPONDING PRESSURE DIFFERENTIALS AT VARIOUS ELEVATIONS ABOVE SEA LEVEL

Elevations above sea level, ft.	Sea level	2,000	4,000	6,000	8,000	10,000	12,000
Maximum suction lift, ft.	25	23	22	20	19	17	16
Corresponding pressure differential, psi	10.8	10.0	9.5	8.6	8.2	7.3	6.8

In rolling or mountainous country, air and vacuum valves should be placed at the summits of water-supply lines to allow air to escape when they are being filled, as well as to admit air to prevent forming a vacuum when they are being drained. It is good practice to install such valves at abrupt breaks in grade where they serve also to permit the escape of air while the line is operating under pressure, thus preventing the accumulation of air that is released from solution in the water. The escape of air in pressure air valves usually is controlled by a ball or float attached to a lever connected to the stem of a needle valve. On larger lines, separate valves may be employed for air release and for vacuum prevention; for smaller pipes, a combination valve usually is provided that incorporates both these features. Typical installation details for a large vacuum valve are shown in Fig. 9. In cold climates due consideration should be given to placing such valves in positions where they can be protected from freezing.

Correct sizes of vacuum valves can be determined only by computation of the probable rate of water drainage from the line, which depends, in turn, on its profile and on the rate at

air is admitted to break the vacuum. The size of the vacuum valve should be computed on the supposition that, for some assumed vacuum inside the pipe, air must enter at the same volumetric rate as water leaves in order to prevent exceeding the assumed vacuum. A cut-and-try method of solution usually is necessary to obtain a satisfactory balance. The assumed vacuum must not exceed that which the pipe can safely support from a consideration of collapsing pressure (see pages 35 to 37).

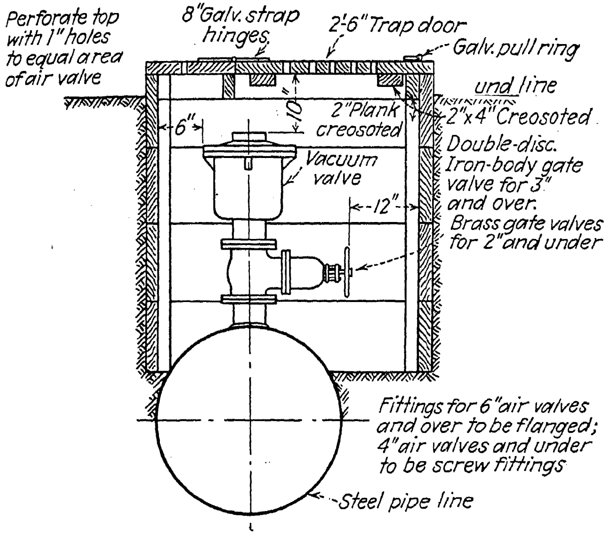


FIG. 9.—Typical installation details for a large vacuum valve on water-supply piping in hilly country.

Sizing Air and Vacuum Vents.—The size of air and vacuum vents required depends on the diameter of pipe and the slope between pockets as well as on the length of time that can be devoted to filling the line. Recommended sizes are given in the chart of Fig. 10 which is reproduced by permission from the "Handbook of Welded Steel Pipe."[1] For example, when the slope S is 0.0192 and the pipe is 24-in.-diameter 10-gauge steel, enter the chart on the left for a value of $S = 0.0192$. Follow across horizontally until the 10-gauge line is reached, then

[1] Published by the California Corrugated Culvert Co. of Berkeley and Los Angeles, Calif.

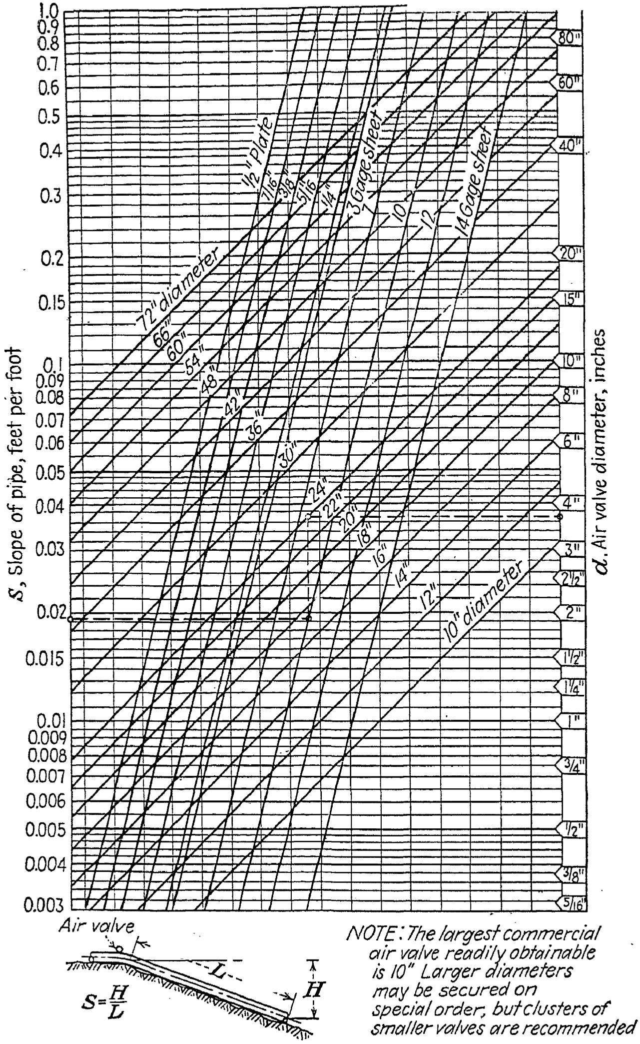


FIG. 10.—Chart for determining recommended sizes of air and vacuum vents on steel water-supply pipe lines. (*From Handbook of Welded Steel Pipe*, 1942.)

vertically to the 24-in.-diameter line, then horizontally to the right-hand scale where it will be found that a 4-in.-diameter vent will be required. For a given diameter of pipe the allowable working pressure depends, of course, on the wall thickness and at higher pressures the rate of escape through a given diameter air vent will increase. Likewise, the thicker pipe can support a greater external pressure under vacuum conditions. Hence, had the 24-in.-diameter pipe of the example had a ¼-in. wall thickness, a 2½-in.-diameter air vent could be assumed to produce an equivalent result at a correspondingly higher pressure.

Earth Loads on Pipe in Trenches.—The following statement of the theory of earth loads on pipe in trenches is abstracted from bulletins[1] of the Engineering Experiment Station of Iowa State College at Ames, Iowa. The original concept of the load theory, which was developed by Dean Anson Marston about 1907, utilized many of the principles of Janssen's formula for pressures in grain bins. The load theory was derived in response to a growing realization of the need for a reliable means for calculating the loads upon sewers and drains, or for rigid clay and concrete pipe in ditches of relatively narrow width. However, this theory of loadings and the later studies of special cases are equally applicable to cast-iron and steel pipe in trenches. The theory is strongly supported, and with remarkable unanimity, by a long series of experimental researches and widespread examination of actual conduits in use.

The general ditch conditions on which the mathematical theory is based are shown in Fig. 11. Any pipe in a ditch or trench may be considered as made up of four equal sectors of 90 deg each, two horizontal and two vertical, as shown in the figure. The two vertical sectors, owing to their position, carry only a negligible amount of the load. The foundation pressure is distributed over the bottom 90-deg sector with greater or less uniformity, depending on the care in pipe-laying. The load on the pipe, due to or transmitted through the ditch filling, rests almost entirely on the top 90-deg. sector, with intensity somewhat greater at the center than at the sides.

[1] This information has been published in a succession of Iowa State College bulletins concerning the supporting strength of sewer tile and cast-iron pipe of which the following are suggested for reference on computing earth loadings:

(*a*) *Bull.* 31, "The Theory of Loads on Pipes in Ditches, and Tests of Cement and Clay Drain Tile and Sewer Pipe."

(*b*) *Bull.* 47, "The Supporting Strength of Sewer Pipe in Ditches and Methods of Testing Sewer Pipe in Laboratories to Determine Their Ordinary Supporting Strength," by A. Marston, W. J. Schlick, and H. F. Clemmer, Oct. 10, 1917.

(*c*) *Bull.* 96, "The Theory of External Loads on Closed Conduits in the Light of the Latest Experiments," by Anson Marston, Feb. 19, 1930.

It has been proven by actual experiment and by careful study of pipe in actual ditches that the pipe in the ditch is so much more rigid than the ditch filling at each side of the pipe (between the two vertical 90-deg sectors and the sides of the ditch), even with careful tamping that the pipe carried practically all of the vertical load at the level of the top of the pipe due to or transmitted through the ditch filling. The load may be said to "arch over" between the pipe and the sides of the ditch at about the extremities of the 90-deg sector, at least for all widths of ditches which are common in pipe sewer construction. In ditches which are extremely wide as compared with the diameter of the pipe the ditch filling at the sides will carry an appreciable part of the load, but it is safer not to count on any such precarious additional support.

It has been proven by experiment and by careful study of pipes in actual ditches that the width of the ditch at the level of the top 90-deg sector (see dimension B in Fig. 11) is the width factor which affects the load on the pipe. The ditch may widen out indefinitely above this level without increasing the load on the pipe appreciably. All filling in such added width will be carried on the solid soil beneath, instead of on the pipe. This filling is shallower than the filling over the pipe and settles less, and hence frictional resistance develops along the same vertical lines, as shown in Fig. 11, and thus will help carry part of the weight of the rest of the ditch filling, just as in the case of vertical sided ditches.

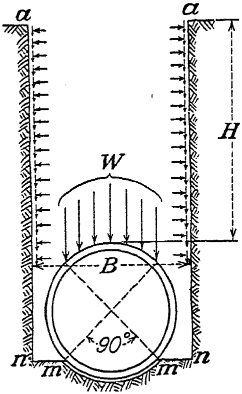


FIG. 11.—Typical condition of bedding and loading of pipes in trenches.

It has been proven by experiment and by careful study of pipe in actual ditches that it is the width B of the ditch a little below the top of the pipe which determines the total load on the pipe, and not the diameter of the pipe itself. For example, in one experiment two pipes of different diameters were tried successively in the same ditch and were found to be subjected to the same load. Sewer pipe have been found cracked in the wide portions and sound in the narrow portions of the same ditch. Pipe from the same factory have been found to crack in wide ditches and remain sound in narrow ditches in the same drainage districts. Hundreds of calculations of loads on pipes in ditches and comparison of the results with the results of experiments and with the condition of the pipe in actual sewers and drains have shown the absolute reliability of a formula for loads on pipes in ditches which does not include the diameter of the pipe as a factor. Of course, the diameter of the pipe generally determines the approximate width of the ditch at a level a little below the top of the pipe, and in this way affects the load on the pipe.

The ditch filling above the top of the pipe settles relative to the sides of the ditch, thereby developing frictional resistances, as shown in Fig. 11, which carry part of the weight of the ditch filling. The total weight of the ditch filling at the time when the maximum load on the pipe occurs consists of two parts, one carried by the pipe and the other by the frictional resistances. At all other times than those of occurrence of maximum loads an additional part of the total weight

is carried to the sides of the ditch by cohesion, thus lightening the load on the pipe. In the long time test at Ames, Iowa, the loads between maxima sometimes became as low as 65 per cent of the maximum, but they averaged about 75 per cent between maxima. Maximum loads may reoccur from time to time, with changes in the condition of ditch filling, especially as to degree of saturation. One is concerned mainly with the probable maximum loads in designing pipe sewers as to structural strength rather than the loads between maxima.

The following mathematical formula has been developed for maximum loads on pipes in ditches based on the conditions shown in Fig. 11. The details of the mathematical work will be found in *Bull.* 31, of the Iowa Engineering Experiment Station, already referred to. The resulting formula is

$$W = CwB^2$$

where W = load on pipe in ditch, lb per lin ft.

C = coefficient of loads on pipes in ditches, to be taken from Table IX.

w = weight of ditch filling material, lb per cu ft (usually 110 to 130).

B = breadth of ditch a little below top of pipe, ft.

The weights per cubic foot of frequently encountered types of backfill are as follows:

Kind of Backfill	w = Weight in Lb per Cu Ft
Loam, sandy loam	110
Sand	115
Gravel	125
Clay, ordinary maximum load	120
Sandy or gravelly clays	120
All clays, saturated	130

From this formula and the coefficients given in Table IX the earth loads on pipe in trenches can be computed readily for the conditions commonly encountered in sewer or other underground pipe construction. For unusual conditions, such as pipes projecting out from under an embankment or laid in trenches with wide sloping sides, reference should be made to the Iowa State College bulletins or to the American Recommended Practice Manual for the Computation of Strength and Thickness of Cast Iron Pipe, ASA A21.1. The latter contains a convenient set of charts from which earth loadings can be read directly for conditions usually encountered.

The loads first imposed by backfilling are not as great as may be experienced later when the material becomes completely saturated and thoroughly compacted. Under favorable conditions individual stretches of pipe may escape being subjected to the theoretical maximum load but, since there always is a strong probability of such exposure, the pipe selected should be strong enough to carry the load without cracking or crushing and still have a considerable margin of safety, say 50 to 150 per cent beyond minimum theoretical requirements based on the best bedding conditions and no settlement.

Referring to Table IX, it is of interest to note that, depending on soil conditions, the values of C do not increase substantially for height-breadth ratios exceeding 5 to 10, and that C is assumed practically to cease increasing for ratios above 15. Or, in other words, after reaching certain proportions, the backfill tends to support itself against the sides of the trench with further increase in height.

TABLE IX.—SAFE WORKING VALUES FOR THE COEFFICIENT C TO USE IN FORMULA $W = CwB^2$ FOR CALCULATING LOADS ON PIPES IN TRENCHES

Ratio H/B = height of fill above top of pipe to breadth of ditch a little below the top of the pipe	Safe working values of C: Minimum possible without cohesion. These values give the loads generally imposed by granular filling materials before tamping or settling	Maximum for ordinary sand. [illegible] ordinary [illegible] of sand filling	Completely saturated top soil	Ordinary maximum for clay (thoroughly we[illegible] for all ordinary cases of clay filling	Extreme maximum for clay (completely [illegible] only for extremely unfavorable conditions
0.5	0.455	0.461	0.464	0.469	0.474
1.0	0.830	0.852	0.864	0.881	0.898
1.5	1.140	1.183	1.208	1.242	1.278
2.0	1.395	1.464	1.504	1.560	1.618
2.5	1.606	1.702	1.764	1.838	1.923
3.0	1.780	1.904	1.978	2.083	2.196
3.5	1.923	2.075	2.167	2.298	2.441
4.0	2.041	2.221	2.329	2.487	2.660
4.5	2.136	2.344	2.469	2.650	2.856
5.0	2.219	2.448	2.590	2.798	3.032
5.5	2.286	2.537	2.693	2.926	3.190
6.0	2.340	2.612	2.782	3.038	3.331
6.5	2.386	2.675	2.859	3.137	3.458
7.0	2.423	2.729	2.925	3.223	3.571
7.5	2.454	2.775	2.982	3.299	3.673
8.0	2.479	2.814	3.031	3.366	3.764
8.5	2.500	2.847	3.073	3.424	3.845
9.0	2.518	2.875	3.109	3.476	3.918
9.5	2.532	2.898	3.141	3.521	3.983
10.0	2.543	2.918	3.167	3.560	4.042
11.0	2.561	2.950	3.210	3.626	4.141
12.0	2.573	2.972	3.242	3.676	4.221
13.0	2.581	2.989	3.266	3.715	4.285
14.0	2.587	3.000	3.283	3.745	4.336
15.0	2.591	3.009	3.296	3.768	4.378
Very great	2.599	3.030	3.333	3.846	4.545

W = load on pipe, lb per lin ft. C = coefficient. w = weight of ditch-filling material, lb per cu ft. B = breadth of [illegible] pipe, ft.

Supporting Strength of Cast-iron Pipe.—The amount of earth load to which a cast-iron pipe can be subjected without damage depends to a considerable extent on the amount of settlement to be expected as well as on how the bed of the trench is prepared and whether blocking is used under the pipe (see pages 430, 431 for what are designated "field conditions"). The strength of cast-iron pipe when subjected to simultaneous external loads and

internal pressure was investigated at the Iowa Engineering Experiment Station for ASA Sectional Committee A21 on Specifications for Cast Iron Pipe and Fittings.[1] The findings of this investigation are the basis for the "American Standard Specifications for Cast Iron Pit Cast Pipe for Water or Other Liquids, ASA A21.2" (abstracted on pages 432 to 437), and for other specifications being formulated by Sectional Committee A21.

Behavior of Steel Pipe in Trenches.—The following analysis of the behavior of steel pipe in trenches, contributed by Russell E. Barnard,[2] is based on tests made by Prof. Spangler of Iowa State College.[3] While Prof. Spangler's work was done on *corrugated* steel pipe, the thin-ring analysis used in correlating the results seems equally applicable to *smooth* steel pipe when due allowance is made for the difference in moment of inertia per linear inch of pipe wall.

Loads on rigid pipe under given conditions have been found to be greater than for flexible pipe under the same conditions. This occurs because the flexible pipe, by yielding vertically and deflecting horizontally, permits a portion of the vertical load which would be carried by a more rigid pipe to be transferred to the earth column in the fill or trench adjacent to the pipe. At the same time the flexible pipe maintains stability by transforming vertical load into horizontal thrust. In view of these facts, the structural design of flexible pipe must be approached on the basis of thin-ring elastic analysis rather than the rigid-ring analysis referred to in the preceding section on Supporting Strength of Cast-iron Pipe. The following general formula for calculating the deflection of *flexible* pipe under embankments and in trenches has been developed by Spangler:

$$d = \frac{fKW_c r^3}{EI + 0.061er^4}$$

where d = initial vertical or horizontal deflection or diameter change, in.
K = a bedding constant.
W_c = fill load per linear *inch* of pipe, lb.
W = fill load per linear *foot* of pipe, lb.
r = mean radius of pipe, in.
t = pipe wall thickness, in.
E = modulus of elasticity of pipe metal, psi (30,000,000 for steel).

[1] See: (*a*) "Supporting Strengths of Cast-iron Pipe for Water and Gas Service," by W. J. Schlick, *Bull.* 146, Iowa State College, 1940.

(*b*) American Recommended Practice Manual for the Computation of Strength and Thickness of Cast Iron Pipe, ASA A21.1. (abstracted on pages 430 to 431).

[2] Spiral Welded Pipe Department, The American Rolling Mill Company, Middletown, Ohio.

[3] See "The Structural Design of Flexible Pipe Culverts," by M. G. Spangler, 1941, *Bull.* 153, Iowa Engineering Experiment Station, Iowa State College, Ames, Iowa.

e = modulus of passive pressure of the sidefill material, psi per linear inch of pipe.

I = moment of inertia per linear inch of pipe wall, in.^{4}

f = deflection lag factor.

The moment of inertia per linear inch of pipe wall used in the thin-ring analysis should not be confused with the moment of inertia of the pipe cross section used elsewhere in this handbook. In the thin-ring analysis, I is the moment of inertia of a rectangle and denotes a property of the pipe wall rather than of its cross section. For a *smooth* pipe $I = lt^3/12$ expressed in in.^{4}, where l is measured in inches along the pipe parallel to its axis. When expressing terms as "per linear inch of pipe," l is unity and the moment of inertia becomes $I = t^3/12$. For a *corrugated* culvert pipe having corrugations about ½ in. deep spaced about 2⅔ in. center to center, $I = t/30$, approximately.

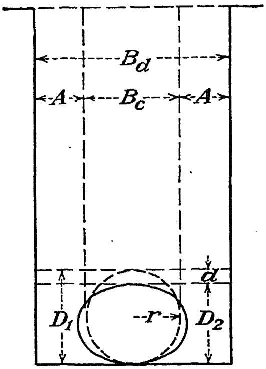


FIG. 12.—Behavior of steel pipe in trenches.

Using the above relation of I and t, the expression for *smooth* pipe can be simplified to

$$d = \frac{fKWr^3}{Et^3 + 0.732er^4}$$

where W is in pounds per linear *foot* of pipe as computed by the Marston methods (see ASA A21.1, or page 1063) and modified as explained in connection with Fig. 12 by multiplying his W by the ratio of the outside diameter of the pipe to the width of the trench as explained in the next paragraph. The initial deflection usually corresponds to a lag factor of $f = 1$. Final deflection which occurs after a long period of time may be 1.25 to 1.50 times the initial deflection. The multiplier 1.25 to 1.50 applied to obtain final deflection is termed the "deflection lag factor."

In the Spangler formula the fill load is calculated according to Marston's methods (see page 1063). Where the conduit is relatively flexible, however, a part of this total calculated load may be considered carried by the sidefill material pushing against the trench walls as shown in Fig. 12. As a minimum loading condition, if the pipe and sidefills A below the top of the pipe are of the same degree of flexibility, then the pipe and the sidefills would each carry the same

TABLE X.—DEFLECTION OF STEEL PIPE UNDER EXTERNAL LOAD IN TRENCHES[1]

Pipe O.D., inches	2% of O.D., inches	Wall thickness, inches		d = deflection in inches			
				5-ft cover		10-ft cover	
		Fraction	Decimal	No truck	With truck	No truck	With truck
6	0.12	1/8	0.1250	0.01	0.02	0.02	0.02
		3/16	0.1875	0.00	0.01	0.01	0.01
8	0.16	1/8	0.1250	0.03	0.07	0.06	0.07
		3/16	0.1875	0.01	0.02	0.02	0.02
10	0.20	1/8	0.1250	0.07	0.17	0.15	0.18
		3/16	0.1875	0.03	0.06	0.05	0.07
12	0.24	1/8	0.1250	0.15	0.32	0.27	0.33
		3/16	0.1875	0.06	0.12	0.10	0.12
		1/4	0.2500	0.03	0.06	0.05	0.06
14	0.28	1/8	0.1250	0.21	0.48	0.42	0.52
		3/16	0.1875	0.09	0.21	0.19	0.24
		1/4	0.2500	0.04	0.10	0.09	0.11
16	0.32	1/8	0.1250	0.30	0.60	0.56	0.68
		3/16	0.1875	0.15	0.31	0.28	0.35
		1/4	0.2500	0.08	0.16	0.14	0.18
18	0.36	3/16	0.1875	0.21	0.41	0.39	0.47
		1/4	0.2500	0.12	0.22	0.21	0.26
		5/16	0.3125	0.07	0.13	0.13	0.15
20	0.40	3/16	0.1875	0.28	0.53	0.49	0.59
		1/4	0.2500	0.17	0.31	0.29	0.35
		5/16	0.3125	0.10	0.19	0.18	0.21
24	0.48	3/16	0.1875	0.41	0.75	0.71	0.83
		1/4	0.2500	0.29	0.54	0.50	0.59
		3/8	0.3750	0.14	0.24	0.23	0.27
30	0.60	1/4	0.2500	0.40	0.71	0.70	0.83
		5/16	0.3125	0.31	0.54	0.54	0.64
		3/8	0.3750	0.23	0.40	0.40	0.48
36	0.72	1/4	0.2500	0.48	0.83	0.87	1.02
		5/16	0.3125	0.41	0.71	0.75	0.87
		3/8	0.3750	0.34	0.58	0.62	0.72
42	0.84	1/4	0.2500	0.53	0.98	0.95	1.14
		3/8	0.3750	0.44	0.80	0.78	0.92
		1/2	0.5000	0.32	0.57	0.57	0.68
48	0.96	5/16	0.3125	0.52	0.87	0.94	1.09
		3/8	0.3750	0.48	0.81	0.87	1.01
		7/16	0.4375	0.43	0.73	0.78	0.91
		1/2	0.5000	0.38	0.65	0.70	0.81
60	1.20	3/8	0.3750	0.57	0.98	1.05	1.22
		1/2	0.5000	0.51	0.87	0.95	1.10

[1] Computed by the methods outlined by Russell E Barnard (see text), for *smooth* steel pipe under the following conditions:

$$e = 30 \quad f = 1.5 \quad K = 0.10$$

Earth loads and truck loads similar to those used by ASA Sectional Committee A21 (see pp. 430, 431 and ASA A21.1), but with earth loads reduced in B_c/D_d ratio as explained in text.
Trench width = pipe O.D. plus 2 ft.
Field conditions: flat-bottomed trench, tamped backfill.

amount of the load per unit of width. Under these conditions the load on the pipe may be determined by multiplying Marston's load expression by the ratio of the outside diameter of the pipe to the width of the trench before substituting in the Spangler formula. Referring to Fig. 12, this ratio is B_c/B_d. The maximum loading condition developing on a pipe in a trench is given by the *unmodified* Marston formula. The probable load on *rigid* pipe will approach this maximum figure; the probable load on a *flexible* pipe will approach the minimum figure given by the *modified* Marston formula.

The value of e, the modulus of passive pressure of the side-fill material, varies with type of soil and degree of compaction. It averages on the order of from 10 to 15 for embankments, wide ditches, and untamped backfill, to 30, 40, or more for tamped backfill in common-width vertical-side trenches.

The value of the *bedding* constant K depends on whether the pipe is laid in a flat-bottomed trench or whether the bottom of the trench is shaped to fit the pipe to some specified degree. Values of K for different degrees of bedding contact are as follows:

Bedding contact in degrees of pipe circumference	Bedding constant, K
0	0.110
30	0.108
45	0.105
60	0.102
90	0.096
120	0.090
180	0.083

Assuming ordinary bedding conditions and tamped backfill in a vertical-wall trench 2 ft wider than the pipe, the probable expected final average diameter deflection of steel pipe is given in Table X. Design loads used in computing tabular values are the same as those developed by A.S.A. Sectional Committee A21 for *rigid* pipe (see ASA A21.1) and then modified for *flexible* pipe as previously explained.

Deflection of flexible pipe approaches 20 per cent before collapse occurs. A permissible figure of 2 per cent (as shown in Table X) often is mentioned for water lines, but many steel pipe lines, especially those of large diameter, are successfully operating with deflections of 5 per cent or more.

When placing steel pipe underground, the bed of the trench should be cleared of rocks and boulders and the backfill tamped to at least the top of the pipe. Side support is best obtained by *hand* tamping. In case excessive deflections are feared with thin-walled steel pipe of large diameter, it may be advisable to provide external stiffening rings, or to put struts, ties, or continuous girders inside the pipe. Thin-walled steel pipe of smaller diameter can be protected where necessary under embankments and in other hazardous locations by placing it inside corrugated culvert pipe capable of supporting any load expected.

PROTECTION AGAINST FREEZING

Water in pipes should not be allowed to freeze solid under any conditions, both to avoid interrupting service and to prevent damaging the pipe. In freezing solid in a confined space water

expands about 10 per cent in volume with an irresistible pressure which no pipe material can support without being permanently stretched or burst. In some cases, however, such as the 3- to 6-ft diameter risers or standpipes used in cold climates with elevated storage tanks, it is expected that a concentric sheeting of ice will form on the inside of the pipe, while the liquid core will continue operation without interruption and at the same time prevent damage to the riser. Other means to prevent freezing which are described in succeeding sections include velocity and insulation for outdoor pipes and depth of bury for underground pipes. Methods for thawing frozen pipes are described. Pipe lines used in irrigation work can be drained at the end of the irrigating season so as to eliminate danger of freezing.

Effect of Velocity and Insulation.—The velocity of flowing water, if uninterrupted, will of itself afford some protection against freezing outdoor pipe if the line is not too long. According to the "Handbook of Welded Steel Pipe" (California Corrugated Culvert Co.) water flowing at a velocity of 4¼ ft per sec will not freeze in an unprotected pipe at a temperature of 32 F. With lower outside temperatures water will freeze irrespective of the velocity if the line is long enough in relation to its diameter. A chart is given in that handbook to relate the length-diameter relation of the line with freeze-prevention velocity for a succession of outdoor temperatures down to −8 F. With 2 in. of hair-felt insulation, it is found by test to require approximately five times as long to freeze 4- to 12-in.-diameter pipes when subjected to the same conditions, *viz.*, with no water flowing and the temperatures about the same.

The rate of freezing of water when motionless in ar ·utdoor pipe can be computed for bare pipe and for different sses of insulation by formulas presented in an AW
Bleich.[1] First the time required to chill the wa
from some higher initial temperature is comp
time for the heat of fusion to be absorbed
is given in the chapter on Insulation o
Bleich investigated the freezing of sr
exposed to outdoor air and found th

[1] See "Protection of Water Services
Jour. AWWA, Vol. 18, pp. 564–573
H. E. Babbitt and J. J. Doland, M
1939.

on a 2-in. pipe is more effective than 12 in. of the same insulation on a ¾-in. or 1-in. pipe. Hence he concluded that service pipes in exposed locations should be of at least 2-in. nominal size and well insulated besides. A 1-in. pipe insulated with 3-in. hair felt took three times as long to freeze as 1-in. bare pipe, both with no flow velocity.

The general methods for computing heat loss and selecting insulation for pipes are given in Chap. V on Heat Insulation. Insulation like hair felt on outdoor pipes should have a weatherproof wrapping of tarred roofing felt or similar material (see note 2, Table IX, page 720). In the western United States and other places where wood is abundant, water pipes sometimes are protected against freezing with sawdust or granulated insulation encased inside wood sheathing put together with tongue-and-groove joints. Even with water-supply systems that are classed as underground, there are apt to be occasional locations, such as river crossings on bridges, in which a considerable run of pipe has to be outdoors where it needs insulation against freezing. Under these circumstances it is desirable to see that there is sufficient circulation through the line in cold weather to prevent freezing through the insulation.

With elevated water-storage tanks two possibilities exist in cold weather: (1) If there is no appreciable flow in or out of the tank to keep the water temperature above freezing and forestall any tendency to form solid ice, either the tank should be drained for the winter or some means of heating and circulating the water in the riser and tank should be provided. Under these circumstances it does not matter particularly whether the standpipe riser is oversize or of normal pipe diameter. (2) If there is a reasonable amount of flow in and out of an elevated tank, a standpipe riser of 3- to 6-ft diameter can be employed to prevent freezing solid, or a normal pipe diameter riser can be used and the water heated and circulated through the tank and riser. For reasons of economy it is usual to keep down heat losses by heating the water to a temperature only slightly above freezing, say 35 to 40 F.

Depth of Bury.—As a protection against freezing, underground pipes should be laid well below the maximum *frost* line. considerations, such as lessened impact shock from trucks ater uniform ground temperature, point the need for a conditions, po[illegible]ount of cover over buried pipes. Heavy motor damaging the pipe. [illegible] develop impacts and vibrations that may

cause joints to leak or be otherwise destructive to buried water lines. A cover of about 4 ft with a flat-bottomed trench and tamped backfill usually results in the most favorable combination of earth loading and truck hazard (see American Recommended Practice Manual for the Computation of Strength and Thickness of Cast-Iron Pipe, ASA A21.1). At lower depths the ground temperature becomes more uniform, while near the surface a difference of 1 or 2 ft in depth has an appreciable effect on contraction and expansion of the line and is measurable in the water temperature. The subject of year-round soil temperatures and depth of frost penetration was investigated by S. G. Highland and reported to the American Water Works Association.[1] The following conclusions are summarized from his work and other available date.

Even in warm climates a minimum of 2 to 3 ft of cover is required for mechanical protection and to minimize expansion and change in water temperature. In cold climates more depth of bury is needed for protection against freezing. Dead ends and services where there may be no flow for long periods require more cover than do distribution networks where some flow goes on at all times. Care should be exercised to see that the goosenecks where services connect to mains do not arch up into the frost zone. Frost penetration reaches lowest depth under dirt roadways where the soil is firmly compacted. It is less under pavements which afford a limited amount of insulation, and below grass where the soil is looser and some protection may be afforded by the grass or snow which remains there. Frost penetration is greatest in wet clay soils, and considerably less in sandy soils.

Along the Canadian border at a latitude of 45 to 50 degrees (except in the Pacific Northwest where a milder climate obtains) frost may penetrate to a depth of 8 to 12 ft, and a cover of 8 or 9 ft often is required while 6 to 8 ft certainly would be needed. In such extremely cold regions it may be advisable to install insulated boxing around services from the main into the building instead of excavating to the full depth required to get below frost. From the 40th to the 45th parallel, 4 to 5 ft of cover is sufficient. South of 40-deg latitude, 2½ to 4 ft of cover usually is ample.

In laying underground water mains below dirt streets, which may be regraded later if paved, consideration should be given to

[1] "Study of Year Round Soil Temperatures," by S. G. Highland, *Jour. AWWA*, Vol. 16, No. 3, pp. 342–354, 1926.

keeping under the frost line corresponding to final grade. The same holds true in laying mains in other places where grade may be changed for any reason.

Thawing Frozen Pipes.—As soon as stoppage of flow through a pipe owing to water freezing is noticed, steps should be taken to thaw it out immediately in the hope of preventing serious damage to the pipe. If flow is restored before solid ice forms all through the pipe, it may be possible to prevent bursting the pipe or to minimize the extent of the damage. The use of electricity for thawing frozen services or house plumbing is described on pages 990 to 992. In the case of larger pipes and street mains, it may be necessary to dig down to the pipe and build a fire over it in the trench. If the pipe is exposed, it can be thawed with a gasoline torch, by pouring hot water over rags wrapped around it, or by blowing low-pressure steam on or into the pipe.

WATERWORKS PRACTICE

Among the available publications describing waterworks practice in America are "Manual of Waterworks Practice," by the American Waterworks Association; *Proceedings American Waterworks Association; Proceedings New England Water Works Association;* "Water Supply Engineering," by H. E. Babbitt and J. J. Doland (McGraw-Hill Book Company, Inc.); "Conveyance and Distribution of Water," by Edward Wegmann (D. Van Nostrand Company Inc.); "Waterworks Handbook," by A. D. Flinn, R. S. Weston, and C. L. Bogert (McGraw-Hill Book Company, Inc.). Since practice west of the Rockies differs considerably from the rest of the United States, the following handbooks, published by the California Corrugated Pipe Company of Berkeley and Los Angeles, Calif., are suggested for additional references on Western construction: "Handbook of Welded Steel Pipe," and "Handbook of Water Control." Considerable valuable data can be obtained from the annual reports of water departments in the larger cities.

Water Consumption and Demand.—Average values of per capita water consumption for homes and for the town as a whole are given on page 984. The latter are considerably higher owing to the amount of water used for public, industrial, and commercial purposes and to that lost through leakage. Where water is metered, this tends to reduce the consumption through eliminating

unnecessary wastage. Detailed information of the per capita water consumption of individual cities will be found in the *Proceedings* of the American Water Works Association,[1] the *Journal* of the New England Water Works Association,[2] and elsewhere.[3]

The use of water varies according to the season, the day of the week, and the hour of the day or night. Consumption tends to be greater during the day than at night, and summer demands are notably higher than winter owing to lawn sprinkling and other seasonal uses. These fluctuations may be taken care of to a certain extent by reservoirs and reserve pumping capacity, or by elevated storage tanks. The distributing system should be large enough to meet the maximum consumer demand and at the same time furnish all the water needed for extinguishing fires. The total amount of water used in a year at fires usually is a negligible part of the total consumption, but during a fire the rate of demand may be so great (see page 1077) as to be the deciding factor in determining the capacities of pumps, storage facilities, and distributing mains in all but the largest cities.

Per capita daily consumptions usually are given in terms of yearly average figures. Maximum and minimum rates of demand frequently are expressed in terms of the average rate. A water system must be capable of meeting peak-load requirements as well as operating economically at lesser rates of demand. Data on consumption and demand often can be obtained from the annual water department reports of various cities. Where specific data are not available, the maximum hourly rate of pumpage for cities may be taken roughly as $2\frac{1}{2}$ times the yearly average rate; the maximum daily pumpage as $1\frac{1}{2}$ times the daily average throughout the year; and the maximum monthly pumpage as $1\frac{1}{4}$ times the average monthly pumpage throughout the year. From 20 to 40 per cent of the water pumped may be lost through leakage or unaccounted for in meter readings.

Distribution System Design.—Methods of analyzing flow through water-distribution networks are discussed on page 90. No two distribution systems are exactly alike, but most of them

[1] See *Proc. AWWA*, 1915 and 1920.

[2] See *Jour. NEWWA*, 1913, Vol. XXVII.

[3] See (*a*) "Conveyance and Distribution of Water for Water Supply," by Edward Wegman, D. Van Nostrand Company, Inc., New York. This reference contains data on foreign as well as American cities.

(*b*) "Water Supply Engineering," *op. cit.*

can be classified according to their resemblance to loops, gridirons, or trees. In the loop system, large feeder pipes that surround areas several blocks square serve smaller cross pipes connected at each end into the main loop. The grid system is laid out checkerboard fashion and may resemble the loop system, but its pipes often diminish in size with increasing distance from the source of supply. The tree system consists of a single main, usually reducing in size the farther it goes from the source, which feeds branch pipes supplying the consumers. The tree type is the least desirable since it lacks the interconnections that give added reliability in a grid. Grid and loop systems often are backed up later with feeder pipes leading directly from the pumping station to remote distribution centers, thus serving to bolster the supply to meet increased demands with growth of population.

Provided it were known in advance along what lines a town would grow and how the population would be dispersed, it would be possible to make a scientific layout of the distribution system to serve the purpose adequately, at least cost. As between different layouts for serving the same town, it may be possible to work out a judicious arrangement of feeders whereby one layout will accomplish the same result as another but with a saving of 25 per cent or more in the weight of pipe used.[1] Further references on the economics of pipe size and water velocity are listed on page 1024 under Hydraulics.

Satisfactory pressures for distribution systems range from 30 to 100 psi. Pressures below 30 psi cannot supply water successfully to three- and four-story buildings without booster pumps. Pressures over 100 psi require heavier distribution pipes, produce more leakage, and are unsuited for use in plumbing fixtures. Pressures in the range of 50 to 75 psi are most generally satisfactory. In cities where there are differences in surface elevation in excess of 100 ft, it often is advisable to zone the system according to elevation in order to avoid excessive pressure at the lower elevations. The different zones usually have independent reservoirs or supplies from the pumping station and are designated as "high service" and "low service." As pointed out under Hydraulics on page 1024, the selection of pipe sizes in distribution networks is influenced more by the necessities of maintaining adequate water pressures than by the economics of pumping costs.

[1] See article by C. D. Ward in *Eng. News*, Aug. 12, 1909, p. 172 and Nov. 4, 1909, p. 297.

Demand for Fighting Fires.[1]—Hose streams should have a capacity of 250 to 300 gpm. In the residential section of small cities and suburban areas of large cities where no congestion exists and where buildings are not too high, two fire streams generally furnish adequate protection. Where the buildings are 50 ft or less apart so that fire in one may spread to others, 1,000 gpm should be provided. Where the district is closely built or buildings approach the dimensions of hotels or high-grade residences, 1,500 gpm are required. Up to 6,000 gpm may be required in densely built sections.

In most cities the maximum fire flow will be required in the principal mercantile district, in the main industrial district, or around lumber yards and other locations where large amounts of combustibles are stored. In the case of great fires in large cities, two hose lines may be Siamesed together so that, with the aid of a pumper, they may be able to deliver a stream of 500 to 1,000 gpm. The pumps and water-distribution system should be capable of supplying the amounts of water needed for fire fighting at the respective points indicated, above and beyond the maximum demand for other purposes. Values of the cofficient C recommended by the National Fire Protection Association for use in the Williams-Hazen formula will be found in Table XLIII-B on page 278.

Although the demand for extinguishing fires varies between districts, as indicated, the requirements for each city as a whole can be predicated on population which is the usual basis for estimating other water requirements. In estimating the quantity of water required for extinguishing fires, an allowance should be made for probable losses from broken connections and hydrants left open. Including these losses the National Board of Fire Underwriters recommends the fire flows shown in Table XI which were computed by the formula $G = 1{,}020\sqrt{P}\,(1 - 0.01\sqrt{P})$ where G is fire flow in gallons per minute for an average city and P is the population in thousands.

Experience with the largest fires that the country has experienced indicate that 20,000 gpm usually should suffice even in big cities.[2] The National Board of Fire Underwriters recommends

[1] Abstracted principally from the rules and recommendations of the NFPA, see "Crosby-Fiske-Forster Handbook of Fire Protection," published by the National Fire Protection Association, 60 Batterymarch St., Boston, Mass.

[2] See article by John R. Freeman, *Jour. NEWWA*, Vol. VII, p. 60, and publications of the National Board of Fire Underwriters.

providing for a fire duration of 5 hr for towns of less than 2,500 population and 10 hr for larger cities. Unless there is an inexhaustible supply of water available to the pumps, sufficient reservoir capacity beyond normal requirements must be provided for the expected fire duration.

TABLE XI.—FIRE FLOW REQUIRED BY NATIONAL BOARD OF FIRE UNDERWRITERS FOR PRINCIPAL MERCANTILE OR INDUSTRIAL DISTRICTS OF AVERAGE CITIES

Population	Gpm	Population	Gpm	Population	Gpm
1,000	1,000	22,000	4,500	125,000	10,000
2,000	1,500	28,000	5,000	150,000	11,000
4,000	2,000	40,000	5,000	200,000	12,000
6,000	2,500	60,000	7,000	Over 200,000	See note
10,000	3,000	80,000	8,000		
13,000	4,000	100,000	9,000		

NOTE.—Over 200,000 population, 12,000 gpm, with 2,000 to 8,000 gpm additional for a second fire.

The values in Table XI, as well as the estimated capacities required for residential districts, apply to average conditions only. There are likely to be special cases requiring greater capacities. Drafts at a number of severe fires have exceeded the maximum limits given in Table XI. Examples are

Fall River conflagration, Feb. 2, 1928.............. 21,600 gpm
Baltimore Post Building fire, Jan. 1, 1931........... 18,000 gpm
Norfolk conflagration, June 7, 1931................ 14,700 gpm
Pulpwood fire, Port Alped, Que., Apr. 29, 1932...... 33,000 gpm
Chicago Stockyards fire, May 19, 1934............. 50,000 gpm

At the Fall River conflagration 32 pumpers were present, 20 at Norfolk, and the 106 at Chicago probably set an all-time record. The rate shown in Table XI and the expected duration were exceeded also in the 1914 conflagration at Salem, Mass., when a city of 50,000 population needed about 17,000 gpm for 14 hr. At the other extreme in a small suburban community of scattered homes, it might be unreasonable to have to plan on fighting any fire for more than an hour at a time.

A distinction should be made between *static* and *flowing pressures* and, unless static pressure is specifically mentioned in fire protection work, the *flow* pressure at the hydrant is meant. Pressures should be such that a portion of the fire flow may be used as

direct hydrant hose streams. Such conditions require: 50 psi in thinly built residential districts and village mercantile districts where buildings do not exceed two stories; 60 psi in closely built residential districts and elsewhere where not more than 10 buildings exceed three stories; 75 psi in high-value districts. Hose-stream data for various hydrant pressures, hose lengths, and nozzle sizes will be found in Table IV on page 1102.

Where the water system does not provide adequate pressure for direct hose streams, fire department pumpers should be used. The minimum satisfactory hydrant pressure for supplying pumpers is 20 psi, although it may be possible to go as low as 10 psi where hydrants are well located and suction lines are short.

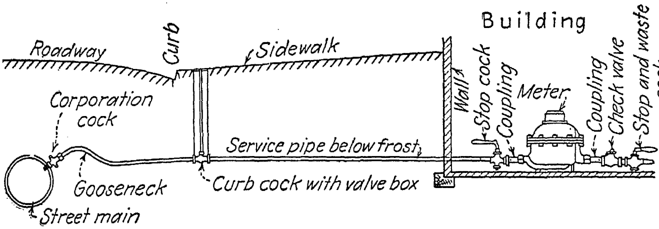


FIG. 13.—Typical water-service connection.

In some cities situated on the seacoast, lakes, or rivers separate *high-pressure* fire systems have been installed for protecting the most important business, manufacturing, and water-front sections. Such systems have their own pumps and mains, and operate on raw water which must be kept out of the regular distribution pipes. Separate systems of this sort usually are operated at pressures of 100 psi or more so as to avoid having to use fire engine pumpers.

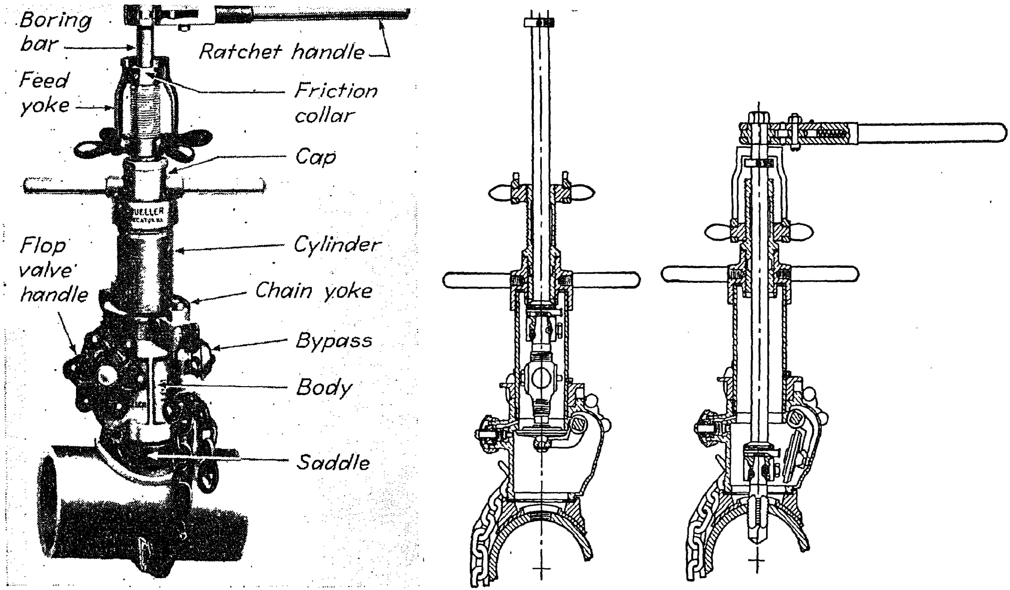
Service Pipes and Connections.—A typical service-pipe installation for connecting a street main to the piping in a consumer's premises is shown in Fig. 13. The service is attached to the main by a tap called a *corporation cock*. Next comes a *gooseneck* of lead or copper which affords enough flexibility to accommodate earth settlement and thermal expansion. In order to avoid having to dig down to the corporation cock to discontinue service to a consumer, a *stopcock* with a *curb box* usually is placed inside the curb line under, or near, the sidewalk. Finally the *meter* is placed, preferably, just inside the consumer's basement wall where

a *stop-and-waste cock* is provided to enable the consumer to shut water off the premises when the plumbing needs repairs, or to drain the piping to prevent freezing if the premises are to be left unoccupied in cold weather. Meters sometimes are installed in buried meter boxes placed somewhere between the curb and the building, but this arrangement is not so satisfactory from an accessibility standpoint and involves the possibility of freezing in cold climates. Underground services frequently are installed with the aid of a pipe-pushing machine as described on pages 1184 to 1186. Materials for water services are discussed on page 1039.

Corporation cocks usually are installed through a water lock in a special tapping machine (see Fig. 14) while the main is under pressure. Since it is generally considered unsafe to insert a threaded tap larger than 2 in. in any size of *cast-iron* main, services of greater capacity should be made with multiple taps joined together, or a service clamp or split tee should be used instead and a hole drilled in the main as described in the second paragraph. Corporation cocks can be installed in *steel* mains of 12-gauge steel and heavier with a tapping machine using special corporation cocks designed expressly for steel pipe. These cocks have a tapered thread and a rubber-covered sealing ring and compression nut which can be turned down to give a watertight connection that will develop the full strength of the pipe wall or fitting without leakage. Corporation cocks with standard tapered pipe thread can be used with steel pipe having a wall thickness of ¼ in. or more.

Another method of making connections to *steel* pipe is with *service clamps* which can be placed over the wrap where wrapped pipe is used. After the clamp is installed and the corporation cock is in place, a tapping machine can be attached to the cock and used to drill a hole in the main by operating the drill through the opened cock. *Outlets* also may be *welded* to steel pipe in the shop or in the field at the time of installation, using a welding saddle, or a short nipple threaded on one end, or a half pipe coupling. Where a nipple is welded to the main, a standard plug cock is used as a shutoff instead of a corporation cock. After the welded connection and cock are attached, the main can be drilled under pressure with a tapping machine.

Larger services and *branch lines* can be connected under pressure to either cast-iron or steel mains through the use of bolted-on saddles, service clamps, or split tees. After one of these devices



a *b* *c*

FIG. 14.—Tapping machine for installing corporation cocks in water mains under pressure. Left, external view; right, drilling and tapping operation; center, corporation cock entering through water lock. (*Courtesy of the Mueller Company, Decatur, Ill.*)

is bolted around the main and calked, a valve is screwed in, calked in, or bolted to, the branch outlet. The valve is then opened, the drilling machine bolted on, the hole bored into the main, the valve closed, and the machine removed. There is no interruption of service and no weakening of the main. The drill operates through the valve and the thrust is taken either through the bolted connection or, in case a spigot-end valve is used, through a built-in clamping device. Branch connections can be installed in this manner in sizes up to 12 in. or larger, provided the main is one pipe size larger than the branch.

TABLE XII.—EQUIVALENT RESISTANCES OF FITTINGS AND VALVES USED IN WATER-SERVICE PIPES[1]

Corporation Cock	Equivalent Length, Ft
½ in. with ¾-in. copper adapter	5 to 7 (of ½-in. pipe)
⅝ in. with ¾-in. copper adapter	4 to 5 (of ⅝-in. pipe)
¾ in. with ¾-in. copper adapter	6 to 10
1 in. with 1-in. copper adapter	8 to 12
Curb stop	
¾ in. for copper service	3 to 4
1 in. for copper service	2 to 3
Meter yokes (used principally in outdoor meter pits)	
⅝-in. ramshorn	10 to 15
⅝-in. straight line	2 to 3
¾-in. ramshorn	8 to 12
Stop-and-waste valves	
½ by ¾-in. compression valve	16 to 19 (of ½-in. pipe)
¾ by ¾-in. compression valve	18 to 23
1 by 1-in. compression valve	30 to 50

[1] Computed largely from Table 1 of "Pressure Losses in Customer's Services," by H. W. Niemeyer and J. A. Bruhn, *Jour. AWWA*, May, 1932, p. 631. The range of values is intended to indicate how generously different makes of valves and fittings are built. For equivalent lengths of usual pipe fittings and valves see Table XIV on p. 100.

Choice of *materials* for water-service pipes is discussed under Pipe and Joints on pages 1039 to 1041, and further ideas on installing services and the use of *pipe-pushing machines* will be found on pages 1184 to 1186 in Chap. XV on Gas Piping. The tendency is to use larger diameter service pipe than formerly since it is not unusual to find that the loss of head through the service connection, meter, and piping in the consumer's premises is greater than the

total loss of head between the pumping station and the corporation cock. The *equivalent resistances* of fittings and valves used in water service pipes are given in Table XII.

IRRIGATION PIPING

In semiarid regions where the natural rainfall is insufficient for growing crops to best advantage, irrigation frequently is resorted to if a supply of water is available from a river or lake. This can be accomplished by gravity flow from ditches or furrows, or by pressure sprinkling from pipes. An excellent discussion of irrigation from a piping man's viewpoint will be found in "Handbook of Water Control," published by the California Corrugated Culvert Co. of Berkeley and Los Angeles, Calif.

Large amounts of lightweight steel pipe (see pages 399 to 416) are used for irrigation purposes in the western United States and elsewhere. Since the advent of fusion welding, most of this pipe is fabricated from strip or sheet steel using straight- or spiral-welded seams (see pages 352, 353). Where used aboveground, such pipe usually is joined with drive joints or with compression couplings of one sort or another such as Dresser or Victaulic, or those having lever-clamping devices instead of bolts to facilitate breaking joints for moving the pipe about (see pages 1040, 1041). Twenty-foot lengths are usual. Portable lightweight pipe for irrigation purposes frequently is galvanized as a protection against rusting.

Overhead Sprinkling Systems.—Sprinkling, or overhead irrigation, may be carried on by the use of permanent systems with underground supply pipes and permanently located sprinklers, or by the use of portable pipe. Overhead irrigation by sprinklers became practicable on a large and economical scale with the development of portable systems using pipe light enough to be moved around readily, yet strong enough to withstand rough field service and equipped with couplings that work easily and quickly without the use of tools, or at most with just a wrench. Portable systems are said to have many advantages over permanently installed overhead systems. The initial investment is considerably less, there is nothing in the field to interfere with cultivating and planting, and the outfits can be moved quickly from one field to another as the need arises. The following recommendations for installing and operating aboveground portable sprinkling systems are taken from the "Handbook of Water Control:"

Experience in the field has shown that generally, with two men, a 1000-ft line can be moved to its next position parallel to the former position and 60 ft from it in from 30 to 35 min, ready for another application.

Often in the field, special fittings such as ells, wyes, tees, etc., may be used to advantage to meet special operating conditions.

When one single line is used it is necessary to stop irrigating long enough to move the line to a new set-up. If an alternate line is provided, no time is lost in moving.

Revolving sprinklers give best results when operating at minimum pressures of from 20 to 30 psi, depending upon their size, the smaller sprinklers requiring less pressure than the large ones. The diameter of coverage of sprinklers varies with the size of the sprinkler and the operating pressure employed.

It has been found, in order to obtain and insure satisfactory coverage, that when small sprinklers discharging less than 7 gpm are used they should be spaced every 20 ft along the line. Sprinklers discharging 7 gpm or more give adequate coverage and distribution when spaced every 40 ft (at every second pipe joint) along the line.

Lines carrying small sprinklers operating under minimum pressures of 20 psi should be moved a maximum distance of 40 ft between set-ups.

Lines carrying sprinklers operating under a minimum pressure of 25 psi should not be moved more than 50 ft between set-ups.

Lines carrying sprinklers operating under a minimum pressure of 30 psi should not be moved more than 60 ft between set-ups.

Manufacturers of sprinklers designed for agricultural service adjust their sprinklers to turn slowly—about 1 rpm. If they turn too fast, the effective diameter of the coverage circle is materially reduced.

The nozzle sizes of the sprinklers can be varied, depending on the amount of water and the pressure available. The rate of application of water depends on the character of the soil. Some types of soil will take water faster than others, and on such soils water can be applied at a higher rate.

Units of Water Measurement.—In irrigation and hydroelectric work as well as in hydraulic mining, the units for *rate* of flow are the cubic foot per second (or second foot) for larger quantities, and the miner's inch for smaller quantities. The cubic foot per minute also is used to some extent in hydraulic mining. The miner's inch was developed in the early mining days and is still used extensively for other purposes as well. It is an awkward term, but its use often is necessary on account of custom and law. A miner's inch is the rate at which water discharges through 1 sq in. of opening under a prescribed head (approximately 6 in.), and the number of miner's inches is equal to the area of the opening in square inches. The value of the miner's inch varies in different localities, ranging from 1/50 to 1/38.4 cu ft per sec (see Table XIII).

The units for *volume* of water delivered are the cubic foot, the acre-foot, and the acre-inch which is one-twelfth the acre-foot. For large volumes the acre-foot is the unit recommended. This

is the volume required to cover 1 acre to a depth of 1 ft, which equals 43,560 cu ft. One cubic foot per second flowing steadily for 24 hr approximately equals 2 acre-feet.

The interrelation of the various units of measurement for rate of flow and volume of water delivered is shown in Table XIII.

TABLE XIII.—CONVERSION OF UNITS OF FLOW USED IN MEASURING WATER

(From "Handbook of Water Control")

Cubic feet per second	Gallons per minute	Million gallons per day	Miner's inches: Arizona, California, Montana, Nevada, Oregon	Miner's inches: Idaho, Kansas, Nebraska, New Mexico, North Dakota, South Dakota, Utah	Miner's inches: Colorado	Acre-inches per hour	Acre-feet per 24 hours
1	448.8	0.646	40	50	38.4	0.992	1.983
0.00223	1	0.001440	0.0891	0.1114	0.0856	0.0022	0.00442
1.547	694.4	1	61.89	77.36	59.44	1.535	3.07
0.025	11.25	0.0162	1	1.25	0.960	0.0248	0.0496
0.020	9.00	0.01296	0.80	1	0.768	0.0198	0.0397
0.026	11.69	0.0168	1.042	1.302	1	0.0258	0.0516
1.01	452.42	0.651	40.32	50.40	38.71	1	2.00
0.504	226.3	0.3258	20.17	25.21	19.36	0.5	1

Measuring Devices.—The miner's inch is measured through a *miner's inch box* which is a special form of free-flowing orifice consisting of an opening in a plank under a head of from 3 to 9 in. The arrangement of the opening, which may be from 1 to 4 in. high, the thickness of the plank, which may be from 1 to 3 in., and the point from which the effective head is measured, as well as the head itself, are largely matters of local custom and of state law. Owing to the uncertainties of this method of measurement, the miner's inch usually is construed now as some fraction of 1 cu ft per sec as shown in Table XIII.

A number of commercial *meters* are on the market which are designed for measuring irrigation water. Usually they regulate the flow of water and may, or may not, be arranged to register the total quantity passed. The measuring element can be a weir, flume, orifice, Venturi meter, or current meter used in connection with a registering device of some sort. Or the registering device may be omitted and the metering element used merely to limit

the rate at which water can be taken. A typical installation of a *metering orifice box* is shown in Fig. 15.

A special form of submerged orifice called a *meter gate* is used extensively for measuring water for irrigation purposes. A meter gate consists of a sluice gate, similar to that shown in Fig. 4, to which are fitted two measuring wells in which the elevation of water on the upstream and downstream sides of the device can be read in inches with a rule. The difference between the two readings gives the available head across the meter gate from which the rate of flow can be computed or taken from a table to corre-

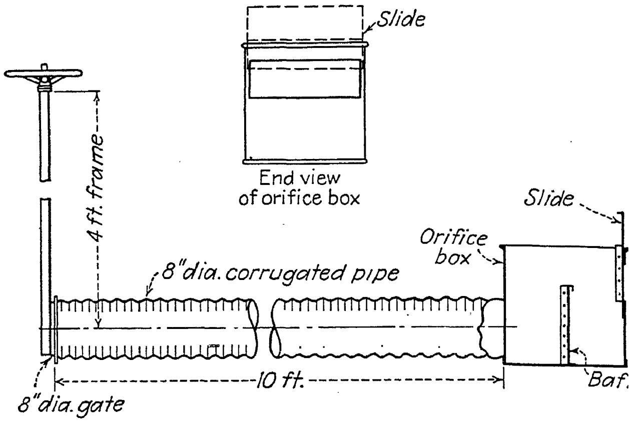


FIG. 15.—A typical installation of a metering orifice box used to measure water for irrigation purposes.

spond with the amount of opening. A typical meter gate installation is shown in Fig. 16.

When the installation of a meter gate is contemplated and the site selected, sufficient excavation is made to place the outlet pipe level with a clear opening inlet and outlet and set low enough to ensure the proper submergence of the outlet. Submergence should not be less than 6 in. under lowest conditions. If full submergence is not maintained, no readings can be taken in the downstream measuring well because of the surging of the water surface. Also, excavation should be made to give full contraction at the entrance if possible. The clearance between the inside of the gate and the side walls and bottom of the canal on the upstream side of the meter gate should be not less than 6 in.

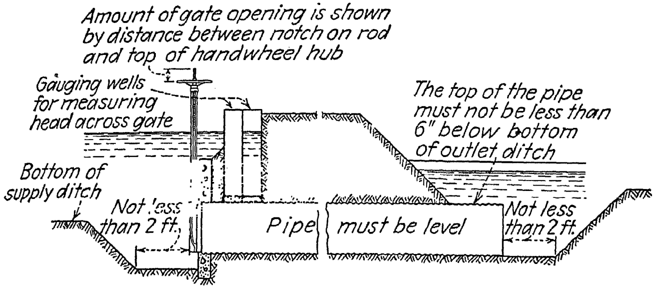


FIG. 16.—A typical meter-gate installation used to measure water for irrigation purposes.

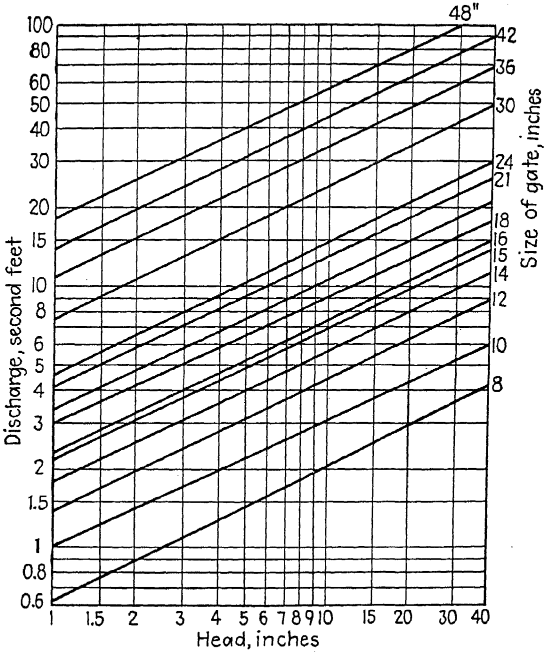


FIG. 17.—Discharge capacity at different heads of fully opened meter gates of diameters shown. (*From "Handbook of Water Control,"* 1943.)

After the gate has been installed, the measuring wells are placed in position and connected by $\frac{3}{4}$-in. pipes, one to the upstream side of the gate and the other to the main pipe 12 in. below the gate. A notch is made with a hack saw in the stem of the gate at a point flush with the top of the handwheel when the handwheel is without slack on the stem and the bottom of the gate slide is exactly level with the bottom of the inside of the gate seat. This is called the point of "zero gate opening." This means that the gate is not tightly closed. The stem will be the width of the gate seat higher than it is in the fully closed position. The amount of any gate opening is obtained by measuring the distance from the hack saw mark to the top of the handwheel.

When the turnout is in use, the water stands in one stilling well at the level of the water in the upstream canal and in the other well at the static level of the water in the turnout pipe at a point 12 in. downstream from the face of the gate seat. The difference in these two levels represents the static pressure on the gate opening and is denoted in these considerations as the "head on the gate."

Meter gates can be fastened to corrugated or smooth pipe of any length, as the only friction involved is in the first foot of pipe which is furnished as part of the meter gate. The discharge capacity at different heads of fully opened meter gates as manufactured by the California Corrugated Culvert Co. are given in Fig. 17. Charts are available in their "Handbook of Water Control" for reduced flows with lesser gate openings.

CHAPTER XIII

FIRE-PROTECTION PIPING

Piping and apparatus for fire protection must satisfy two essential requirements: they must be capable of producing, without question, the desired performance, and they must be designed so as to function invariably regardless of age or weather conditions. The National Board of Fire Underwriters and allied organizations have established elaborate rules for the installation of all forms of protective apparatus and, for an extensive installation or when insurance rates might be affected, these should be consulted and followed exactly. It is possible here to treat only the main features of fire-protection piping.

Automatic Sprinkler Systems.[1]—The piping of an automatic sprinkler system must be designed so as to ensure (*a*) an adequate and reliable water supply, (*b*) ample and complete distribution, and (*c*) proper protection from freezing.

Automatic sprinkler systems must be served by at least one automatic water supply of adequate pressure, capacity, and reliability. The necessity for a second supply depends on various factors such as the quality and ready availability of public fire department response, construction features of the building affecting the spread of fire, occupancy, etc. A connection from a reliable waterworks system having adequate capacity and pressure is preferable as a single or primary supply. An elevated tank makes a good primary supply and may be acceptable as a single supply. A pressure tank also is satisfactory as a primary supply and in some cases may be acceptable as a single supply. A desirable auxiliary supply for any of the primary supplies is provided by a connection through which the public fire department can pump

[1] See "Standards of the National Board of Fire Underwriters for the Installation of Sprinkler Equipment," NBFU *Pamphlet* 13. NBFU pamphlets may be obtained from the National Fire Protection Association, 60 Batterymarch St., Boston, Mass. See also Crosby-Fiske-Forster" Handbook of Fire Protection," published by the National Fire Protection Association, Boston, Mass., 1941, 9th ed.

water into the sprinkler system. Pipe size shall not be less than 4 in. for fire-engine connections except that 3-in. pipe may be used to connect a single hose connection to a 3-in. or smaller riser. A suitable fire-pump installation taking water from a water main of adequate volume, or taking suction under a head from a reliable storage tank of adequate capacity, makes a good secondary supply and in some cases may be accepted as a single supply.

The spacing and location of sprinkler heads are governed by NBFU rules which take into account the type of building construction and the nature of the occupancy hazard. In general, one sprinkler head is required for each 100 sq ft of floor area, the light-hazard occupancies requiring fewer sprinklers, and the extra-hazard occupancies requiring more. Light-hazard occupancies include buildings such as apartment houses, dwellings, hospitals, hotels, or office buildings; ordinary-hazard occupancies include manufacturing buildings, warehouses, etc.; and extra-hazard occupancies include buildings or portions thereof where the hazard is severe. Table I shows the number of sprinkler heads on one floor of one fire section which may be supplied by the various sizes of pipes for the three types of occupancy.

TABLE I.—CAPACITIES OF SPRINKLER PIPES IN TERMS OF NUMBER OF SPRINKLERS ALLOWED
(From NBFU Bull. 13)

Nominal pipe size, inches	Maximum number of sprinklers allowed on supply pipe of size indicated		
	Light-hazard occupancy	Ordinary-hazard occupancy	Extra-hazard occupancy[1]
1	2	2	1
1¼	3	3	2
1½	5	5	5
2	10	10	8
2½	40	20	15
3	No limit	40	27
3½	..	65	40
4	..	100	55
5	..	160	90
6	..	250	150

[1] Applies also to thermostatically operated open-head systems (see p. 1093).

Automatic sprinkler heads with nominal ½-in. discharge orifices usually will be required. Discharge capacities of approved sprinkler heads having a ½-in. orifice or its equivalent in discharge,

at various pressures up to 100 psi are shown in Table II. The flow in gallons per minute is approximately equal to one-half the pressure in pounds per square inch plus 15. Thus with 20 psi at the sprinkler head, the flow would be $(\frac{1}{2} \times 20) + 15 = 25$ gpm. A pressure of 20 psi at the sprinkler head is generally considered satisfactory for sprinkler operation.

TABLE II.—DISCHARGE IN GALLONS PER MINUTE FROM ½-IN. SPRINKLER HEADS
(From NBFU *Pamphlet* 13)

Pressure at sprinkler, psi	Discharge, gpm	Pressure at sprinkler, psi	Discharge, gpm
10	18	35	34
15	22	50	41
20	25	75	50
25	28	100	58

Piping should be arranged so that the number of heads on any branch line on either side of a cross main should not exceed eight. Larger pipes than shown in Table I are required in case of unusually long or crooked runs. Each riser should be of sufficient size to supply all the sprinklers connected to it on any one floor, or if there are no approved fire stops between floors, then the riser should be sufficient to accommodate the total number of sprinklers connected. Where exposed to frost, supply pipes or risers should be well protected by means of insulating materials. Provision should be made to drain all parts of the system properly.

Sprinkler piping should not be used in any way for domestic water service. Where a secondary nonpotable water supply is connected with a primary supply drawing from the public water system, particular attention should be directed to avoid a hazardous cross connection between the two which could result in pollution of the public supply (for a discussion of cross connections, see pages 977 to 980).

Pipe used in sprinkler systems should be suitable for the particular service conditions involved. Where corrosive conditions exist, for instance, consideration should be given to the use of types of pipe, fittings, and hangers intended to resist corrosion, including use of protecting coatings, depending on the severity and nature of the corrosion (for further data on corrosion see Chap. XVII).

All valves on connections to water supplies and in supply pipes to sprinklers should be of the outside-screw-and-yoke (O. S. & Y.) pattern. Each system should be provided with a gate valve so located as to control all sources of water supply except that from fire-department sources. There should be no shutoff valve in the fire-department connection. Valves on each floor are desirable in large buildings or where the contents are unusually susceptible to water damage. Where there is more than one source of water supply, a check valve should be installed in each connection; where cushion tanks are used with automatic fire pumps, no check valve is required in the tank connection. With but one supply connection, no check valve is required except where there is likelihood of water circulation, or if there is a fire department connection on the system.

Fittings should be of cast iron designed to withstand a working pressure of 175 psi. American Standard Class 125 cast-iron flanged fittings in sizes 12 in. and below in accordance with ASA B16b are acceptable for this service. Screwed cast-iron fittings in accordance with the 125-lb class of the American Standard 125-lb Cast-iron Screwed Fittings, ASA B16b, are acceptable. Where the normal pressure exceeds 175 psi, 250-lb fittings which are good for 400-psi water pressure should be used. Malleable-iron fittings, unless of a type specially approved for sprinkler work, are not to be used. Sprinkler fittings formerly made to the American Standard for 175- and 250-lb Long-turn Sprinkler Fittings, ASA B16g-1929 and 1937, are no longer mentioned in NBFU rules and their manufacture has been discontinued. Screwed, flanged, or other approved types of pipe joints should be used. Welding of joints in risers or large feed lines may be allowed on approval of the inspection department having jurisdiction. Hangers should be of approved types.

All new systems should be given a hydrostatic test at not less than 200 psi for 2 hr, or at 50 psi in excess of the normal pressure where the latter exceeds 150 psi. Test pipes of 2-in. size must be provided at suitable locations to assure that water supplies are in order. A 1-in.-diameter test pipe terminating in an outlet giving a flow equivalent to one sprinkler should be provided on wet systems. The discharge should be at a point where it can be readily observed.

A dry-pipe system is required in rooms which cannot be properly heated. In this system the piping is ordinarily filled with air at a

pressure considerably less than the water pressure. When a sprinkler head opens, water enters the system and drives the air out ahead of it.

The most important feature of a dry-pipe system is the dry-pipe valve, a device that normally prevents water from entering the system but opens when the air pressure is lowered as the result of the opening of a head.

Nonfreezing solutions may be used in automatic sprinkler systems in minor unheated areas subject to freezing, but they are not encouraged for use as a substitute for a dry-pipe valve.

Thermostatically operated sprinkler systems are normally without water in the system piping, the water supply being controlled by an automatic valve operated by thermostatic devices that are independent of the sprinkler heads. Sprinkler heads accordingly may be opened or closed. In general, not more than 1,000 closed-head sprinklers nor more than 75 open-head sprinklers should be controlled by any one valve. On closed-head systems, pipe schedules for sprinklers are the same as for wet systems. Pipe schedules for open-head systems are the same as for extra-hazard occupancy given in Table I. The number of open-head sprinkler heads allowed on thermostatically operated valves are as follows:

Valve Size, Inches	Sprinklers
1½	5
2	10
3	36
4	75
6	150

Outside sprinklers also may be installed for protection against exposure fires. Only sprinklers of such types as are approved for window, cornice, sidewall, or ridgepole service should be installed for such use. Separate rules for location and number of sprinklers, type of material and sizing of pipe, valves, and fittings for outside sprinkler service are contained in NBFU *Pamphlet* 13.

Tanks.[1]—Overhead gravity tanks of adequate capacity and elevation may be used for a primary water supply and may be acceptable as a single supply for automatic sprinkler systems and hose connections. They usually are built either of wood staves or

[1] See "Standards of the National Board of Fire Underwriters for the Construction and Installation of Gravity and Pressure Tanks," NBFU *Pamphlet* 22.

of steel. The size of tank for a given service is determined individually by the insurance authorities. In general, when feeding sprinklers only, the tank should have a capacity of at least 5000 gal located not less than 35 ft above the underside of the roof. When feeding both sprinklers and hose, a minimum capacity of 30,000 gal is required.

Elevated gravity tanks must have a discharge pipe of not less than 6-in. size up to 25,000 gal capacity; generally not less than 8 in. for 30,000 to 100,000 gal; and 10 in. for greater capacities. The pipe must be of flanged cast-iron or steel material, welded steel, or of approved corrosion-resistant materials with flanged or welded connections. If flanges are used, copper, lead, or good-quality rubber gaskets may be used. A slip expansion joint is required at the tank bottom if the pipe is over 30 ft long. The proper arrangement of a gravity tank located over a building is shown in Fig. 1.

An important provision for an outside tank is the means of protecting it against freezing. The usual arrangement consists of a tubular steam heater to which a connection is made from the base of the tank discharge pipe. The heated water is carried up to the tank by a separate pipe. This arrangement permits the temperature of the coldest water to be observed readily and is the simplest and most reliable method. The coldest water should not be allowed to go below 40 or above 50 F. The piping arrangement is shown in Fig. 1. Self-contained coal-fired heaters are permitted when unavoidable.

Pressure Tanks.[1]—In general, pressure tanks are acceptable only for supplying automatic sprinkler systems and for this service may be considered a good primary supply; in some cases, they may be acceptable as a single supply. Pressure tanks are ordinarily kept two-thirds full of water, and an air pressure of at least 75 psi should be maintained. As the last of the water leaves the tank, the pressure must be such that at least 15 psi pressure is available at the highest sprinkler head. The capacity of the tank is set by the insurance inspection authorities having jurisdiction and is usually between 3,000 and 5,000 gal per tank. The construction of the tank must be in accordance with the Rules for the Construction of Unfired Pressure Vessels of the ASME. An additional $\frac{1}{16}$ in. should be added to the computed plate thickness to allow for

[1] See "Standards of the National Board of Fire Underwriters for the Construction and Installation of Gravity and Pressure Tanks," NBFU *Pamphlet* 22.

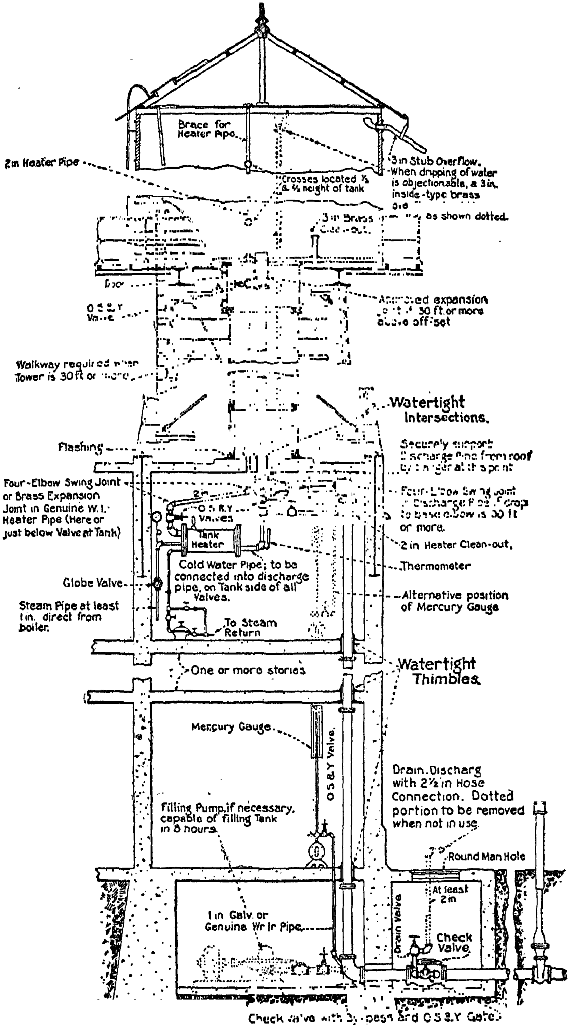


FIG. 1.—Pipe connections for a gravity storage tank located over a building. (*Courtesy of National Board of Fire Underwriters.*)

corrosion. Piping connections should be made as shown in Fig. 2. Other appliances include a relief valve of ¾-in. size or larger in the air-supply pipe; a relief valve of 1½-in. size or larger at the filling pump; and an air compressor with a capacity of from 16 to 20 cu ft

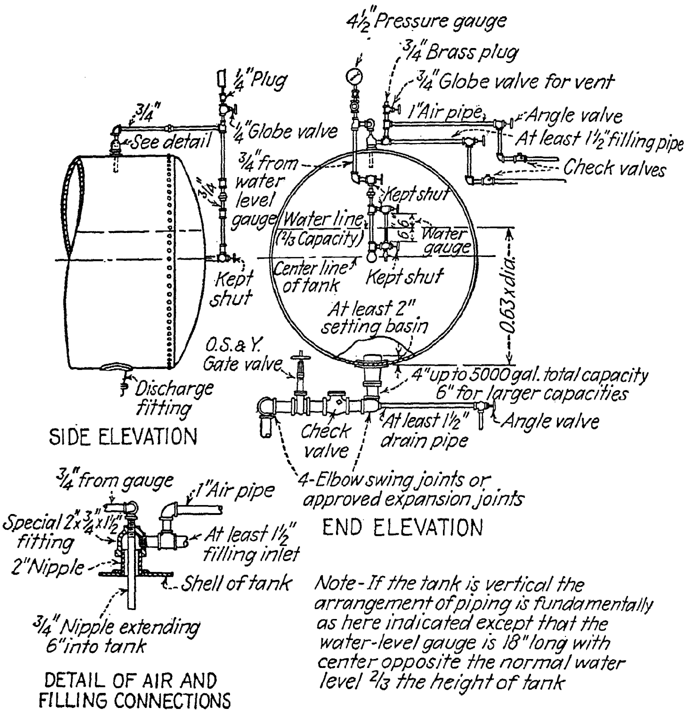


FIG. 2.—Pipe connections for pressure tanks. (*Courtesy of National Board of Fire Underwriters.*)

of free air per minute. The water-filling supply shall be capable of delivering at least 65 gal per min.

Standpipes and Hose.[1]—A standpipe and hose system for a building consists of one or more vertical pipes or risers of adequate size to which fire hose is attached at numerous stations ready for

[1] See "Standards of the National Board of Fire Underwriters for the Installation of Standpipe and Hose Systems," NBFU *Pamphlet* 14.

instant use. The system is connected to a source of water supply which is under pressure at all times, or is arranged to admit water to the system automatically or through the use of manually operated remote-control devices located at each hose station. The standpipe system furnishes the only reliable means of obtaining effective fire streams at the upper stories of high buildings and serves to complement the automatic sprinkler equipment.

Standpipe systems may be designed for two types of service: (1) for small hose streams for use by the building occupants and (2) for large hose streams for use by local or public fire departments. For small hose less than 2½ in. in size, the standpipes should be not less than 2 in. for buildings not exceeding four stories or 50 ft in height, or 2½ in. minimum for higher buildings. For the standard 2½-in. fire hose, standpipes should be at least 4 in. for buildings not exceeding 6 stories or 75 ft in height, or 6 in. minimum for higher buildings. If provision is to be made for attaching fire-department hose, standpipes should be at least 6 in. in size. The number and arrangement of standpipe equipments necessary for proper protection are governed by occupancy, character, and construction of the building, exterior exposures, and accessibility. In general, valved outlets should be located and spaced so that all portions of each story will be within reach of a stream from a nozzle on 100-ft length of hose. The best location for the standpipes is near fire-resistive stairways, which are provided in many buildings.

The quantity of water required to supply a standpipe system depends on the size and number of streams that may be required, and the length of time such streams are likely to be in operation. The probable number of streams required will have to be estimated before deciding on the water supply. The value of water supplies for standpipe systems should be assessed as follows: (1) city waterworks systems where an adequate pressure is available, (2) automatic fire pumps (500 gpm minimum size), (3) manually controlled fire pumps with pressure tanks (minimum sizes, pumps 500 gpm, tank 4,500 gal), (4) pressure tanks (4,500 gal minimum), (5) gravity tanks (5,000 gal minimum except that, for standpipes for small hose, a domestic gravity tank may be accepted with a minimum of 3,000 gal reserved for fire protection). As in other fire-protection systems, two independent sources of supply are preferable.

At least one fire-department connection having as many inlets as required for the service should be made for each standpipe riser where there is a public fire department with pumpers. Connec-

tions to standpipes from gravity and pressure tanks usually are made at the top of the standpipe system, whereas connections from fire pumps and sources outside the building should be made at the base of the standpipes. Where two or more standpipes are installed in the same building, they should be cross connected at the bottom.

Where the flow pressure at a standpipe outlet exceeds about 50 psi, pressure may be regulated preferably by a pressure-reducing device or by the insertion of a disk with a restricting orifice. The approximate size of the orifice required may be calculated as follows:

$$a = 0.0425 \frac{G}{\sqrt{p_1 - p_2}}$$

where a = area of orifice, sq in.
G = water discharged, gpm.
p_1 = pressure in standpipe at hose outlet in question, psi.
p_2 = pressure on nozzle side of orifice, *i.e.*, nozzle pressure plus friction loss in hose line, psi.

American standard hose couplings are shown on pages 486 to 488. Unfortunately, there are a number of different standard threads in use. The general adoption of the national standard would assure the interchangeability of apparatus, which is highly desirable in severe conflagrations. The thread used by the nearest local fire department should ordinarily be chosen however, even if it is non-standard.

Hose Streams.—Fire hose generally is of two types: (1) cotton with smooth rubber lining usually referred to as fire-department hose, which can be used many times over a long period if reasonable care is exercised in drying, and (2) unlined linen hose, intended only for emergency service as it may deteriorate rapidly after being wet. Although it is possible to reuse it if it is thoroughly dried, it is not intended for domestic use such as washing down floors, etc. Cotton rubber-lined hose ordinarily is provided at outdoor hydrants, whereas linen is used for indoor standpipe systems. If a fire is detected in its early stages, 1½-in. hose often is brought into action first because it requires fewer men, can be handled faster, requires less water, and also minimizes water damage.

The generally recognized authority on hose streams, nozzles, etc., is the late John R. Freeman, whose test data and formulas derived therefrom were published in *Trans. ASCE*, 1889, and *Trans. New*

Eng. Water Works Assoc., 1892. The data on fire streams given in Table IV, for $2\frac{1}{2}$-in. smooth rubber-lined hose, are taken from these tables. The smoothness of the interior surface of the hose is of the utmost importance in determining friction loss. For instance, in unlined linen hose the pressure drops per foot were found to be about two and one-half times those given in Table IV for smooth rubber-lined hose, and the losses per foot for inferior rubber-lined hose were about twice those given in the table. A poor grade of rubber lining or a poor fabric, or both, allow the rubber to be forced into the fabric when under water pressure and the interior of the hose may become nearly as rough as if there were no rubber lining at all.

According to Freeman:

> The ring nozzle is inferior to the smooth nozzle and actually delivers less water. For instance, a $\frac{7}{8}$-in. ring nozzle discharges the same quantity of water as a $\frac{3}{4}$-in. smooth, and a 1-in. ring nozzle the same as a $\frac{7}{8}$-in. smooth. Two hundred and fifty gallons per minute is a good standard fire stream with 80 lb pressure at the hydrant. One hundred pounds pressure should not be exceeded, except for very high buildings or lengths of hose exceeding 300 ft.

Another authority states that a *first-class* fire stream for a $2\frac{1}{2}$-in. hose is one from a nozzle of at least $1\frac{1}{8}$-in. tip diameter with at least 40 psi pressure in the hose close to the nozzle, which would give a discharge of 240 gpm, or 50 gpm for $1\frac{1}{2}$-in. hose with $\frac{1}{2}$-in. nozzle tip.

Additional pressure drop will occur from the water main to the hose connection due to 90-deg turns, restriction of the hydrant valve, hose-coupling washer, etc. With as many streams flowing as there are connections on the hydrant, this loss should not exceed 10 psi, and with proper design the loss can be kept between 2 and 5 lb.

The proper nozzle size to use will depend on the length of hose in use. The following rule of thumb often is used: For 300 to 550 ft of hose, use a nozzle one-half the diameter of the hose. For 600 to 850 ft of hose, use the next size smaller than one-half the hose diameter. For 900 to 1,200 ft of hose, use the second size smaller than one-half the hose diameter. The approximate nozzle reaction or pullback may be estimated as follows: $1\frac{1}{2} \times$ diameter of nozzle (in.) squared $\times$ nozzle pressure. For a $1\frac{1}{4}$-in. nozzle with 50 psi pressure, the nozzle reaction would be $1\frac{1}{2} \times 1\frac{1}{4} \times 1\frac{1}{4} \times 50 =$ 117 lb. The reaction resulting when the nozzle is shut off suddenly may be approximated by multiplying by 1.8 instead of $1\frac{1}{2}$.

In fighting electrical fires, it is desirable in the interest of safety to deenergize the equipment if possible. In case the equipment is energized, mimimum safe distances at which it is believed safe to use fresh water from fire hoses in the vicinity of electrical equipment are given in Table III.

TABLE III.[1]—MINIMUM SAFE DISTANCES FOR HOSE STREAMS FROM HIGH-TENSION CONDUCTORS

Voltage	Fresh water	
	1⅛-in. nozzle	1½-in. nozzle
1,100	6 ft	9 ft
2,200	11 ft	16 ft
3,300	15 ft	22 ft
5,500	18 ft	27 ft
6,600	19 ft	29 ft
11,000	20 ft	30 ft
22,000	25 ft	33 ft
33,000	30 ft	40 ft

NOTES.—Stream from foam extinguisher has 2⅛ times conductivity of fresh water. Well water has 15 times conductivity of fresh water.

Stream from soda-acid extinguisher has 27 times conductivity of fresh water.

Stream from antifreeze extinguisher has 36 times conductivity of fresh water. Sea water has 200 times conductivity of fresh water.

[1] From "Fighting Power Plant Fires—I," by W. E. Rossnagel, *Power*, November, 1941.

Underground Piping.[1]—When it is desired to install a system of underground piping for fire protection for a group of buildings, the layout should be made with the idea of having at least two hose streams available to every part of the interior of each building not otherwise protected, and to provide hose-stream protection to the exterior of all buildings using the length of hose normally attached to the hydrants, and with sufficient hydrants to permit a concentration of flow about any important building with hose lines not to exceed 500 ft in length. The system will also supply such sprinkler systems and standpipes as are installed. It should have a carefully selected source of supply and connections for the city fire department. A loop system is desirable because of its larger capacity.

The source of supply (which must be approved by the insurance authorities) may be (1) the public water supply using not less than a 6-in. connection and preferably two connections of ample capacity, (2) a fire pump of not less than 500 gpm capacity for sprinklers

[1] See also "Regulations of the National Board of Fire Underwriters for Outside Protection."

only and not less than 750 gpm when hydrants also are supplied, or (3) a gravity tank of at least 30,000-gal capacity and 40 ft above the top of the tallest building. No pipe smaller than 6 in. should be used for mains or hydrant branches and this size should also be the minimum for sprinkler connections to provide for possible enlargement of buildings. Minimum satisfactory pressure for fire fighting is considered to be 50 psi at the hydrant, measured while water is flowing.

Friction losses in underground piping may be computed from Tables XLIII-*B* and XLIV on pages 278 to 281.

The depth of cover should be greater for fire lines than for ordinary water-supply pipes because normally there is no flow in them. The proper depth varies from 2½ ft in the Southern states to about 10 ft in northern Canada. The depth should always be somewhat below the lowest known frost line (see pages 1072 to 1074).

Cast-iron pipe conforming to American Standard Specifications for Cast-iron Pit-Cast Pipe for Water or Other Liquids, ASA A21.2 (ASTM Specification A44), Class 150, is acceptable for pressures less than 150 psi. For higher pressures, correspondingly heavier pipe should be used. Cast-iron fittings should be in accordance with American Water Works Association standard specifications. Bell-and-spigot joints should be used unless otherwise approved, and elbows and bends should be clamped or braced to avoid being forced apart by the water pressure. Abstracts of these specifications will be found on pages 432 to 437.

Hydrants should be located about 50 ft from the buildings whenever possible so as to be accessible in case of fire. They should rest on concrete slabs or flat stones and the drain connections should be surrounded with a quantity of small stones to ensure quick drainage. Hydrants should conform to the requirements of AWWA Specifications for Fire Hydrants for Ordinary Water Works Service (see pages 1043 to 1048). Capacities of fire hydrants may be identified by a color scheme for painting hydrants which has been adopted by the Main Water Utilities Association, the National Fire Protection Association, the New England Water Works Association, and the American Water Works Association.[1]

Oil-fire Protection.—Three types of piped fire-extinguishing systems are available for the protection of oil-storage tanks,

[1] See "Uniform Marking of Fire Hydrants; Standard Colors for Painting to Indicate Flow Capacity—AWWA 7F.3.1-1937," published in mimeographed form and in *Jour. AWWA*, April, 1937, p. 449.

Table IV.—Hydrant and Hose Stream Data

Pressure at nozzle, psi	Gallons discharged per minute	Vertical distance of stream, ft.	Horizontal distance of stream, ft.	Pressure in psi required at hydrant or pump to maintain pressure at nozzle through various lengths of 2½-in. smooth, rubber-lined hose. 50 ft.	100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	800 ft.	1,000 ft.
¾-in. Smooth Nozzle												
35	97	55	41	37	38	40	42	44	46	48	53	57
40	104	60	44	42	43	46	48	50	53	55	60	65
45	110	64	47	47	48	51	54	57	59	62	68	73
50	116	67	50	52	54	57	60	63	66	69	75	81
55	122	70	52	58	59	63	66	69	73	76	83	89
60	127	72	54	63	65	68	72	76	79	83	90	97
65	132	74	56	68	70	74	78	82	86	90	98	106
70	137	76	58	73	75	80	84	88	92	97	105	114
75	142	78	60	79	81	85	90	94	99	104	113	122
80	147	79	62	84	86	91	96	101	106	111	120	130
85	151	80	64	89	92	97	102	107	112	117	128	138
90	156	81	65	94	97	102	108	113	119	124	135	146
95	160	82	66	99	102	108	114	120	125	131	143	154
100	164	83	68	105	108	114	120	126	132	138	150	163
⅞-in. Smooth Nozzle												
35	133	56	46	38	40	44	48	52	56	60	68	76
40	142	62	49	43	46	50	55	59	64	68	78	87
45	150	67	52	49	51	57	62	67	72	77	87	97
50	159	71	55	54	57	63	69	74	80	86	97	108
55	166	74	58	60	63	69	75	82	88	94	107	119
60	174	77	61	65	69	75	82	89	96	103	116	130
65	181	79	64	71	74	82	89	96	104	111	126	141
70	188	81	66	76	80	88	96	104	112	120	136	152
75	194	83	68	82	86	94	103	111	120	128	145	162
80	201	85	70	87	91	101	110	119	128	137	155	173
85	207	87	72	92	97	107	116	126	136	145	165	184
90	213	88	74	98	103	113	123	134	144	154	174	195
95	219	89	75	103	109	119	130	141	152	163	184	206
100	224	90	76	109	114	126	137	148	160	171	194	216
1-in. Smooth Nozzle												
35	174	58	51	40	44	51	57	64	71	78	92	105
40	186	64	55	46	50	58	66	73	81	89	105	120
45	198	69	58	52	56	65	74	83	91	100	118	135
50	208	73	61	57	62	72	82	92	102	111	131	151
55	218	76	64	63	69	79	90	101	112	122	144	166
60	228	79	67	69	75	87	98	110	122	134	157	181
65	237	82	70	75	81	94	107	119	132	145	170	196
70	246	85	72	80	87	101	115	128	142	156	183	211
75	255	87	74	86	94	108	123	138	152	167	196	226
80	263	89	76	92	100	115	131	147	162	178	209	241
85	271	91	78	98	106	123	139	156	173	189	222	
90	279	92	80	103	112	130	147	165	183	200	236	
95	287	94	82	109	118	137	156	174	193	211	249	
100	295	96	83	115	125	144	164	183	203	223	.	

From tables published by John R. Freeman, C. E. The pressures given are indicated pressures, not effective pressures. Effective pressures would be slightly greater. For discharge coefficients of various types of fire-hose nozzles, see Fig. 7, p. 56.

TABLE IV.—(*Concluded*)

Pressure at nozzle, psi	Gallons discharged per minute	Vertical distance of stream, ft.	Horizontal distance of stream, ft.	Pressure in psi required at hydrant or pump to maintain pressure at nozzle through various lengths of 2½-in. smooth, rubber-lined hose.								
				50 ft.	100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	800 ft.	1,000 ft.
1⅛-in. Smooth Nozzle												
35	222	59	54	43	49	60	71	82	94	105	127	149
40	238	65	59	50	56	69	81	94	07	120	145	171
45	252	70	63	56	63	77	92	106	120	135	163	192
50	266	75	66	62	70	86	102	118	134	150	181	213
55	279	80	69	68	77	95	112	130	147	165	200	235
60	291	83	72	74	84	103	122	141	160	180	218	256
65	303	86	75	81	91	112	132	153	174	195	236	
70	314	88	77	87	98	120	143	165	187	209	254	
75	325	90	79	93	105	129	153	177	201	224		
80	336	92	81	99	112	138	163	188	214	239		
85	346	94	83	106	119	146	173	200	227	254		
90	356	96	85	112	126	155	183	212	241			
95	366	98	87	118	133	163	194	224	254			
100	376	99	89	124	140	172	204	2[illegible]6				
1¼-in. Smooth Nozzle												
35	277	60	59	48	57	74	91	109	126	142	178	212
40	296	67	63	55	65	84	104	124	144	164	203	243
45	314	72	67	62	73	95	117	140	162	184	229	
50	331	77	70	68	81	106	130	155	180	204	254	
55	347	81	73	75	89	116	143	170	198	225		
60	363	85	76	82	97	127	156	186	216	245		
65	377	88	79	89	105	137	169	201	234			
70	392	91	81	96	113	148	182	217	252			
75	405	93	83	103	121	158	195	232				
80	419	95	85	110	129	169	208	148				
85	432	97	88	116	137	179	221					
90	444	99	90	123	145	190	234					
95	456	100	92	130	154	201	247					
100	468	101	93	137	162	211	261					
1⅜-in. Smooth Nozzle												
35	340	62	62	54	67	94	120	146	172	198	250	
40	363	69	66	62	77	107	137	166	196	226		
45	385	74	70	70	87	120	154	187	221	254		
50	406	79	73	78	96	134	171	208	245			
55	426	83	76	86	106	147	188	229	270			
60	445	87	79	93	116	160	205	250				
65	463	90	82	101	125	174	222					
70	480	92	84	109	135	187	239					
75	497	95	86	117	145	201	256					
80	514	97	88	124	154	214						
85	529	99	90	132	164	227						
90	545	100	92	140	173	240						
95	560	101	94	148	183	254						
100	574	103	96	156	193							

From tables published by John R. Freeman, C. E. The pressures given are indicated pressures, not effective pressures. Effective pressures would be slightly greater. For discharge coefficients for various types of fire-hose nozzles, see Fig. 7, p. 56.

flammable liquids, and oil-filled electrical equipment, such as transformers. These types are (1) carbon dioxide gas, (2) foam, and (3) water mist (emulsifying).

Carbon Dioxide System.—In case of fire this type employs carbon dioxide gas (CO_2) to dilute the oxygen content of the air in the room containing the flammable liquid to a point where it will not support combustion. This system is particularly applicable to the protection of equipment in rooms or inclosures where the gas can be confined. Gas may be supplied by means of a high-pressure system where liquid carbon dioxide is stored at pressures of about 850 psi at atmospheric temperatures, or from a low-pressure system where the liquid carbon dioxide is stored in tanks at about 325 psi, the tanks being kept at a low temperature by refrigeration. According to NBFU rules, piping systems must be free from scale and, if of iron or steel, must be protected inside and outside from corrosion. Pipe and fittings for high-pressure systems must have a minimum bursting pressure of 6,000 psi. For low-pressure systems protected by relief valves set at 325 psi and having automatic controls designed to keep container pressures below 305 psi, standard-weight (Schedule 40) steel pipe and 300-lb malleable-iron, or equivalent, fittings may be used. Cast-iron fittings are not permitted. All dead-end lines are required to extend at least 2 in. beyond the last orifice and to be closed with a cap or plug. For further requirements, reference may be made to NBFU *Pamphlet* 12.

Foam Extinguishing Systems.—This type employs foam-producing chemicals, in either liquid or dry form, which when combined with water give a foam solution. There are two types of foams: Type I which is suitable for use on flammable liquids not miscible with water, principally petroleum and its products, and Type II which is suitable for use on flammable liquids which are miscible with water, such as alcohols and acetone, and may be used on oil as well.

Systems for outdoor fires using Type I foam must be able to discharge at least ¾ gal of foam per min per sq ft of the largest individual oil-surface area to be extinguished at one time. Systems for outdoor fires using Type II foam must be able to discharge a minimum of ¾ to 2 gal of foam per min depending on the nature of the flammable liquid for which protection is sought. Foam systems for rooms and buildings are designed to discharge not less than 17 cu ft of foam per min per 100 sq ft of area to be protected.

For further requirements, reference may be made to NBFU *Pamphlet* 11.

Water-mist System.—The water-mist type of fire-extinguishing system applies water in finely divided form by means of special spray nozzles or fog nozzles. The fire is extinguished probably by a combination of several effects including smothering, cooling by exposing a large amount of surface to the radiant heat from flames, and, in the case of some oils, forming an incombustible foam or emulsion over the surface which effectively excludes oxygen and thus smothers the flame. The evaporation of the water produces a marked cooling effect which also tends to extinguish the flame. This type of protection is not approved for the general protection of buildings, which is best accomplished by automatic sprinklers or hose streams, but is for use where these may be ineffective or undesirable. The water-mist form of protection has been found particularly adaptable to the protection of large outdoor electrical transformers,[1] but is used also for protection of oil-filled electrical equipment and oil-storage tanks in buildings. Nozzles should be spaced and located so that the hazard protected should be completely covered when the nozzles are in operation; in addition, at least a 10 per cent overlapping factor shall be used. The effective area and effective volume specified by the nozzle manufacturer should be used as a basis in determining the spacing, considering also the nozzle pressure required for the particular design. For flammable liquids, electrical transformers, and other oil hazards, the nozzles should be arranged so as to cover all portions of the hazard protected, the nozzles to be arranged to discharge downward, in general.

A water supply of 1,100 gpm for a transformer on a 60,000-kw turbogenerator and 700 gpm for a 30,000-kw turbogenerator was required in one typical installation for an electric utility. Electrically or hydraulically operated valves admit water to the group of nozzles protecting each piece of equipment when a predetermined temperature is exceeded. Manually controlled systems also are acceptable providing the control valve is located at a safe distance from the hazard protected.

The water-mist system is approved for protection of electrical apparatus, since it is safe and effective, the conductivity of properly atomized spray being low. However, it is recommended that

[1] See "Fighting Power Plant Fires—III," by W. E. Rossnagel, *Power*, April, 1942.

automatic means be provided for cutting off all electricity from live apparatus in advance of the application of water. Clearances between any portion of the equipment and live electrical apparatus should not be less than those shown in the accompanying table, and should be increased wherever possible. Insulation damage is not an important factor since drying out is not a major problem and takes less time than to repair the damage resulting if the fire is not promptly extinguished. For further requirements, reference may be made to NBFU *Pamphlet* 15.

Voltages	Distance, in.	Voltages	Distance, in.
Up to 7,500	6	73,000 to 88,000	52
7,500 to 15,000	12	88,000 to 110,000	64
15,000 to 25,000	17	110,000 to 132,000	77
25,000 to 37,000	24	132,000 to 154,000	89
37,000 to 50,000	32	154,000 to 187,000	106
50,000 to 73,000	44	187,000 to 220,000	124

Pressure Relief Equipment.—One of the most severe hazards to which a vessel may be subjected involves the increase in internal pressure which may result from exposure to external fire of an accidental nature. Extensive tests indicate that pressure relief areas should be designed on the basis of a heat input of 20,000 Btu per hr per sq ft of wetted surface exposed to the fire. For a detailed consideration of formulas necessary to determine sizes and capacities of the relief connections and apparatus and data regarding their application to pressure vessels and atmospheric tanks, see "Requirements for Relief of Overpressure in Vessels Exposed to Fire," by J. J. Duggan, C. H. Gilmour, and P. F. Fisher, *Trans. ASME*, Vol. 66, No. 1, pp. 1–53, January, 1944.

CHAPTER XIV

OIL PIPING

Oil piping systems may be divided into two general classifications: (1) oil-field piping for the production of petroleum and (2) transmission, refinery, and distribution piping. The more important features of the American Petroleum Institute's standard specifications and dimensional standards and of the Oil Section of the American Standard Code for Pressure Piping, ASA B31.1, are abstracted in this chapter. For complete information or latest revisions, reference should be made directly to the API standards[1] and to the Code for Pressure Piping.[2] The API dimensional standards and material specifications applicable to casing, tubing, and pipe used in gas or oil wells are included in this chapter, although requirements for such uses are not included in the Code for Pressure Piping. Many ASA dimensional standards and ASTM materials specifications applicable to oil piping are abstracted in Chap. IV on Pipe, Valves, and Fittings (see pages 348 to 693).

Data on the properties of petroleum and natural gas with their resistance to flow in pipes will be found in Chap. II, on pages 81 to 137 and 200 to 224.[3] A typical selection of heat insulation for oil-refinery piping is furnished in Table VI on page 718. Several excellent textbooks on the petroleum industry are available.[4]

[1] An official list of all API publications with prices of individual standards and codes may be obtained from the American Petroleum Institute, Dallas, Tex.

[2] Copies of the complete code may be obtained from the American Standards Association, 29 West 39th St., New York 18, N.Y.

[3] See also references listed in footnote on p. 200.

[4] Reference to either of the following is suggested:

(a) "Chemical Refining of Petroleum," by V. A. Kalichevsky and B. A. Allen, 2d ed., Reinhold Publishing Corporation, New York, 1942. Concerns the action of various refining agents and chemicals on petroleum and its products, with extensive bibliographies.

(b) "Petroleum Refinery Engineering," by W. L. Nelson, 2d ed., McGraw-Hill Book Company, Inc., New York, 1941.

API Specifications.—The API Committee on the Standardization of Oil Country Tubular Goods has issued the following standard specifications:

5 A Casing, Drill Pipe, and Tubing.
5 B Inspection of Threads (on Oil Country Tubular Goods).
5 F Threads in Fittings (Tentative).
5 G-1 Pipe Line Valves (Tentative).
5 G-2 Gate Valves for Well Control Service (Tentative).
5 G-2A Plug Valves for Well Control Service (Tentative).
5 G-3 Ring Joint Flanges and Flange Unions (Tentative).
5 L Line Pipe (Prepared jointly by API and the Technical Committee of the Natural Gas Association of America).

This committee also has published the following codes:

5 C-1 Recommended Field Practice on Care and Use of Oil Country Tubular Goods.
5 C-2 Setting Depth Properties of Casing.

Other standards formulated by the API Committee on Standardization of Refinery Equipment in cooperation with representatives of manufacturers are:

600 A Standard on Flanged Steel Outside-screw-and-yoke Wedge Gate Valves.
600 B Standard on Flanged Steel Plug Valves.

In addition, the API has cooperated with other trade associations and societies under the procedure of the American Standards Association in formulating the Oil Section of the American Standard Code for Pressure Piping, ASA B31.1.

API Specifications for CASING, DRILL PIPE, AND TUBING

API Standard 5A-1944

Abstracted[1]

The API casing list restricts the grades to be furnished to certain sizes and weights of casing. This restriction in permissible weight-size-grade relationships results in a reduction in the number of items in the casing list from approximately 250 to 95.

[1] See text and footnote 1, p. 1107.

Manufacture.—Drill pipe shall be seamless of open-hearth or electric-furnace steel. Casing and tubing shall be lap-welded of Bessemer, electric-furnace or open-hearth steel, or seamless or electric-welded of open-hearth or electric-furnace steel. (For duration of the war, drill pipe, seamless and electric-welded casing, and tubing may be made from deoxidized acid Bessemer steel. Electric-welded casing and tubing may be made from open-hearth iron.)

API casing regularly is furnished with nonupset ends. Internal-upset ends may be obtained by special agreement.

Physical Properties.—Physical properties of drill pipe are given in Table I, of casing and tubing in Table II.

TABLE I.—PHYSICAL PROPERTIES OF API DRILL PIPE
(Table 3 of API 5A)

	API grade symbols		
	C	*D*	*E*
Yield strength, min, psi[1]	45,000	55,000	75,000
Tensile strength, min, psi	75,000	95,000	100,000
Percentage elongation, 2 in., min	20	18	18

[1] Stress required to produce a total elongation of 0.5 per cent of the gage length.

TABLE II.—PHYSICAL PROPERTIES OF API CASING AND TUBING
(Table 4 of API 5A)

	API grade symbols			
	F-25	*H*-40	*J*-55	*N*-80
Yield strength, min, psi	25,000	40,000	55,000	80,000
Tensile strength, min, psi	40,000	60,000	75,000	100,000
Percentage elongation, 8 in., min	20			
Percentage elongation, 2 in., min	40	27	20[1]	16

[1] For duration of the war the percentage of elongation for J-55 casing and tubing may be 18 as a means of [illegible] purposes.

No[illegible] Casing and tubing with round threads [illegible] with sharp thread casing and tubing. Old style material may be ordered [illegible] for maintenance purposes. Casing conforming to dimensions of California BX, Diamond BX, Boston, South Penn, and Boston [illegible] only on special order.

Hydrostatic Tests.—Each length of casing and tubing shall be tested at the mill to a pressure based on 60 per cent of the specified yield strengths. Hydrostatic tests are not required on drill pipe.

Threads.—Each end of casing, drill pipe, and tubing shall be threaded, except casing for welding and drill pipe with non-API joints. Casing in sizes 13⅜ in. O.D. and under shall have round threads; sizes 16 in. and 20 in. shall have sharp

threads. Casing of all grades shall be supplied with short threads and couplings. Grades J-55 and N-80 may be supplied with long threads and couplings when so specified on the orders. See Fig. 1 and Tables IV and V for thread dimensions.

Couplings.—Couplings for furnace-welded casing or tubing may be made of wrought iron or steel, seamless or welded. Couplings for drill pipe, seamless and electric-welded casing, and tubing shall be made of seamless material at least [illegible] ion of the war, Grade J-55 couplings may be [illegible] asing.)

[illegible] tubing shall be furnished in lengths as shown in Table III.

TABLE III.—RANGE LENGTH OF CASING, DRILL PIPE, AND TUBING (Table 5 of API 5A) (All lengths in feet)

	Range		
	1	2	3
Casing:			
Total range length, incl	16 to 25	25 to 34	34 or more
Range length for 95% or more of carload:			
Permissible variation, max	6	5	6
Minimum permissible length	18	28	36
Range length for 5% or less of carload:			
Permissible variation, max	9	9	
Minimum permissible length	16	25	34
Jointers:[1] min length of shortest piece	5	5	5
Drill pipe:			
Total range length, incl	18 to 22	27 to 30	38 or more
Range length for 95% or more of carload:			
Permissible variation, max	2		
Minimum permissible length	20		
Range length for 90% or more of carload:			
Permissible variation, max	..	2	3
Minimum permissible length	..	27	38
Range length for 5% or less of carload:			
Permissible variation, max	4		
Minimum permissible length	18		
Range length for 10% or less of carload:			
Permissible variation, max	..	3	4
Minimum permissible length	..	27	38
Tubing:			
Total range length, incl	20 to 24	28 to 32	
Range length for 100% of carload:			
Permissible variation, max	2	2	
Minimum permissible length	20	28	

[1] Jointers, two lengths connected by coupling, may be shipped to a maximum of 5% of the order; they are not permissible on drill pipe and tubing.

Marking.—Couplings shall be stamped with the API monogram and all casing, drill pipe, and tubing shall be marked by stamping with a steel die within 1 ft of the coupling end to give the following data in the order named:

Manufacturer's name or mark.

API monogram.

* Grade (letter symbol); drill pipe = C, D, or E,
casing or tubing = F, H, J, or N.

* Class; furnace lap welded = L,
electric welded = E,
seamless = S, except that letter S is not required on drill pipe.

* Material; open-hearth iron = I,
wrought iron = WI,
rephosphorized open-hearth lap-welded steel = R, Bessemer = B.

* Nominal weight per foot, pounds.

The items above marked with an asterisk also are to be stenciled on each length of pipe in addition to the following:

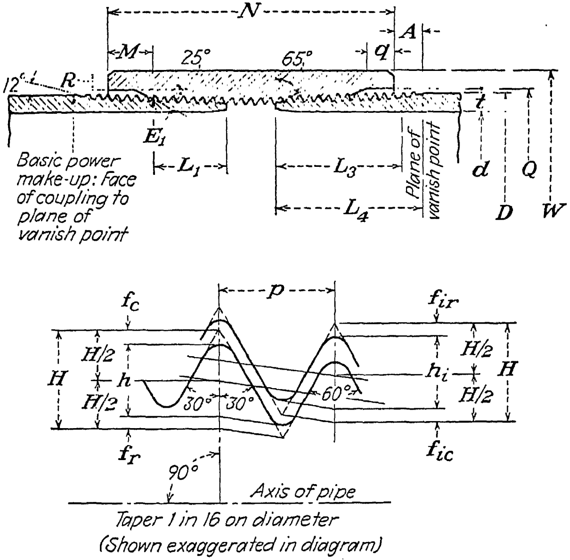
API size (as given in tables in API 5A).

Wall thickness (decimals of an inch).

Length (English units, feet and inches near coupling end).

Total weight (English units near coupling end on sizes 4½ in. O.D. and larger, not required on dipped or coated pipe).

NOTE.—"Material" markings not mandatory on open-hearth steel or electric-furnace steel. Marking of Bessemer steel with letter B mandatory only when used in seamless or electric-welded casing or tubing under the war emergency measure adopted November, 1942 and November, 1943; this marking not mandatory when Bessemer steel is used in furnace lap-welded casing and tubing.



THREAD HEIGHT DIMENSIONS
(Inches)

Thread element	8 threads per inch $p = 0.125$
$H = 0.866p$	0.10825
$h = h_i = 0.626p - 0.007$	0.07125
$f_r = f_{ir} = 0.120p + 0.002$	0.01700
$f_c = f_{ic} = 0.120p + 0.005$	0.02000

FIG. 1.—API standard 5A round-thread casing, short threads and couplings, hand-tight assembly.

TABLE IV.—API STANDARD CASING, SHORT THREADS AND COUPLINGS. PIPE AND THREAD DIMENSIONS (See Fig. 1)
(Table 8 of API 5A)
(All dimensions in inches at 68 deg F)

1	2	3	4	5	6	7	8	9
Pipe				Threads				
Size, outside diameter	Inside diameter	Wall thickness	Nominal weight, lb per ft	Length, end of pipe to handtight plane	Effective length	Total length, end of pipe to vanish point	Pitch diameter at hand-tight plane	End of pipe to center of coupling, made-up
D	d	t		L_1	L_3	L_4	E_1	
4½	4.090	0.205	9.50	1.055	1.715	2.000	4.41174	0.500
4½	4.000	0.250	11.60	1.055	1.715	2.000	4.41174	0.500
4½	3.920	0.290	13.50	1.055	1.715	2.000	4.41174	0.500
5	4.560	0.220	11.50	1.555	2.215	2.500	4.91174	0.750
5	4.494	0.253	13.00	1.805	2.465	2.750	4.91174	0.500
5	4.408	0.296	15.00	1.805	2.465	2.750	4.91174	0.500
5	4.276	0.362	18.00	1.805	2.465	2.750	4.91174	0.500
5½	5.044	0.228	13.00	1.680	2.340	2.625	5.41174	0.750
5½	5.012	0.244	14.00	1.930	2.590	2.875	5.41174	0.500
5½	4.950	0.275	15.50	1.930	2.590	2.875	5.41174	0.500
5½	4.892	0.304	17.00	1.930	2.590	2.875	5.41174	0.500
5½	4.778	0.361	20.00	1.930	2.590	2.875	5.41174	0.500
5½	4.670	0.415	23.00	1.930	2.590	2.875	5.41174	0.500
6	5.524	0.238	15.00	2.055	2.715	3.000	5.91174	0.500
6	5.424	0.288	18.00	2.055	2.715	3.000	5.91174	0.500
6	5.352	0.324	20.00	2.055	2.715	3.000	5.91174	0.500
6	5.240	0.380	23.00	2.055	2.715	3.000	5.91174	0.500
6⅝	6.135	0.245	17.00	2.180	2.840	3.125	6.53674	0.500
6⅝	6.049	0.288	20.00	2.180	2.840	3.125	6.53674	0.500
6⅝	5.921	0.352	24.00	2.180	2.840	3.125	6.53674	0.500
6⅝	5.791	0.417	28.00	2.180	2.840	3.125	6.53674	0.500
6⅝	5.675	0.475	32.00	2.180	2.840	3.125	6.53674	0.500
7	6.538	0.231	17.00	1.430	2.090	2.375	6.91174	1.250
7	6.456	0.272	20.00	2.180	2.840	3.125	6.91174	0.500
7	6.366	0.317	23.00	2.180	2.840	3.125	6.91174	0.500
7	6.276	0.362	26.00	2.180	2.840	3.125	6.91174	0.500
7	6.184	0.408	29.00	2.180	2.840	3.125	6.91174	0.500
7	6.094	0.453	32.00	2.180	2.840	3.125	6.91174	0.500
7	6.004	0.498	35.00	2.180	2.840	3.125	6.91174	0.500
7	5.920	0.540	38.00	2.180	2.840	3.125	6.91174	0.500
7⅝	7.125	0.250	20.00	1.863	2.590	2.875	7.53255	0.875
7⅝	7.025	0.300	24.00	2.238	2.965	3.250	7.53255	0.500
7⅝	6.969	0.328	26.40	2.238	2.965	3.250	7.53255	0.500
7⅝	6.875	0.375	29.70	2.238	2.965	3.250	7.53255	0.500
7⅝	6.765	0.430	33.70	2.238	2.965	3.250	7.53255	0.500
7⅝	6.625	0.500	39.00	2.238	2.965	3.250	7.53255	0.500

TABLE IV.—(*Concluded*)

1	2	3	4	5	6	7	8	9
	Pipe			Threads				
Size, outside diameter	Inside diameter	Wall ickness	ominal weight, per ft	ength, end of pipe to andtight plane	ffective length	Total length, end of pipe to vanish point	Pitch diameter t hand-tight plane	End of pipe to enter of coupling, made-up
D	d	t		L_1	L_3	L_4	E_1	
	8.097	.264	24.00	1.988	2.715	3.000	8.53255	0.875
	8.017	.304	28.00	2.363	3.090	3.375	8.5325	0.500
	7.921	.352	32.00	2.363	3.090	3.375	8.5325	0.500
	7.825	.400	36.00	2.363	3.090	3.375	8.5325	0.500
	7.725	.450	40.00	2.363	3.090	3.375	8.5325	0.500
	7.625	.500	44.00	2.363	3.090	3.375	8.5325	0.500
	7.511	.557	49.00	2.363	3.090	3.375	8.5325	0.500
	9.063	0.281	29.30	2.238	2.965	3.250	9.5325	0.625
9⅝	9.001	0.312	32.30	2.363	3.090	3.375	9.5325	0.500
9⅝	8.921	0.352	36.00	2.363	3.090	3.375	9.5325	0.500
9⅝	8.835	0.395	40.00	2.363	3.090	3.375	9.5325	0.500
9⅝	8.755	0.435	43.50	2.363	3.090	3.375	9.5325	0.500
	8.681	0.472	47.00	2.363	3.090	3.375	9.5325	0.500
	8.535	0.545	53.50	2.363	3.090	3.375	9.5325	0.500
	10.192	0.279	32.75	1.738	2.465	2.750	10.6575	1.250
	10.050	0.350	40.50	2.488	3.215	3.500	10.6575	0.500
	9.950	0.400	45.50	2.488	3.215	3.500	10.6575	0.500
10¾	9.850	0.450	51.00	2.488	3.215	3.500	10.6575	0.500
10¾	9.760	0.495	55.50	2.488	3.215	3.500	10.6575	0.500
11¾	11.150	0.300	38.00	2.238	2.965	3.250	11.6575	0.750
11¾	11.084	0.333	42.00	2.488	3.215	3.500	11.6575	0.500
11¾	11.000	0.375	47.00	2.488	3.215	3.500	11.6575	0.500
11¾	10.880	0.435	54.00	2.488	3.215	3.500	11.6575	0.500
11¾	10.772	0.489	60.00	2.488	3.215	3.500	11.6575	0.500
13⅜	12.715	0.330	48.00	2.488	3.215	3.500	13.2825	0.500
13⅜	12.615	0.380	54.50	2.488	3.215	3.500	13.28255	0.500
13⅜	12.51	0.430	61.00	2.488	3.215	3.500	13.28255	0.500
13⅜	12.41	0.480	68.00	2.488	3.215	3.500	13.28255	0.500
13⅜	12.34	0.514	72.00	2.488	3.215	3.500	13.28255	0.500
16	15.375	0.312	55.00	2.862	3.431	3.875	15.87575	0.625
16	15.25	0.375	65.00	2.862	3.431	3.875	15.87575	0.625
16	15.125	0.437	75.00	2.862	3.431	3.875	15.87575	0.625
16	15.01	0.495	84.00	2.	3.431	3.875	15.87575	0.625
20	19.16	0.417	90.0	2.862	3.431	3.875	19.87575	0.625

Sizes 13⅜ in. and under, are round thread; sizes 16 in. and 20 in. are sharp thread.
All sizes 8 threads per in.; included taper, 0.0625 in. per in.

TABLE V.—API STANDARD TUBING, NON-UPSET. PIPE AND THREAD DIMENSIONS (See Fig. 1)
(Table 20 of API 5A)
(All dimensions in inches at 68 deg F)

1	2	3	4	5	6	7	8	9
Pipe					Threads			
Size nominal	Outside diameter D	Inside diameter d	Wall thickness t	Nominal weight, lb per ft	Length, end of pipe to hand-tight plane	Effective length L_3	Total length, end of pipe to vanish point	Pitch diameter at hand-tight plane E_1
1½	1.900	1.610	0.145	2.75	0.729	1.206	1.375	1.83826
2	2.375	2.041	0.167	4.00	0.979	1.456	1.625	2.31326
†2	2.375	1.995	0.190	4.60	0.979	1.456	1.625	2.31326
2½	2.875	2.441	0.217	6.40	1.417	1.894	2.063	2.81326
*3	3.500	3.068	0.216	7.70	1.667	2.144	2.313	3.43826
3	3.500	2.992	0.254	9.20	1.667	2.144	2.313	3.43826
†3	3.500	2.922	0.289	10.20	1.667	2.144	2.313	3.43826
3½	4.000	3.548	0.226	9.50	1.591	2.140	2.375	3.91395
4	4.500	3.958	0.271	12.60	1.779	2.328	2.563	4.41395

Sizes 1½ to 3 in. incl., 10 threads per inch; sizes, 3½ to 4 in., 8 threads per inch; included taper, all sizes, 0.0625 in. per in.

Dimension J, end of pipe to center of coupling, made up, all sizes: 0.500 in.

* Tentative.

† On direct mill shipment only.

API Specifications for THE INSPECTION OF THREADS ON OIL COUNTRY TUBULAR GOODS

API Standard 5B-1941

Abstracted[1]

This specification supplements API Standards 5-A, 5-F, and 5-L. It covers instruments and methods for the inspection of thread elements such as lead, taper, thread height, angle, and thread form to determine compliance with threads specified in above standards.

The measurement of taper, lead, and height of thread are required to be made at 1-in. intervals, except where length of threads is less than 1 in.

Taper.—Taper is defined as the increase in the pitch diameter of the threads per foot of threads. Limiting values for taper of threads on pipe and couplings having a nominal taper of ¾ in. per ft are 13⁄16 in. maximum and 23⁄32 in. minimum. Measurements are made with a dial gage thread caliper.

[1] See text and footnote, 1, p. 1107.

Lead.—For inspection purposes, lead is defined as the distance from a point on a thread to a corresponding point on the next thread measured parallel to the axis of the threaded section. The tolerance on lead is ± 0.003 in. in any inch of thread and ± 0.006 in. cumulative in the total length of perfect threads. Actual measurements are made parallel to the cone of taper rather than to the axis. The standard template for adjustment of lead gages is corrected for this difference.

Height of Thread.—For inspection purposes, height of thread is defined as the distance between the crest and the root normal to the surface of the cone of the taper. The permissible errors from the basic height of thread for the API round thread form are $+0.002$ in. and -0.004 in.

Thread Angle.—Thread angle is the included angle between the flanks of the thread. The threads are at right angles to the pipe axis. The tolerance is 1½ deg from the basic thread angle of 60 deg. Measurements are made with a thread contour microscope.

Thread inspection may be made at option of the purchaser to any portion of the product either during manufacture, just prior to shipment, at district warehouse, or after delivery at destination.

API Tentative Specification for THREADS IN VALVES, FITTINGS, AND FLANGES

API Standard 5F, 1940

Abstracted[1]

This specification covers dimensions and gaging of female threads in valves, fittings, and flanges, but not in couplings for use with API 5A casing, drill pipe, and tubing and API 5L line pipe.

Lighter weight valves, fittings, and flanges ordinarily used for 125- to 300-lb steam service when threaded in accordance with ASA Std B2.1, American Standard Pipe Threads, can be used satisfactorily with pipe threaded with API line-pipe threads.

The basic specifications for threads correspond exactly with thread details for the different types of thread given in API 5A and 5L for casing, tubing, drill pipe, and line pipe, respectively.[2]

The minimum length of thread in valves and fittings is equal to the effective length of male thread given in API 5A and 5L. When not counterbored, the minimum length of thread is measured from the face of valve or fitting and includes the chamfer. When counterbored, the length of thread is measured from points distance V from the face of the valve or fitting.

$$V = X - \frac{P}{2}$$

[1] Abstract contains revision of *Supplement* 1, April, 1941. (See text and footnote 1, p. 1107.)

[2] When valves and fittings designed for steam service with shorter threads are used with API tubular goods, care should be taken that there is no interference or shouldering on the inside beyond the end of the shorter thread in valve or fitting. In the case of line pipe the 2-in. size is the only one having threads longer than in ASA B2.1.

where X = depth of counterbore plus one-half the difference between the diameter of counterbore and the pitch diameter.

$$P = \text{pitch} = \frac{1}{\text{number of threads}}$$

The length of thread in flanges threaded with line pipe thread, API 5L, is in accordance with requirements of ASA B16e (see page 633).

The length of thread in flanges threaded with API casing, drill pipe, or tubing threads is taken equal to the effective length of male threads in API 5A plus ⅜ in. for clearance, except in cases where resulting depth of counterbore varies only slightly from the depth of counterbore determined for valves and fittings. In those cases it is taken the same as for fittings. In some instances the length through hub of standard ASA flanges is more than sufficient and the depth of counterbore is increased accordingly. In all cases the thread extends from the bottom of the counterbore to the face of the flange.

The normal length of engagement between taper male-and-female threads when screwed together by hand is shown in tables for each type of thread and class of tubular product.

Female threads for use with API line pipe and female threads for use with American Standard taper pipe threads are identical in pitch diameter at the large end of the thread. Fittings, valves, and flanges threaded to the American Standard may be used with pipe threaded to the API line pipe standard except in the 2-in. pipe size which has a longer thread.

Valves.—Valves for pipe-line service are specifically covered by API Standard 5G-1 which makes reference to design details of API Standard 600-A and facing dimensions of API Standard 5G-3. Valves for refinery use are covered in API Standards 600-A and 600-B. Drilling-through and flow-line valves for well-control service are covered by API Standards 5G-2 and 5G-2A for solid-wedge and double-disk gate valves and for round-opening steel plug valves, respectively. The bodies and ports of well-control valves intended for drilling through have minimum throat bore dimensions sufficiently large to permit passage of the various drilling tools. Oversized bores are provided in some sizes so that one string of casing may be suspended inside another.

The working pressure ratings for pipe-line valves are the same as the 100 F ratings for oil service given for carbon steel flanges and flanged fittings with standard facings in ASA B16e (see Table CV, page 628). The cold working pressure ratings for well-control valves are given in API 5G-2 and 5G-2A together with the corresponding nominal ASA B16e ratings. For ASA B16e primary service ratings 600 lb and higher, the ratings for well-control valves are higher than the hydrostatic shell test pressures specified in ASA B16e for carbon-molybdenum flanges with ring joints. Hydrostatic tests of well-control valves are made at pressures two times

their ratings. Kerosene often is specified both for hydrostatic test of the valve body and for testing the seat for tightness. In some cases, the test pressure used with kerosene is only 100 psi, the normal hydrostatic test being made with water.

API Tentative Standard on PIPE-LINE VALVES

Serial Designation 5G-1, 1938

Abstracted[1]

This specification covers flanged, rising stem and nonrising stem, double-disk and solid wedge-gate valves, lubricated plug valves, and swing check valves for pipe line service. Except for drilling-through valves and flow-line valves which are covered in API Standards 5G-2 and G-2A, this specification also is suitable for general use in producing operations such as lease lines and general service on drilling rigs and producing wells.

Dimensions.—Flanged steel rising-stem wedge-gate valves covered by this standard have the same contact-face-to-contact-face and flange dimensions as valves covered by API Standard 600-A (see page 1141) and shall be finished in strict accordance with that specification as regards design details. The following are the corresponding classes:

5G-1	API 600-A and ASA B16e
230 lb	Class 150
500 lb	Class 300
670 lb	Class 400
1,000 lb	Class 600

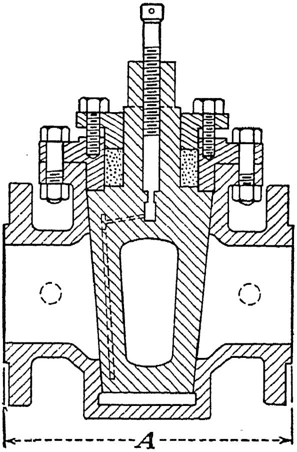


FIG. 2.—Plug valve.

Contact-face-to-contact-face dimensions, drilling, and flange dimensions, except when ring joints are used, shall be as shown in Tables VI and VII. For ring-groove facing, the flange thicknesses and face-to-face dimensions are increased to provide for the ring grooves in accordance with API Standard 5G-3 (see page 1125).

Materials.—Body and bonnet materials shall be in accordance with ASTM Specification A126 if of cast iron, A95 if of cast steel, or A105 if of forged steel. (For abstracts of these specifications see Chap. IV.) Factory repairs by welding, peening, plugging, or filling on cast-iron and brass valve parts are prohibited.

Trim.—Trim for iron valves shall be bronze, unless otherwise specified on the order. Standard trim for steel valves shall be 11½ to 13½ per cent chrome alloy, unless otherwise specified on the order.

[1] See text and footnote 1, p. 1107.

Plug Valves.—Valves furnished to the face-to-face dimensions specified herein will, in some cases, have openings with smaller areas than usually provided for gate valves. Manufacturers of plug valves shall furnish purchasers, upon request, all available information on pressure loss through their plug valves.

TABLE VI.—VALVES, DOUBLE-DISK AND WEDGE-GATE AND PLUG TYPES, FLANGED IRON AND FLANGED STEEL, CONTACT-FACE-TO-CONTACT-FACE AND FLANGE DIMENSIONS (Tables 1 and 2 of API 5G-1)

Nominal pipe size, inches	Contact-face-to-contact-face dimensions, inches							
	Flanged iron[1]				Flanged steel[5]			
	175-lb[2]	350-lb[3]	500-lb[3]	800-lb[4]	230-lb[6,7]	500-lb[6,8]	670-lb[9]	1,000-lb[9]
2	7	7¼	8½	11½	7	8½	....	11½
2½	7½	8	9½	13	7½	9½	....	13
3	8	9¼	11⅛	14	8	11⅛	....	14
4	9	10½	12	17	9	12	16	17
6	10½	13	15⅞	22	10½	15⅞	19½	22
8	11½	14¼	16½	26	11½	16½	23½	26
10	13	16¾	18	31	13	18	26½	31
12	14	17½	19¾	33	14	19¾	30	33

[1] Flange thicknesses and drilling [illegible] standards (abstracted in Chap. IV):

175-lb: ASA B16a (American 125-lb Standard).
350-lb: ASA B16b (American 250-lb Standard).
500-lb: ASA B16b (American 250-lb Standard).
800-lb: ASA B16b1 (American 800-lb Standard).

[2] 175-lb has no raised face but is interchangeable with 230-lb steel (Series 15).
[3] The contact-face-to-contact-face dimensions of the 350- and 500-lb valves include a 1/16-in. raised face. 500-lb are interchangeable with 500-lb steel (Series 30).
[4] The contact-face-to-contact-face dimensions of the 800-lb valves include a ¼-in. raised face. 800-lb is interchangeable with 1,000-lb steel (Series 60).
[5] Flange thickness, drilling templates, and spot facing shall conform to the following standards (abstracted in Chap. IV):

230-lb: ASA B16e (American 150-lb Standard).
500-lb: ASA B16e (American 300-lb Standard).
670-lb: ASA B16e (American 400-lb Standard).
1,000-lb: ASA B16e (American 600-lb Standard).

[6] The contact-face-to-contact-face dimensions of the 230-lb and 500-lb valves include a 1/16-in. raised face.
[7] 230-lb is interchangeable with 175-lb iron.
[8] 500-lb is interchangeable with 500-lb iron.
[9] The contact-face-to-contact-face dimensions of the 670-lb and the 1,000-lb valves include a ¼-in. raised face. 1,000-lb is interchangeable with 800-lb iron. For 670-lb valves in sizes 2 in., 2½ in., and 3 in., use dimensions for 1,000-lb valves.
NOTE.—Double-disk valves are not commonly made of steel. If furnished of steel, they shall conform to this standard.

Contact Face to Contact Face.—Dimension *A* of Fig. 2 is defined as the largest over-all distance measured on the horizontal center line of valves between machined surfaces of the bolting flanges of the valve. It includes the raised

faces of raised-face flanges. In this specification only the 175-lb iron valves have flat-faced flanges; all others have raised-face flanges.

Tests.—All valves shall be subjected to a hydrostatic body test of two times the rated working pressure. After the body test, valves shall be tested for seat tightness at a hydrostatic pressure equal to the rated working pressure. Pressures shall be applied successively on each side of the gate or plug with the other side open to atmosphere. Check valves shall be tested with pressure against the disk only.

When so specified on the order, valve seats shall be tested with air at 100 psi applied on each side of gate or plug with other side under water.

TABLE VII.—SWING CHECK VALVES, FLANGED IRON AND FLANGED STEEL, CONTACT-FACE-TO-CONTACT-FACE AND FLANGE DIMENSIONS

(Tables 3 and 4 of API 5G1)

Nominal pipe size, in.	Contact-face-to-contact-face dimensions, in.						
	Flanged iron[1]			Flanged steel[6]			
	175-lb[2]	500-lb[3,4]	800-lb[5]	230-lb[7,8]	500-lb[7,9]	670-lb[10]	1,000-lb[10]
2	8	10½	11½	8	10½	....	11½
2½	8½	11½	13	8½	11½	....	13
3	9½	12½	14	9½	12½	....	14
4	11½	14	17	11½	14	16	17
6	14	17½	22	14	17½	19½	22
8	....	21	26	....	21	23½	26
10	....	24½	31	....	24½	26½	31
12	....	28	33	....	28	30	33

[1] Flange thicknesses and drilling templates shall conform to the following standards (abstracted in Chap. IV):

175-lb: ASA B16a (American 125-lb Standard).
500-lb: ASA B16a (American 250-lb Standard).
800-lb: ASA B16.1 (American 800-lb Standard).

[2] 175-lb has no raised face but is interchangeable with 230-lb steel (Series 15) check.
[3] The contact-face-to-contact-face dimensions of the 500-lb valves include a 1/16-in. raised face.
[4] 500-lb is interchangeable with 500-lb steel (Series 30) check.
[5] The contact-face-to-contact-face dimensions of the 800-lb include a 1/4-in. raised face. 800-lb is interchangeable with 1,000-lb steel (Series 60) check.
[6] Flange thickness, drilling templates, and spot facing shall conform to the following standards (abstracted in Chap. IV):

230-lb: ASA B16e (American 150-lb Standard).
500-lb: ASA B16e (American 300-lb Standard).
670-lb: ASA B16e (American 400-lb Standard).
1000-lb: ASA B16e (American 600-lb Standard).

[7] The contact-face-to-contact-face dimensions of the 230-lb and 500-lb valves include a 1/16-in. raised face.
[8] 230-lb is interchangeable with 175-lb iron check valves.
[9] 500-lb is interchangeable with 500-lb iron check valves.
[10] The contact-face-to-contact-face dimensions of the 670-lb and the 1000-lb valves include a 1/4-in. raised face. 1,000-lb is interchangeable with 800-lb iron check valves. For 670-lb valves in sizes 2 in., 2½ in., and 3 in., use dimensions for 1,000-lb valves.

Marking.—Valves shall be marked in accordance with the Standard Marking System SP-25 of the Manufacturers Standardization Society of the Valve and Fittings Industry (see page 687).

API Tentative Standard on
SOLID-WEDGE AND DOUBLE-DISK GATE VALVES FOR WELL-CONTROL SERVICE

Serial Designation 5G-2, 1941

Abstracted[1]

This standard covers solid-wedge and double-disk flanged nonrising stem (N.R.S.) Steel Drilling-through Flow-line (Well-control) Gate Valves having integrally forged or cast flanges.

Classification.—Valves shall be identified by the following designations:

Cold Working Pressure, API 5G-2, Lb	Corresponding Nominal, ASA B16e Rating—Lb
500	300
1,000	400
2,000	600
3,000	900
5,000	1,500

Sizes.—For sizes of valves in each class, see Table VIII. Where two sizes are specified as 10 by 11, the first number indicates flange size while the second gives the nominal bore. The oversize bores are required where one string of casing is suspended inside another.

Material.—Chemical analyses are purposely omitted but physical properties of body metal for the different classifications shall be as tabulated below:

PHYSICAL PROPERTIES
(Table 1 of 5G-2)

	500 and 1,000 psi working pressure	2,000, 3,000, and 5,000 psi working pressure
Tensile strength, psi	70,000	100,000
Yield strength, psi	36,000	65,000
Elongation in 2 in.	22%	[illegible]
Reduction in area	30%	

Dimensions.—End-to-end dimensions and minimum throat bore diameters shall be as shown in Table VIII. Flange and facing proportions shall be in accordance with API Standard 5G-3 (see page 1125).

Design.—Bodies and bonnets shall be of circular type for working pressures 3,000 lb and above. Bonnet joints may be of the ring-joint, tongue-and-groove,

[1] See text and footnote 1, p. 1107.

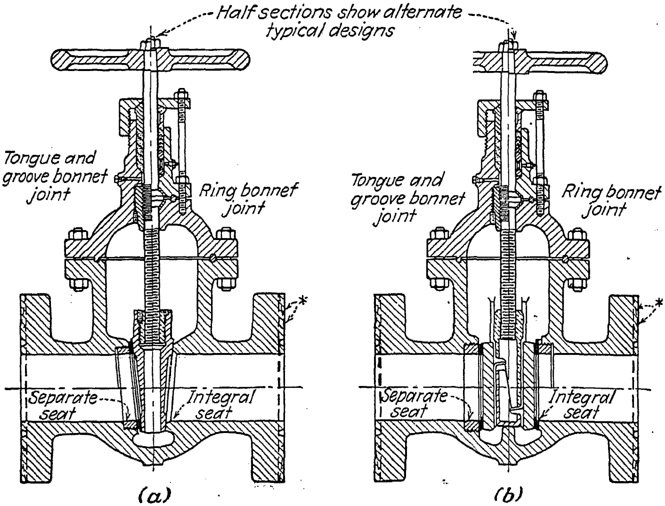
TABLE VIII.—DIMENSIONS[1] OF SOLID-WEDGE AND DOUBLE-DISC GATE VALVES FOR WELL-CONTROL SERVICE, OIL FIELD USE
(Tables 2 to 6, inclusive, of API Standard 5G-2)
(All dimensions in inches at 68 F)

Nominal size of valve, inches	Minimum throat bore		500-lb end-to-end		1,000-lb end-to-end		2,000-lb end-to-end		3,000-lb end-to-end		5,000-lb end-to-end	
	Drilling-through valve	Flow-line valve	Drilling-through valve	Flow-line valve	Drilling-through valve	Flow-line valve	Drilling-through valve	Flow-line valve	Drilling-through valve	Flow-line valve	Drilling-through valve	Flow-line valve
1 1/2	....	1 5/8	....	7 15/16	....	9 7/16	....	9 7/16	....	11 3/4	....	11 15/16
2	....	2	....	9 1/8	....	11 5/8	....	11 5/8	....	14 3/8	....	14 5/8
2 1/2	....	2 1/2	....	10 1/8	....	13 1/8	....	13 1/8	....	15 7/8	....	16 5/8
3	....	3	....	11 3/4	....	14 1/8	....	14 1/8	....	17 1/8	....	18 5/8
4	4 1/4	4	16 5/8	12 5/8	18 1/8	16 1/8	19 1/8	17 1/8	20 1/8	20 1/8	23 5/8	21 5/8
5	5 7/16	5	18 5/8	....	20 1/8	....	22 1/8	....	24 1/8	....	28 5/8	
6	6 5/8	6	19 5/8	16 1/2	21 5/8	19 5/8	24 1/8	22 1/8	26 1/8	24 1/8	30	
6 × 7	7	7	....	....	23 5/8	21 5/8	26 1/8	24 1/8	28 1/8	26 1/8	32	
8	8 7/16	8	22 5/8	17 1/8	25 5/8	23 5/8	28 1/8	26 1/8	31 1/8	29 1/8	35 1/8	
8 × 9	9	9	....	....	....	....	30 1/8	28 1/8	33 1/8	31 1/8	37 1/8	
10	10 3/16	10	25 5/8	18 5/8	23 5/8	26 5/8	33 1/8	31 1/8	35 1/8			
10 × 11	11	....	....	....	....	....	35 1/8	....	37 1/8			
12	12 7/8	....	25 5/8	....	32 1/8	....	35 1/8	....	40 1/8			
12 × 13	13 5/8	....	....	....	....	....	37 1/8	....	42 1/8			
16	15 1/2	....	35 5/8	....	37 5/8							
20 × 18[2]	18	....	38 5/8	....	40 5/8	....	45 1/8					
20	20 7/8	....	41 3/4	....	43 3/4							

[1] Minimum wall thicknesses and outside diameters of end flanges are same as for corresponding ASA B16e flange ratings. See API Standard 5G-3 for dimensions of flanges, groove dimensions, and API ring numbers.
[2] This size inactive; obtainable only on special order.

male-and-female, parent-metal-to-parent-metal, or welded-on type. A reduction in wall thicknesses over that required by ASA B16e may be permitted where a higher strength material is used than specified above upon agreement between the manufacturer and the purchaser.

The sketches shown in Fig. 3 are composite for the purpose of showing typical variation in individual details. However, a product utilizing a combination of



*End flanges for ring joints may be obtained full face, beveled, or with ASA B16e face as required.

FIG. 3.—Typical nonrising stem gate valves: (*a*) solid-wedge valve, (*b*) double-disk valve.

these details (except when such combination may be specifically prohibited in the text) or similar construction will be acceptable, provided it complies with this specification in all other respects.

API Tentative Standard on FLANGED ROUND-OPENING STEEL PLUG VALVES FOR DRILLING AND PRODUCTION WELL-CONTROL SERVICE

Serial Designation 5G-2A, 1941

Abstracted[1]

This standard covers round-opening, steel drilling-through and flow-line plug valves with integrally cast or forged steel flanges having a single through circular passage.

[1] See text and footnote 1, p. 1107.

Classification.—Valves shall be identified by the following designations:

Maximum Cold Working Pressure, API—5G-2A, lb	Corresponding Nominal ASA B16e Rating, lb
500	300
1,000	400
2,000	600
3,000[1]	900[1]
5,000	1,500

[1] The flanges for these valves in sizes less than 3 in. are not ASA standard.

Sizes.—For sizes of valves in each class, see Table IX. Where two sizes are specified, *e.g.*, 10 × 11, the first number indicates the nominal flange size while the second number gives the minimum throat and plug bore.

Materials.—Chemical analyses are omitted but physical properties of steel used for body and cover of valves for ratings of 1,000 lb and under shall have a

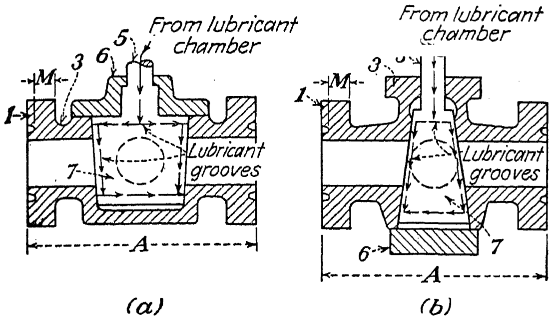


FIG. 4.—Flanged steel plug valves: (*a*) round-port full-bore type, (*b*) inverted tapered-plug, round-port, full-bore type. (1) ring-groove facing; (3) body; (5) plug stem; (6) cover; (7) plug; (*M*) basic (minimum) flange thickness.

tensile strength exceeding 70,000 psi, while for ratings of 2,000 lb and higher the tensile strength shall exceed 100,000 psi. See abstract of API 5G2 on page 1120 for other physical properties.

Dimensions.—End-to-end dimensions and minimum throat bore dimensions are given in Table IX. Flanges and facing proportions shall be in accordance with API 5-G3 (see page 1125).

Design.—Bodies shall be designed to provide against any permanent distortion when under a test pressure equal to two times the working pressure, and to resist distortion under lubricant pressure in conjunction with operating condition. Cover joints may be of the ring-joint, tongue-and-groove, male-and-female, or threaded type. Plug stems shall be heavy enough to resist distortion under maximum required turning load. Suitable indication of the position of the plug port opening shall be provided.

Figure 4 is intended to illustrate some of the basic valve types that are considered as being covered by this standard. Reference to these types, however, is to be construed neither as prohibiting such other designs as may be offered nor

TABLE IX.—DIMENSIONS[1] OF FLANGED PLUG VALVES FOR WELL-CONTROL SERVICE
(Tables 2 to 6, inclusive, of API Standard 5G-2A)
(All dimensions in inches at 68 F)

Nominal size of valve, inches	Minimum throat bore		500-lb end-to-end		1,000-lb end-to-end		2,000-lb end-to-end		3,000-lb end-to-end		5,000-lb end-to-end	
	Drilling-through valve	Flow-line valve	Drilling-through valve	Flow-line valve	Drilling-through valve	Flow-line valve	Drilling-through valve	Flow-line valve	Drilling through valve	Flow-line valve	Drilling-through valve	Flow-line valve
1½	....	1⅝	....	9 15/16	....	12 7/16	....	12 7/16	....	13 15/16	....	13⅝
2	....	2	....	11¾	....	13⅛	....	13⅛	....	15⅛	....	15½
2½	....	2½	....	13⅝	....	15⅛	....	15⅛	....	17½	....	18
3	....	3	....	15⅞	....	17⅝	....	17⅝	....	18⅝	....	20¾
4	....	4	....	18⅝	....	19⅛	....	20⅛	....	22⅛	24¾	24¾
5	5 7/16	...	21⅝	....	22⅛	....	25⅛	....	26⅛	....	31⅛	
6	6⅝	...	21⅝	....	26⅛	....	28⅝	....	30⅛	....	36⅛	
6 × 7	7	...	25⅛	....	26⅝	....	29⅛	....	31⅝	....	38⅛	
8	8 7/16	...	29⅝	....	31⅛	....	34⅝	....	36⅝	....	43⅛	
8 × 9	9	...	30⅝	....	32⅛	....	36⅛	....	38⅛	....	45⅛	
10	10 3/16	...	35⅛	....	36⅝	....	40⅝	....	42⅝			
10 × 11	11	...	36⅝	....	38⅛	....	42⅝	....	44⅛			

[1] Minimum wall thicknesses are same as for corresponding ASA B16e flange ratings. Where the minimum throat and plug bore is greater than the nominal valve size, minimum wall thicknesses are determined by interpolating. Flange and groove dimensions and ring numbers shall be in accordance with API Standard 5G-3 for corresponding ASA B16e flange rating, except for the 1½, 2, and 2½ sizes of the 3,000-lb valves which are API flanges for oil-field use only.

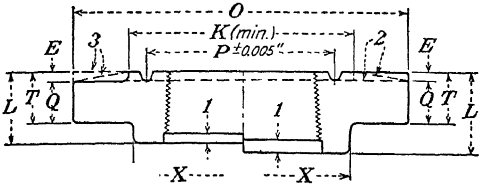
specifically recommending those shown for the full range of service conditions. Although the sketches indicate lubricated designs, nonlubricated valves may be of similar basic type except for the omission of the lubrication groove and substitution of mechanical means of freeing the plug.

API Tentative Standard on
RING JOINTS FOR STEEL FLANGES AND FLANGE UNIONS

Serial Designation 5G-3, 1940

Abstracted[1]

This standard covers API ring-joint dimensions for ASA B16e steel flanges for piping connections and end flanges on valves and fittings. This standard also covers ring-joint flange unions for use with line pipe, casing, drill pipe, and tubing.



(*a*) (*b*)

Fig. 5.—Flange and groove dimensions for ring joints: (*a*) flange for general or oil-field use when threaded with line-pipe thread; (*b*) flange for oil-field use except when threaded with line-pipe thread (see Table XI).

Note: Q, minimum flange thickness; see ASA B16e.

1. For depth and diameter of counterbore, also threading, thread tolerance and thread-gaging practice, see API Standard 5-F.
2. Flanges will be furnished with full face as shown by solid line in sketch unless otherwise specified by the purchaser. Bevel or ASA B16e face will be furnished only when specified by purchaser.
3. Flanges will be furnished with ASA B16e face as shown by solid line in sketch unless otherwise specified by purchaser. Bevel or full face will be furnished only when specified by purchaser.

General Requirements and Materials.—Material for carbon-steel flanges and for bolting shall be as specified in ASA B16e (see page 626). Material for alloy-steel flanges shall conform to Grades C1, C11, or C5 of ASTM A157 or to Grades F1, F4, or F5 of ASTM A182 as specified. See abstracts of ASTM Specifications in Chap. IV.

Allowable pressure ratings for cast or forged carbon-steel and carbon-molybdenum-steel ring joints for general use are identical with ratings given in ASA B16e (see pages 629, 631). Ratings for cast or forged 5 per cent chromium-molybdenum ring joints are given in Table X. Ring-joint flange unions for oil-field use may be used for cold-working pressures up to, but not exceeding, the coldworking-pressure ratings given in the general use tables.

[1] Abstract contains revisions of *Supplement* 1, January, 1941. See text and footnote 1, p. 1107.

Bolting material, dimensions, and threading are covered by ASA B16e requirements.

Dimensions.—Ring-joint flanges for general use and oil field use and flange unions for use with line pipe, casing, drill pipe, and tubing are based on ASA B16e dimensions (see Figs. 7 and 8). Except for a few small sizes in which the groove cuts $\frac{1}{32}$ in. into the minimum flange thickness, additional material is added to the face of the flanges equal to the depth of the groove.

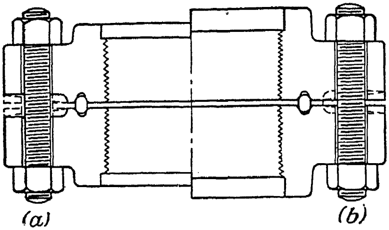


FIG. 6.—API standard ring-joint flange union: (*a*) flange for general or oil-field use when threaded with line-pipe thread; (*b*) flange for oil-field use when threaded with other than line-pipe thread and also for the $1\frac{1}{2}$- to $2\frac{1}{2}$-in. sizes of the 3000-lb cold-working pressure class when threaded for line-pipe thread.

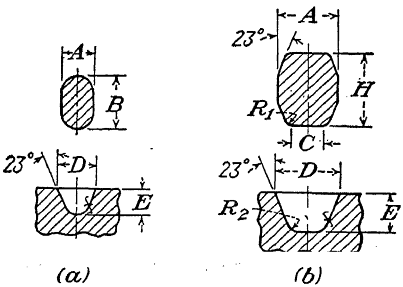


FIG. 7.—Ring and groove dimensions: (*a*) detail of ring-and-groove sections, oval-ring type; (*b*) octagonal-ring type.

Tolerances on dimensions of ring and groove given in Fig. 7 and Table XI are as follows:

B and *H*, depth of ring	$\pm\frac{1}{64}$ in.
A, width of ring	± 0.008 in.
C, width of flat on octagonal ring	± 0.008 in.
D, width of groove	± 0.008 in.
E, depth of groove	± 0.0156 in. -0.0000 in.
P, average pitch diameter of ring	± 0.007 in.
P, average pitch diameter of groove	± 0.005 in.
23 deg, angle	$\pm\frac{1}{2}$ deg

Dimensions and numbers of API rings and the sizes of the flanges of the various standards for which they are suitable are given in Table XI. Two cross-sectional shapes, the oval and the octagonal, are provided as illustrated in Fig. 7. (Octagonal rings should not be ordered for use with grooves having oval bottoms.)

Threading.—Threading of flanges for general use shall be in accordance with ASA Standard B2.1 or API line pipe threads of API Standard 5-F. Line pipe threads and American Standard threads are interchangeable except for the 2-in. size. API Standard 5-F gives length of thread required, dimensions of counterbore, etc., for use with line pipe threads (see page 1115). Threading of ring-joint flanges for oil-field use and of all ring-joint flange unions shall be in accordance with API Standard 5-F for line-pipe, casing, tubing, or drill-pipe threads as specified.

Transmission Lines.—The cross-country oil pipe line was first introduced about the year 1870. With the exception of tankers,

TABLE X.—PRESSURE-TEMPERATURE RATINGS FOR AMERICAN STANDARD CAST AND FORGED 5 PER CENT CHROMIUM, MOLYBDENUM[1] ALLOY STEEL VALVES, FITTINGS, AND FLANGES WHEN USED WITH RING JOINTS
(Table 3 of API Standard 5G-3)

Temperature, degrees F	Primary service rating, psi					
	300	400	600	900	1,500	2,500
100	720	960	1,400	2,160	3,600	6,000
150	700	925	1,400	2,100	3,500	5,825
200	680	900	1,360	2,040	3,400	5,660
250	660	875	1,320	1,980	3,300	5,500
300	640	850	1,280	1,920	3,200	5,330
350	620	825	1,240	1,860	3,100	5,165
400	600	800	1,200	1,800	3,000	5,000
450	575	775	1,150	1,725	2,875	4,800
500	550	725	1,100	1,650	2,750	4,575
550	525	700	1,050	1,575	2,625	4,375
600	500	675	1,000	1,500	2,500	4,175
650	475	625	950	1,425	2,375	3,950
700	450	600	900	1,350	2,250	3,750
750	425	575	850	1,275	2,125	3,500
800	400	525	800	1,200	2,000	3,325
850	375	500	750	1,125	1,875	3,125
900	350	475	700	1,050	1,750	2,925
950	325	425	650	975	1,625	2,700
1000	300	400	600	900	1,500	2,500
1050	225	275	425	650	1,075	1,775
1100	150	200	275	425	700	1,175

it has been found to be the most economical means of transporting crude oil from the fields to the refineries. With the advent of the

TABLE XI.—RING-JOINT NUMBERS AND BASIC DIMENSIONS (SEE FIG. 7)
(Table 4 of API Standard 5G-3)

Ring number	Pitch diameter of ring and groove	Width of ring	Height of ring		Width of flat on octagonal rings	Size of flange (inches)						
						General use					Oil-field use	
			Oval	Octagonal		150 lb	300 to 600 lb	900 lb	1,500 lb	2,500 lb	500, 1,000, 2,000, 3,000 lb	5,000 lb
	P	A	B	H	C							
R 11	1 11/32	1/4	7/16	3/8	0.170	....	1/2					
R 12	1 9/16	5/16	9/16	1/2	0.206	....	....	....	1/2			
R 13	1 11/16	5/16	9/16	1/2	0.206	....	3/4	....	....	1/2		
R 14	1 3/4	5/16	9/16	1/2	0.206	....	....	....	3/4			
R 15	1 7/8	5/16	9/16	1/2	0.206	1						
R 16	2	5/16	9/16	1/2	0.206	....	1	....	1	3/4	1	1
R 17	2 1/4	5/16	9/16	1/2	0.206	1 1/4						
R 18	2 3/8	5/16	9/16	1/2	0.206	....	1 1/4	....	1 1/4	1	1 1/4	1 1/4
R 19	2 9/16	5/16	9/16	1/2	0.206	1 1/2						
R 20	2 11/16	5/16	9/16	1/2	0.206	....	1 1/2	....	1 1/2	....	1 1/2	1 1/2
R 21	2 27/32	7/16	11/16	5/8	0.305	....	....	....	....	1 1/4		
R 22	3 1/4	5/16	9/16	1/2	0.206	2						
R 23	3 1/4	7/16	11/16	5/8	0.305	....	2	....	....	1 1/2	2	
R 24	3 3/4	7/16	11/16	5/8	0.305	....	....	....	2	....	....	2
R 25	4	5/16	9/16	1/2	0.206	2 1/2						
R 26	4	7/16	11/16	5/8	0.305	....	2 1/2	....	....	2	2 1/2	
R 27	4 1/4	7/16	11/16	5/8	0.305	....	....	....	2 1/2	....	....	2 1/2
R 28	4 3/8	1/2	3/4	11/16	0.341	....	....	....	....	2 1/2		
R 29	4 1/2	5/16	9/16	1/2	0.206	3						
R 30	4 5/8	7/16	11/16	5/8	0.305	....	3					

R 31	4⅞	7/16	11/16	⅝	0.305	....	3	3	....	....	3	
R 32	5	½	¾	11/16	0.341	....	....	....	....	3		
R 33	5 3/16	5/16	9/16	½	0.206	3½						
R 34	5 3/16	7/16	11/16	⅝	0.305	....	3½	3½	....	....	3½	
R 35	5⅜	7/16	11/16	⅝	0.305	....	....	....	3	....	....	3
R 36	5⅞	5/16	9/16	½	0.206	4						3½
R 37	5⅞	7/16	11/16	⅝	0.305	....	4	4	....	....	4	
R 38	6 3/16	⅝	⅞	13/16	0.413	....	....	....	....	4		
R 39	6⅜	7/16	11/16	⅝	0.305	....	....	....	4	....	....	4
R 40	6¾	5/16	9/16	½	0.206	5						
R 41	7⅛	7/16	11/16	⅝	0.305	....	5	5	....	....	5	
R 42	7½	¾	1	15/16	0.485	....	....	....	....	5		
R 43	7⅝	5/16	9/16	½	0.206	6						
R 44	7⅝	7/16	11/16	⅝	0.305	....	....	....	5	....	....	5
R 45	8 5/16	7/16	11/16	⅝	0.305	....	6	6	....	....	6	
R 46	8 5/16	½	¾	11/16	0.341	....	....	....	6	....	....	6
R 47	9	¾	1	15/16	0.485	....	....	....	....	6		
R 48	9¾	5/16	9/16	½	0.206	8						
R 49	10⅝	7/16	11/16	⅝	0.305	....	8	8	....	....	8	
R 50	10⅝	⅝	⅞	13/16	0.413	....	....	....	8	....	....	8
R 51	11	⅞	1⅛	1 1/16	0.583	....	....	....	....	8		
R 52	12	5/16	9/16	½	0.206	10						
R 53	12¾	7/16	11/16	⅝	0.305	....	10	10	....	....	10	
R 54	12¾	⅝	⅞	13/16	0.413	....	....	....	10	....	....	10
R 55	13½	1⅛	1 7/16	1⅜	0.780	....	....	....	....	10		
R 56	15	5/16	9/16	½	0.206	12						
R 57	15	7/16	11/16	⅝	0.305	....	12	12	....	....	12	
R 58	15	⅞	1⅛	1 1/16	0.583	....	....	....	12			
R 59	15⅝	5/16	9/16	½	0.206	14 O.D.						
R 60	16	1¼	1 9/16	1½	0.879	....	....	....	....	12 O.D.		

TABLE XI.—(*Concluded*)

Ring number	Pitch diameter of ring and groove	Width of ring	Height of ring		Width of flat on octagonal rings	Size of flange (inches)						
						General use					Oil-field use	
			Oval	Octagonal		150 lb	300 to 600 lb	900 lb	1,500 lb	2,500 lb	500,1,000, 2,000, 3,000 lb	5,000 lb
	P	*A*	*B*	*H*	*C*							
R 61	16½	7⁄16	11⁄16	⅝	0.305	....	14 O.D.	....	....	....	14	
R 62	16½	⅝	⅞	13⁄16	0.413	....	....	14 O.D.				
R 63	16½	1	1⁵⁄16	1¼	0.681	....	....	....	14 O.D.			
R 64	17⅞	5⁄16	9⁄16	½	0.206	16 O.D.						
R 65	18½	7⁄16	11⁄16	⅝	0.305	....	16 O.D.	....	....	....	16	
R 66	18½	⅝	⅞	13⁄16	0.413	....	....	16 O.D.				
R 67	18½	1⅛	1⁷⁄16	1⅜	0.780	....	....	....	16 O.D.			
R 68	20⅜	5⁄16	9⁄16	½	0.206	18 O.D.						
R 69	21	7⁄16	11⁄16	⅝	0.305	....	18 O.D.	....	....	....	18	
R 70	21	¾	1	15⁄16	0.485	....	....	18 O.D.				
R 71	21	1⅛	1⁷⁄16	1⅜	0.780	....	....	....	18 O.D.			
R 72	22	5⁄16	9⁄16	½	0.206	20 O.D.						
R 73	23	½	¾	11⁄16	0.341	....	20 O.D.	....	....	....	20	
R 74	23	¾	1	15⁄16	0.485	....	....	20 O.D.				
R 75	23	1¼	1⁹⁄16	1½	0.879	....	....	....	20 O.D.			
R 76	26½	5⁄16	9⁄16	½	0.206	24 O.D.						
R 77	27¼	⅝	⅞	13⁄16	0.413	....	24 O.D.					
R 78	27¼	1	1⁵⁄16	1¼	0.681	....	....	24 O.D.				
R 79	27¼	1⅜	1¾	1⅝	0.977	....	....	....	24 O.D.			

Second World War, ocean shipping difficulties made it necessary to revert to railroad tank-car transportation as an emergency measure and forced the construction of a 24-in.-diameter pipe line termed the "big inch" from the Texas oil fields to Phoenixville, Pa. This line is designed to deliver about 300,000 bbl per day at a flow velocity of 6.6 ft per sec. The extent of oil transmission lines can be appreciated by reference to a map of crude oil and products pipe lines such as was published in the *Oil Gas J.* for Sept. 24, 1942.

Pipe lines for conveying oil are laid on the surface of the ground or at a depth varying from 18 in. to 3 ft beneath the surface. Main lines generally are 8 in. in diameter and operate at pressures of 500 to 1,000 psi, with a velocity of about 3 ft per sec, and deliver approximately 30,000 bbl each in 24 hr. Practically all the pipe-line companies engaged in the transportation of petroleum have, in addition to their trunk lines, extensive systems of gathering lines of smaller size than the main trunk line, most of them being 4 to 6 in. in diameter. Oil is forced through the pipe lines by means of pumps located at stations from 2 to 100 miles apart, the distance varying with the nature of the country and the viscosity of the oil being pumped. The line pipe used for transmission lines and to a large extent around refineries and in the oil fields usually is bought to conform to API Specification 5L abstracted in the next section.

Line Pipe.—Line pipe for both threaded and welded construction is covered by API Specification 5L. A wide range of wall thicknesses and grades of welded and seamless pipe is provided. Line-pipe threads are of the same form and taper as American Standard Taper Pipe Threads, ASA B2.1. The effective lengths of thread given in Table XVI now correspond exactly with those given in ASA B2.1, with the exception of the 2-in. nominal size. The American Standard taper pipe thread length for the 2-in. size is included as an alternate in this table. Line-pipe couplings are all taper tapped, are longer than standard pipe couplings, and have recesses to cover all exposed threads. American Standard pipe couplings in sizes 2 in. and smaller are tapped with straight threads rather than with taper threads. Hence, if a taper-threaded coupling is desired, the line-pipe coupling should be specified. Plain- and beveled-end pipe is designated by the external diameter rather than by nominal pipe size, although the actual outside diameters correspond with dimensions of standard pipe. The angle of bevel on line pipe is 30 deg. The following abstract covers the principal requirements of API Specification 5L.

API Specification for LINE PIPE

API Standard 5L-1944

Abstracted[1]

This specification covers tubular goods for line-pipe purposes, commonly used to convey gas, water, or oil, and includes threaded pipe, "outside-diameter"

TABLE XII.—PROCESS OF MANUFACTURE
(Table 1 of API 5L)

F* = furnace butt welded.		
L* = furnace lap welded.		
Steel:		
Bessemer		†
Electric furnace		†
Open-hearth	Class I	†
	Class II‡	†
Iron:		
Open-hearth		†
Wrought		†
S* = seamless.		
Steel:		
Bessemer†		B, C
Electric furnace		A, B, C
Open-hearth		A, B, C
Iron:		
Open-hearth		†
E* = electric welded.		
Steel:		
Bessemer†		B, C
Electric furnace§		A, B, C
Open-hearth		A, B, C
Iron:		
Open-hearth§		‡

* Symbols indicating process of manufacturer:
F = furnace butt-welded.
L = furnace lap-welded.
E = electric-welded.
S = seamless.

† No grade designation.

‡ Rephosphorized.

§ *War emergency measure.* The use of open-hearth iron and electric-furnace steel in the manufacture of electric-welded pipe, and the use of Bessemer steel in the manufacture of electric-welded and seamless pipe is a war emergency measure, for the duration only. Bessemer steel for this use is defined as "Killed Deoxidized Acid Bessemer Steel," to be marked with the letter "B." These provisions are effective only for the duration of the war. Adopted November, 1942 and November, 1943.

[1] See text and footnote 1, p. 1107.

plain-end pipe in regular and light weights and special diameters, and extra-strong plain-end pipe. (The term "plain-end" as used herein means "not threaded.")

Manufacture.—Pipe shall be seamless, furnace butt- or lap-welded, or electric-welded of the materials indicated in Table XII.

Chemical Properties.—The steel shall conform to the following requirements as to chemical composition:

TABLE XIII.—CHEMICAL PROPERTIES
(Table 2 of API 5L)

Steel*	Manganese, per cent	Phosphorus: Not over, per cent	Phosphorus: Not less than, per cent	Sulphur not over, per cent	Carbon not over, per cent
Furnace butt and lap welded:					
Bessemer	.30– .60	.11	....	.065	
Electric furnace	.30– .60	.045	....	.06	
Open-hearth					
Class I	.30– .60	.045	....	.06	
Class II†	.30– .60	.08	.045	.06	
mless:					
Bessemer,‡					
Grade *B*	35–1.50	.11	....	.06	.30
Grade *C*	35–1.50	.11	....	.06	§
Electric furnace,					
Grade *A*	30– .90	.045	....	.06	
Grade *B*	35–1.50	.045	....	.06	.30
Grade *C*	35–1.50	.045	....	.06	§
Open-hearth,					
Grade *A*	30– .90	.045	....	.06	
Grade *B*	35–1.50	.045	....	.06	.30
Grade *C*	35–1.50	.045	....	.06	§
Electric welded:					
Bessemer,‡					
Grade *B*	35–1.50	.11	....	.06	.30
Grade *C*	35–1.50	.11	....	.06	§
Electric furnace,‡					
Grade *A*	30– .90	.045	....	.06	
Grade *B*	35–1.50	.045	....	.06	.30
Grade *C*	35–1.50	.045	....	.06	§
Open-hearth,					
Grade *A*	30– .90	.045	....	.06	
Grade *B*	35–1.50	.045	....	.06	.30
Grade *C*	35–1.50	.045	....	.06	§

* Open-hearth iron shall not contain more than 0.16 per cent total of carbon, sulphur, phosphorus, silicon, copper, and manganese.

† Rephosphorized. To be marked with letter "R."

‡ War emergency measure, for the duration only.

§ In case Grade "C" pipe is to be joined by welding, the purchaser may wish to stipulate the carbon content by special agreement.

Tensile Properties.—Material shall conform to the following tensile properties:

Table XIV.—Minimum Physical Properties (Table 3 of API 5L)

	Yield strength, psi	Tensile strength, psi	Elongation, per cent 8 in.	8 in.
Furnace butt and lap welded:				
Steel,				
Bessemer	30,000	50,000		
Electric furnace	25,000	45,000		
Open-hearth				
Class I	25,000	45,000		
Class II†	28,000	48,000		
Iron,				
Open-hearth	24,000	42,000		
Wrought	24,000	42,000		
Steel,				
Bessemer‡				
Grade *B*	35,000	60,000		
Grade *C*	45,000	75,000		
Electric furnace,				
Grade *A*	30,000	48,000		
Grade *B*	35,000	60,000		
Grade *C*	45,000	75,000		
Open-hearth,				
Grade *A*	30,000	48,000		
Grade *B*	35,000	60,000		
Grade *C*	45,000	75,000		
Iron,				
Open-hearth	24,000	42,000	20	
Electric welded:				
Steel,				
Bessemer‡				
Grade *B*	35,000	60,000		
Grade *C*	45,000	75,000		
Electric furnace,‡				
Grade *A*	30,000	48,000		
Grade *B*	35,000	60,000		
Grade *C*	45,000	75,000		
Open-hearth,				
Grade *A*	30,000	48,000		
Grade *B*	35,000	60,000		
Grade *C*	45,000	75,000		
Iron,				
Open-hearth‡	24,000	42,000	20	

* Minimum elongation for butt-welded pipe in sizes ¾ inch and smaller are as follows:

	Per Cent
¾-in. and ½-in. size, elongation in 6 in.	18
⅜-in. and ¼-in. size, elongation in 4 in.	18
⅛-in. size, elongation in 2 in.	18

† Rephosphorized. To be marked with letter "R."

‡ War emergency measure, for the duration only.

§ For elongation of grades *A* and *B* seamless and electric-welded steel line pipe (p. 1135):

Hydrostatic Tests.—Each length of pipe shall be tested at the mill to hydrostatic pressures tabulated in Table 7, 11A, 11B, 12 and 13, of API 5L. These pressures are designed to develop a fiber stress of 60 per cent of the specified yield strength, except that test pressures for the three classes of lap-welded pipe are based on the higher yield strength of the Bessemer pipe.

The hydrostatic mill test pressures are inspection pressure tests and do not necessarily have any direct relationship to working pressures. NOTE.—Mill test pressures, under changes adopted in November, 1942, are no longer required to be marked on line pipe.

Dimensions and Tolerances.—Lengths of threaded line pipe shall be not less than 18 ft for 95 per cent of shipment with a minimum of 16 ft for any length. Regular weight plain-end line pipe and lightweight pipe 20 in. O.D. and under shall have a minimum average length of 17 ft 6 in. with a minimum length of 9 ft. When average length agreed upon is in excess of 20 ft, not over 10 per cent shall be less than 75 per cent of the agreed length and no pipe less than 40 per cent of that length.

Tolerances on outside diameter of pipe 1½-in. nominal size and less is tabulated O.D. + 1/64 in., −1/32 in. For pipe 2-in. nominal size and larger, tabulated O.D. ± 1 per cent. Tolerance on wall thickness of body of pipe is tabulated thickness less 12½ per cent. Chamfer on the outside ends of threaded pipe; +0 deg, −10 deg. Plain-end pipe with beveled ends; width of flat at end of pipe 1/16 in. ± 1/32 in. Angle of bevel for welding 30 deg.;[1] +5 deg, −0 deg.

Weights and dimensions for standard threaded line pipe are given in Table XV.

Plain-end pipe for Dresser or similar type couplings shall be reamed inside and outside to remove burrs or shall be beveled the same as for welding. All such pipe shall be sufficiently free from indentation, projection, or roll marks for a distance of 8 in. from the end to make a tight joint with the rubber gasket. The outside diameter of all plain-end pipe 10¾ in. O.D. and smaller for such couplings or for welding, for a distance of 8 in. from the end, shall not be more than 1/64 in.

Wall thickness (inches)	Minimum elongation in 2 in., per cent	
	Grade *A*	Grade *B*
5/16 and over	35.00	30.00
9/32	33.25	28.50
1/4	31.50	27.00
7/32	29.75	25.50
3/16	28.00	24.00
5/32	26.25	22.50
1/8	24.50	21.00
3/32	22.75	19.50
1/16	21.00	18.00

NOTE.—Where wall thickness of pipe lies between two values shown above, the elongation specified for the larger wall thickness shall apply.

[1] The API standard outside bevel for plain-end line pipe for welding is 30 deg. For butt-welding fittings (ASA B16.9) and flanges (ASA B16e), the chamfer is 37½ deg; it is recognized by industry that pipe with a 30-deg bevel can be welded readily to pipe or fittings having a 37½-deg bevel or chamfer.

smaller than the basic O.D. and shall permit the passing of a ring gage which has a bore 1/16 in. larger than the basic O.D. Sizes larger than 10¾ in. O.D. shall not be more than 1/32 in. smaller and the ring gage not more than 3/32 in. larger than the basic outside diameter.

Threads.—Threaded line pipe shall be in accordance with dimensions given in Fig. 8 and Table XVI. Measurement of these threads shall be as specified in

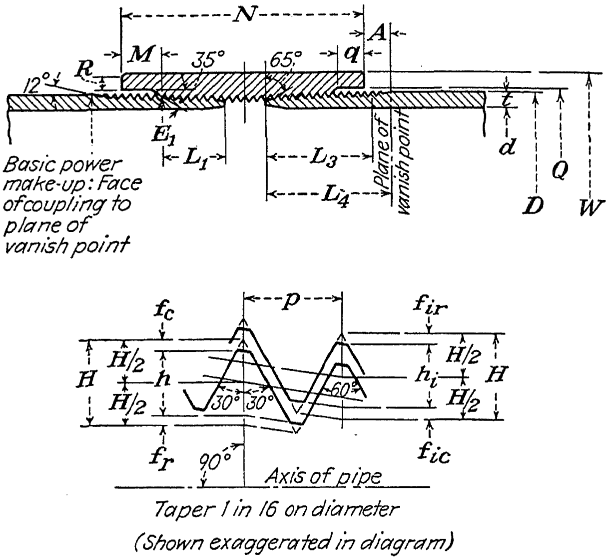


FIG. 8.—API standard 5L threaded line pipe, hand-tight assembly.

THREAD HEIGHT DIMENSIONS
(Inches)

Thread element	27 threads per in. $p = 0.0370$	18 threads per in. $p = 0.0556$	14 threads per in. $p = 0.0714$	11½ threads per in. $p = 0.0870$	8 threads per in. $p = 125$
$H = 0.866p$	0.0321	0.0481	0.0619	0.0753	0.1082
$h = h_i = 0.760p$	0.0281	0.0422	0.0543	0.0661	0.0950
$f_r = f_{ir} = 0.033p$	0.0012	0.0018	0.0024	0.0029	0.0041
$f_c = f_{ic} = 0.073p$	0.0027	0.0041	0.0052	0.0063	0.0091

API Standard 5B (see page 1114). NOTE.—Line-pipe threads and American Standard taper pipe threads, ASA Standard B2.1 are interchangeable except for the 2-in. size.

Threads in couplings and on pipe shall be so nearly alike in form and size and so well finished, that joints can be screwed together with suitable lubricant far enough to make a tight joint and unscrewed four times without injury to the threads. Adequate thread protectors must be provided.

TABLE XV —API STANDARD THREADED LINE PIPE—WEIGHTS AND DIMENSIONS (SEE FIG. 8)
(Table 8 of API 5L)
(All dimensions in inches at 68 F)

Size, nominal	Nominal weight, lb per ft	Outside diameter D	Inside diameter d	Wall thickness t
⅛	0.25	0.405	0.269	0.068
¼	0.43	0.540	0.364	0.088
⅜	0.57	0.675	0.493	0.091
½	0.86	0.840	0.622	0.109
¾	1.14	1.050	0.824	0.113
1	1.70	1.315	1.049	0.133
1¼	2.30	1.660	1.380	0.140
1½	2.75	1.900	1.610	0.145
2	3.75	2.375	2.067	0.154
2½	5.90	2.875	2.469	0.203
3	7.70	3.500	3.068	0.216
3½	9.25	4.000	3.548	0.226
4	11.00	4.500	4.026	0.237
5	15.00	5.563	5.047	0.258
6	19.45	6.625	6.065	0.280
8	25.55	8.625	8.071	0.277
8	29.35	8.625	7.981	0.322
10	32.75	10.750	10.192	0.279
10	35.75	10.750	10.136	0.307
10	41.85	10.750	10.020	0.365
12	45.45	12.750	12.090	0.330
12	51.15	12.750	12.000	0.375
14 *D*	57.00	14.000	13.250	0.375
15 *D*	61.15	15.000	14.250	0.375
16 *D*	65.30	16.000	15.250	0.375
17 *D*	73.20	17.000	16.214	0.393
18 *D*	81.20	18.000	17.182	0.409
20 *D*	90.00	20.000	19.182	0.409

"Size" of threaded line pipe indicates nominal size in the body portion and is not the outside diameter D except on sizes 14 to 20 in., inclusive.

TABLE XVI.—THREAD AND COUPLING DIMENSIONS API STANDARD THREADED LINE PIPE (SEE FIG. 8)
(Tables 9 and 10 API 5L) (All dimensions in inches at 68 F)

Size, nominal	No. of threads per inch	Length, end of pipe to hand-tight plane L_1	Effective length L_2	Total length, end of pipe to vanish point L_4	Pitch diameter at hand-tight plane E_1	End of pipe to center of coupling, made-up J	Outside diameter of coupling W	Length N	Diameter of recess Q	Depth of recess q	Length, face of coupling to hand-tight plane M	Width of bearing face R	Pitch diameter at hand-tight plane E_1	Hand-tight stand-off number threads A
1/8	27	0.180	0.2639	0.3974	0.37476	0.1389	0.563	1 1/16	0.468	0.0347	0.1014	1/32	0.37476	3
1/4	18	0.200	0.4018	0.5946	0.48989	0.2179	0.719	1 5/8	0.603	0.1471	0.2278	1/32	0.48989	3
3/8	18	0.240	0.4078	0.6006	0.62701	0.2119	0.875	1 5/8	0.738	0.1147	0.1938	1/32	0.62701	3
1/2	14	0.320	0.5337	0.7815	0.77843	0.2810	1.063	2 1/8	0.903	0.1582	0.2473	1/16	0.77843	3
3/4	14	0.339	0.5457	0.7935	0.98887	0.2690	1.313	2 1/8	1.113	0.1516	0.2403	1/16	0.98887	3
1	11 1/2	0.400	0.6828	0.9845	1.23863	0.3280	1.576	2 5/8	1.378	0.2241	0.3235	3/32	1.23863	3
1 1/4	11 1/2	0.420	0.7068	1.0085	1.58338	0.3665	2.054	2 3/4	1.723	0.2279	0.3275	3/32	1.58338	3
1 1/2	11 1/2	0.420	0.7235	1.0252	1.82234	0.3498	2.200	2 3/4	1.963	0.2439	0.3442	3/32	1.82234	3
*2	11 1/2	0.436	0.7565	1.0582	2.29627	0.5668	2.875	3 1/4	2.469	0.2379	0.3611	1/8	2.29627	3
2	11 1/2	0.668	0.9884	1.2901	2.29627	0.3349	2.875	3 1/4	2.469	0.2379	0.3611	1/8	2.29627	3
2 1/2	8	0.682	1.1375	1.5712	2.76216	0.4913	3.375	4 1/8	2.969	0.4915	0.6392	3/16	2.76216	2
3	8	0.766	1.2000	1.6337	3.38850	0.4913	4.000	4 1/4	3.594	0.4710	0.6177	3/16	3.38850	2
3 1/2	8	0.821	1.2500	1.6837	3.88881	0.5038	4.625	4 3/8	4.094	0.4662	0.6127	3/16	3.88881	2
4	8	0.844	1.3000	1.7337	4.38713	0.5163	5.200	4 1/2	4.594	0.4920	0.6397	1/4	4.38713	2
5	8	0.937	1.4063	1.8400	5.44929	0.4725	6.296	4 5/8	5.657	0.5047	0.6530	1/4	5.44929	2
6	8	0.958	1.5125	1.9462	6.50597	0.4913	7.390	4 7/8	6.719	0.5861	0.7382	1/4	6.50597	2
8	8	1.063	1.7125	2.1462	8.50003	0.4788	9.625	5 1/4	8.719	0.6768	0.8332	1/4	8.50003	2
10	8	1.210	1.9250	2.3587	10.62094	0.5163	11.750	5 3/4	10.844	0.7394	0.8987	3/8	10.62094	2
12	8	1.360	2.1250	2.5587	12.61781	0.5038	14.000	6 1/8	12.844	0.7872	0.9487	3/8	12.61781	2
14*D*	8	1.562	2.2500	2.6837	13.87263	0.5038	15.000	6 3/8	14.094	0.7136	0.8717	3/8	13.87263	2
15 *D*	8	1.687	2.3500	2.7837	14.87419	0.5288	16.000	6 5/8	15.094	0.6897	0.8467	3/8	14.87419	2
16 *D*	8	1.812	2.4500	2.8837	15.87575	0.4913	17.000	6 3/4	16.094	0.6658	0.8217	3/8	15.87575	2
17 *D*	8	1.900	2.5500	2.9837	16.87500	0.5163	18.000	7	17.094	0.6773	0.8337	3/8	16.87500	2
18 *D*	8	2.000	2.6500	3.0837	17.87500	0.4788	19.000	7 1/8	18.094	0.6773	0.8337	3/8	17.87500	2
20 *D*	8	2.125	2.8500	3.2837	19.87031	0.5288	21.000	7 5/8	20.094	0.7490	0.9087	3/8	19.87031	2

NOTES.—Included taper, all sizes, is 0.0625 in. per in.
* American Standard taper pipe thread length, ASA B2.1
"Size" of threaded line pipe indicates nominal size in the body portion and is not the outside diameter *D*, except on 14 to 20-in. sizes, inclusive.

TABLE XVII-*A*.—API STANDARD PLAIN END LINE PIPE STANDARD WEIGHT, NOMINAL SIZE PIPE (WEIGHTS AND DIMENSIONS)
(Table 11-A of API 5L)
(All dimensions in inches at 68 F)

1	2	3	4	5
Size nominal	Outside diameter *D*	Inside diameter *d*	Wall thickness *t*	Weight per ft (lb) plain end
1/8	.405	.269	.068	.24
1/4	.540	.364	.088	.42
3/8	.675	.493	.091	.57
1/2	.840	.622	.109	.85
3/4	1.050	.824	.113	1.13
1	1.315	1.049	.133	1.68
1 1/4	1.660	1.380	.140	2.27
1 1/2	1.900	1.610	.145	2.72
2	2.375	2.067	.154	3.65
2 1/2	2.875	2.469	.203	5.79
3	3.500	3.068	.216	7.58

TABLE XVII-*B*.—API STANDARD PLAIN-END LINE PIPE, REGULAR WEIGHT, "OUTSIDE DIAMETER" PIPE (WEIGHTS AND DIMENSIONS)

(Table 11-B of API 5L)

(All dimensions in inches at 68 F)

Size, outside diameter D	Inside diameter d	Wall thickness t	Weight per ft (lb) plain end
3½	3.124	0.188	6.63
3½	3.068	0.216	7.58
3½	3.000	0.250	8.68
3½	2.938	0.281	9.67
3½	2.900	0.300	10.25
4½	4.124	0.188	8.64
4½	4.062	0.219	10.00
4½	4.026	0.237	10.79
4½	4.000	0.250	11.35
4½	3.938	0.281	12.67
4½	3.876	0.312	13.98
4½	3.826	0.337	14.98
6⅝	6.249	0.188	12.89
6⅝	6.187	0.219	14.97
6⅝	6.125	0.250	17.02
6⅝	6.065	0.280	18.97
6⅝	6.001	0.312	21.07
6⅝	5.937	0.344	23.06
6⅝	5.875	0.375	25.03
6⅝	5.761	0.432	28.57
8⅝	8.249	0.188	16.90
8⅝	8.187	0.219	19.64
8⅝	8.125	0.250	22.36
8⅝	8.071	0.277	24.70
8⅝	8.001	0.312	27.74
8⅝	7.981	0.322	28.55
8⅝	7.937	0.344	30.40
8⅝	7.875	0.375	33.04
8⅝	7.749	0.438	38.26
8⅝	7.625	0.500	43.39
10¾	10.312	0.219	24.60
10¾	10.250	0.250	28.04
10¾	10.192	0.279	31.20
10¾	10.136	0.307	34.24
10¾	10.062	0.344	38.20
10¾	10.020	0.365	40.48
10¾	9.874	0.438	48.19
10¾	9.750	0.500	54.74
12¾	12.250	0.250	33.38
12¾	12.188	0.281	37.45
12¾	12.126	0.312	41.51
12¾	12.090	0.330	43.77
12¾	12.062	0.344	45.55
12¾	12.000	0.375	49.56
12¾	11.874	0.438	57.53
12¾	11.750	0.500	65.42
14	13.376	0.312	45.68
14	13.312	0.344	50.14
14	13.250	0.375	54.57
14	13.124	0.438	63.37
14	13.000	0.500	72.09
16	15.376	0.312	52.36
16	15.312	0.344	57.48
16	15.250	0.375	62.58
16	15.124	0.438	72.72
16	15.000	0.500	82.77
18	17.376	0.312	59.03
18	17.312	0.344	64.82
18	17.250	0.375	70.59
18	17.124	0.438	82.06
18	17.000	0.500	93.45
20	19.376	0.312	65.71
20	19.312	0.344	72.16
20	19.250	0.375	78.60
20	19.124	0.438	91.41
20	19.000	0.500	104.13

"Size" of plain-end line pipe, regular and light weights, is outside diameter *D*.

NOTE. Light weight and special-diameter "outside diameter" plain-end line pipe may be obtained on special order, see Table 12 of API 5L. Mill test pressures of Table 11-B of API are not reproduced here as they are no longer used as a basis for design.

TABLE XVIII.—API STANDARD PLAIN-END LINE PIPE
EXTRA-STRONG (WEIGHTS AND DIMENSIONS)
(Table 13 of API 5L)
(All dimensions in inches at 68 F)

Size, nominal	Outside diameter D	Inside diameter d	Wall thickness t	Weight per ft (lb) plain end
1/8	0.405	0.215	0.095	0.31
1/4	0.540	0.302	0.119	0.54
3/8	0.675	0.423	0.126	0.74
1/2	0.840	0.546	0.147	1.09
3/4	1.050	0.742	0.154	1.47
1	1.315	0.957	0.179	2.17
1 1/4	1.660	1.278	0.191	3.00
1 1/2	1.900	1.500	0.200	3.63
2	2.375	1.939	0.218	5.02
2 1/2	2.875	2.323	0.276	7.66
3	3.500	2.900	0.300	10.25
3 1/2	4.000	3.364	0.318	12.51
4	4.500	3.826	0.337	14.98
5	5.563	4.813	0.375	20.78
6	6.625	5.761	0.432	28.57
8	8.625	7.625	0.500	43.39
10	10.750	9.750	0.500	54.74
12	12.750	11.750	0.500	65.42

"Size" of extra-strong plain-end line pipe indicates nominal size in the body portion and is not the outside diameter D.

NOTE.—Mill test pressures of Table 13 of API 5L are not reproduced here.

API Standard on FLANGED STEEL OUTSIDE-SCREW-AND-YOKE WEDGE-GATE VALVES

Serial Designation 600A-1942

Abstracted[1]

This standard covers the major dimensions for flanged rising-stem, outside-screw-and-yoke, solid wedge-gate valves for refinery use and for other services such as steam generating plants, long-distance pipe lines, producing operations, and natural-gas plants.

Face-to-face dimensions given in Table XIX are identical with ASA B16.10 for ferrous flanged-end wedge-gate valves (see page 557).

Classification.—Valves are identified by the 150- to 1,500-lb pressure designations given in ASA B16e (see page 626) for carbon steel and carbon molybdenum and equivalent alloys. Cast and forged 4 to 6 per cent chrome and cast 8 to 10 per cent chrome alloys are rated at 950 F for joints other than ring type and at

[1] See text and footnote 1, p. 1107.

1000 F for the ring joint with suitable adjustment of working pressures for temperature above and below the primary service pressure rating temperature.

Materials.—Materials for cast and forged carbon and alloy steel valve bodies conform to ASTM Specifications A95, A105, A157 and A182 (see Chap. IV).

Body and Bonnet.—Valve body thickness shall not be less than those shown in Table XIX. Bodies and bonnets are required to be of circular type, except the 150- and 300-lb classes which may be oval.

Line Flanges.—Facings on line flanges are required to conform to standard facings of ASA B16e, API 5G3, or ball-type joint as approved by purchaser.

Bonnet Flanges.—The joint between body and bonnet is required to be of the male-and-female, tongue-and-groove, ring-joint, or ball-joint type, except for 150-lb valves which may have plain faces.

Gasket.—Bonnet gaskets are required to be aluminum, soft steel, or iron unless otherwise specified, except for 150-lb valves which may be furnished with asbestos gaskets.

Bonnet Bolting.—Bolt studs are required to be equal to SAE 4140 analysis heat-treated to Class C physicals of ASTM A96 (see page 537).

Wedge Guides.—Integral guides are to be provided to ensure that the seating surfaces of the wedge will not touch the seat rings until point of closure is reached. On cast-steel valves, 400 lb and heavier, guides are to be machined.

Glands and Gland Bolting.—Split glands are not permissible. Bushings of one-piece glands and followers of two-piece glands are required to be made of corrosion-resistant material with a melting point above 2000 F. Gland bolting is required to be made of steel equivalent to ASTM A107.

Yoke and Yoke Nut.—Yokes may be integral with bonnet or bolted to it. Yoke nuts are required to be made of nonrusting material having a melting point of 2000 F or higher. Antifriction, ball, or roller bearings are required on 400-lb valves, 10 in. and over, and on 600-, 900-, and 1,500-lb valves, 6 in. and over.

Back Seat Bushing.—Back seat bushings of corrosion-resistant material equal to Grade C5-A of ASTM A157 or F5 of A182 (see pages 682 and 531) are required on all valves, except the 150-lb class which may have the seat machined directly on the bonnet.

Stem, Wedge, and Seat Rings.—The minimum diameters of stems are given in Table XIX. Solid wedges only are covered in this standard. Facing rings may be attached to wedge by rolling or threading.[1]

Unless otherwise specified by purchaser, facing rings and seat rings are required to be 11.5 to 13.5 per cent chromium stainless steel with carbon limited to 0.12 per cent. A difference of 50 points Brinell between wedge facing and seat is required to prevent a tendency toward galling. If hard facing is applied to seat and disk faces, no difference in hardness is required.

Body and Seat Tests.—Body hydrostatic tests of three times the service pressure designation and seat tightness tests of two times are required for the 300- to 1,500-lb classes. The 150-lb class is considered as rated at 100 lb in establishing these test pressures. An air test at 100 psi gage is used as a further check on seat tightness.

Marking.—Valves are required to be marked in accordance with MSS Standard Practice SP-25 (see page 687). Where valves comply completely with requirements of this standard, marking shall include the API monogram.

[1] *Author's Note:* Welded-in seat rings should be considered as alternate construction.

TABLE XIX.—DIMENSIONS OF FLANGED STEEL WEDGE GATE VALVES
(Table 8 of API STD 600A)
(All dimensions in inches)

Nominal size	Class 150				Class 300				Class 400				Class 600				Class 900				Class 1,500			
	Body thickness,[1] minimum	Dimension A^2: Raised face	Dimension A^2: Ring and groove	Stem diameter, minimum	Body thickness,[1] minimum	Dimension A^2: Raised face	Dimension A^2: Ring and groove	Stem diameter, minimum	Body thickness,[1] minimum	Dimension A^2: Raised face	Dimension A^2: Ring and groove	Stem diameter, minimum	Body thickness,[1] minimum	Dimension A^2: Raised face	Dimension A^2: Ring and groove	Stem diameter, minimum	Body thickness[1] minimum	Dimension A^2: Raised face	Dimension A^2: Ring and groove	Stem diameter, minimum	Body thickness,[1] minimum	Dimension A^2: Raised face	Dimension A^2: Ring and groove	Stem diameter, minimum
1	.....	.....	....	...	...	....	.....	...	...	...	...	...	0.310	8½	8[illegible]/16	⅝	.....	..	....	...	0.500	10	9[illegible]/16	¾
1¼	.....	...	......	...	...	...	.....	...	...	...	...	...	0.340	9	8 15/16	⅝	.....	..	....	...	0.560	11	10[illegible]/16	¾
1½	.....	...	......	...	0.310	7½	7 15/16	¾	...	...	...	...	0.365	9½	9[illegible]/16	¾	.....	..	....	...	0.590	12	11[illegible]/16	⅞
2	0.344	7	7½	¾	0.375	8½	9⅛	¾	...	...	...	...	0.438	11½	11[illegible]/8	⅞	.....	..	....	...	0.750	14½	14[illegible]	1
2½	0.375	7½	8	¾	0.438	9½	10⅛	¾	...	...	...	...	0.469	13	13[illegible]/8	⅞	.....	..	....	...	0.875	16½	16[illegible]	1⅛
3	0.406	8	8½	⅞	0.469	11⅛	11¾	⅞	...	...	...	...	0.500	14	14[illegible]/8	1	0.750	15	15⅛	1⅛	0.938	18½	18[illegible]	1¼
4	0.438	9	9½	1	0.500	12	12⅝	1	0.500	16	16[illegible]	1⅛	0.625	17	17[illegible]/8	1¼	0.844	18	18⅛	1¼	1.125	21½	21[illegible]	1⅜
6	0.469	10½	11	1⅛	0.625	15⅞	16½	1¼	0.656	19½	19[illegible]	1⅜	0.750	22	22[illegible]/8	1½	1.031	24	24⅛	1⅝	1.500	27¾	28	1¾
8	0.500	11½	12	1¼	0.688	16½	17⅛	1⅜	0.750	23½	23[illegible]	1⅝	1.000	26	26[illegible]/8	1¾	1.250	29	29⅛	1⅞	1.875	32¾	33[illegible]	2⅛
10	0.563	13	13½	1⅜	0.750	18	18⅝	1½	0.844	26½	26[illegible]	1¾	1.125	31	31⅛	1⅞	1.438	33	33⅛	2⅛	2.250	39	39[illegible]	2½
12	0.625	14	14½	1½	0.813	19¾	20⅜	1⅝	0.938	30	30[illegible]	1⅞	1.250	33	33⅛	2	1.656	38	38⅛	2¼	2.265	44½	45[illegible]	2¾

[1] Minimum casting thickness at any point upon inspection.

[2] For ring-groove facing, dimension A is from outside face to outside face of flanges. For raised facing, it is from contact face to contact face. Permissible tolerance $\pm 1/16$ in.

Flange and facing proportions of raised-face flanges conform to ASA B16e.

Flange and facing proportions of ring-and-groove flanges conform to API Standard 5-G-3.

API Standard on FLANGED STEEL PLUG VALVES

Serial Designation 600B-1942

Abstracted[1]

This standard covers the major dimensions for flanged steel lubricated and nonlubricated plug valves for use in refineries, natural-gas and natural-gasoline plants, and for any other service for which this type of valve (see Fig. 9) may be considered suitable.

Four general design groups are covered, *viz:*

(*a*) *Short Pattern.*—Same face-to-face dimensions as flanged steel wedge-gate valves.

(*b*) *Regular Pattern.*—Plug ports are larger than in the short pattern. Although designed to approximate streamline flow, design does not approach shape of a Venturi throat.

(*c*) *Venturi Pattern.*—Conjunction of body and port area approximates a Venturi throat, giving minimum pressure loss consistent with the reduced port

(*d*) *Round Port Full-bore Pattern.*—Circular port through both plug and body equal to at least the inside diameter of the comparable ASA fitting. Face-to-face dimensions are necessarily greater than any of the other three types.

Dimensions.—Body thicknesses and face-to-face dimensions of valves of these four designs are given in Table XX.

Pressure Designations.—Valves are identified, by the 150-, 300-, 400-, 600-, 900-, and 1,500-lb pressure designations given in ASA B16e (see page 626) for carbon steel and carbon-molybdenum and equivalent alloys. Cast and forged 4 to 6 per cent chrome and cast 8 to 10 per cent chrome alloys are rated at 950 F for joints other than ring type and at 1000 F for the ring joint with suitable adjustment of working pressures for temperatures above and below the primary service pressure rating temperature.

Materials.—Materials for cast and forged carbon and alloy valve bodies conform to ASTM Specifications A95, A105, A157, and A182 (see Chap. IV).

Cover Bolting.—Bolt studs conform to the material and threading practice used in ASA B16e. Wrench fit studs and the threaded holes are required to be threaded to Class 5 fit as given in Screw Thread Standards for Federal Services, Handbook H-28.

Gland and Gland Bolting.—Split glands are prohibited. Bushings of one-piece glands and follows of two-piece glands are required to be made of corrosion-resistant material with a melting point 2000 F or higher.

Stops and Indicators.—Stops are required for both full-open and full-closed position. Indicators are required to show the open and closed position of the plug.

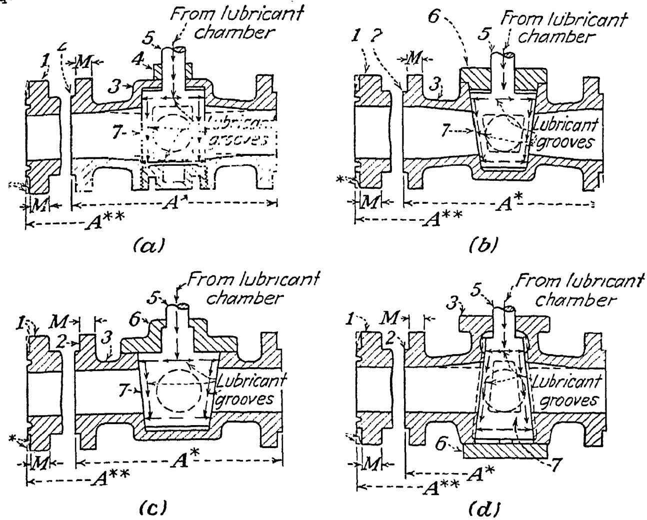
Locking Devices.—Provision for locking valve in either open or closed position is to be made at purchaser's option.

Body and Seat Tests.—Body hydrostatic tests of three times the service pressure designations and seat tightness hydrostatic tests of two times are required for the 300- to 1,500-lb classes. The 150-lb class is considered as rated at 100 lb

[1] See text and footnote 1, p. 1107.

in establishing these test pressures. An air test at 100 psi gage is used as a further check on seat tightness.

Marking.—Valves are required to be marked in accordance with MSS Standard Practice SP-25 (see page 687). Where valves comply completely with the requirements of this specification, marking shall include the API monogram.



PART NAME

1. Ring-Groove Facing.
2. Raised Facing.
3. Body.
4. Indicator and Stop.
5. Plug Stem.
6. Cover.
7. Plug.

FIG. 9.—Flanged steel-plug valves: (*a*) cylindrical-plug type; (*b*) round-port full-bore type; (*c*) tapered plug for short pattern, regular and Venturi designs; (*d*) inverted tapered-plug type for short pattern, regular and Venturi designs.

NOTE.—Figure 9 illustrates some of the basic valve types that are considered as being covered by API 600B. Reference to these types, however, is to be construed neither as prohibiting such other designs as may be offered nor as specifically recommending those shown for the full range of service conditions covered herein. Although the sketches indicate lubricated designs, nonlubricated valves may be of a similar basic type except for the omission of the lubrication groove and the substitution of mechanical means of freeing the plug. (Manufacturers supplying valves or parts of same do so at their own responsibility in respect to patent infringement.)

* For the raised facing, dimension "A" is from outside raised face to outside raised face.

** For ring-groove facing, dimension "A" is from outside face to outside face of flanges.

*** End flanges for ring joints may be obtained full face, beveled, or with ASA B16e face as required.

TABLE XX.—DIMENSIONS OF FLANGED STEEL PLUG VALVES
(Tables 9 to 14, inclusive, of API Standard 600 B)

Nominal size, inches	Class, pounds	Body thickness, min[a]	Face-to-face dimension[b]							
			Short pattern		Regular		Venturi		Round port full bore	
			Raised face[c]	Ring and groove[d]	Raised face[c]	Ring and groove[d]	Raised face[c]	Ring and groove[d]	Raised face[c]	Ring and groove[d]
1	150	.....	....	....	....	......	....	......	7	7 7/16
	300	.....	....	.....	....	......	....	......	7½	7 15/16
	400	(e)								
	600	0.310	....	.....	8½	8 7/16	8½	8 7/16	10	9 15/16
	900									
	1,500	0.500	....	.....	10	9 15/16	10	9 15/16	11¼	11 3/16
1¼	150									
	300									
	400	(e)								
	600	0.340	....	.....	9	8 15/16	9	8 15/16		
	900									
	1,500	0.560	....	....	11	10 15/16	11	10 15/16		
1½	150	.....	....	....	....	......	....	......	8¾	9 3/16
	300	0.310	....	....	7½	7 15/16	....	......	9½	9 15/16
	400	(e)								
	600	0.365	....	....	9½	9 7/16	9½	9 7/16	12½	12 7/16
	900	(e)								
	1,500	0.590	....	....	12	11 15/16	12	11 15/16	14	13 15/16
2	150	0.344	....	....	7	7½	....	......	10½	11
	300	0.375	....	....	8½	9⅛	....	......	11⅛	11¾
	400	(e)								
	600	0.438	....	....	11½	11⅝	11½	11⅝	13	13⅛
	900	(e)								
	1,500	0.750	....	....	14½	14⅝	14½	14⅝	15	15⅛
2½	150	0.375	....	....	7½	8	....	......	11¾	12¼
	300	0.438	....	....	9½	10⅛	....	......	13	13⅝
	400	(e)								
	600	0.469	....	....	13	13⅛	13	13⅛	15	15⅛
	900	(e)								
	1,500	0.875	....	....	16½	16⅝	16½	16⅝	17	17⅛
3	150	0.406	....	....	8	8½	....	......	13½	14
	300	0.469	....	....	11⅛	11¾	....	......	15¼	15⅞
	400	(e)								
	600	0.500	....	....	14	14⅛	14	14⅛	17½	17⅝
	900	0.750	....	....	15	15⅛	15	15⅛	18½	18⅝
	1,500	0.938	....	....	18½	18⅝	18½	18⅝	20½	20⅝
4	150	0.438	....	....	9	9½	....	......	17	17½
	300	0.500	....	....	12	12⅝	....	......	18	18⅝
	400	0.500	....	....	16	16⅛	16	16⅛	19	19⅛
	600	0.625	....	....	17	17⅛	17	17⅛	20	20⅛
	900	0.844	....	....	18	18⅛	18	18⅛	22	22⅛
	1,500	1.125	....	....	21½	21⅝	21½	21⅝	24	24⅛

TABLE XX.—(*Concluded*)

Nominal size, inches	Class, pounds	Body thickness, min[a]	Face-to-face dimension[b]							
			Short pattern		Regular		Venturi		Round port full bore	
			Raised face[c]	Ring and groove[d]	Raised face[c]	Ring and groove[d]	Raised face[c]	Ring and groove[d]	Raised face[c]	Ring and groove[d]
6	150	0.469	10½[f]	11	15½	16	15½	16	21	21½
	300	0.625	....	....	....	....	15⅞	16½	22	22⅝
	400	0.656	....	....	19½	19⅝	19½	19⅝	24	24⅛
	600	0.750	....	....	22	22⅛	22	22⅛	26	26⅛
	900	1.031	....	....	24	24⅛	24	24⅛	29	29⅛
	1,500	1.500	....	....	27¾	28	27⅜	28	31	31¼
8	150	0.500	11½[f]	12	18	18½	18	18½	25	25½
	300	0.688	....	....	....	....	16½	17⅛	27	27⅝
	400	0.750	....	....	23½	23⅝	23½	23⅝	29	29⅛
	600	1.000	....	....	26	26⅛	26	26⅛	31¼	31⅜
	900	1.250	....	....	29	29⅛	29	29⅛	32	32⅛
	1,500	1.875	....	....	32¾	33⅛	32¾	33⅛	35	35⅜
10	150	0.563	13[f]	13½	21	21½	21	21½	31	31½
	300	0.750	....	....	....	....	18	18⅝	32½	33⅛
	400	0.844	....	....	26½	26⅝	26½	26⅝	35	35⅛
	600	1.125	....	....	31	31⅛	31	31⅛	37	37⅛
	900	1.438	....	....	33	33⅛	33	33⅛	38	38⅛
	1,500	2.250	....	....	39	39⅜	39	39⅜	42	42⅜
12	150	0.625	14[f]	14½	24	24½	24	24½	36	36½
	300	0.813	....	....	....	....	19¾	20⅜	38	38⅝
	400	0.938	....	....	30	30⅛	30	30⅛	40	40⅛
	600	1.250	....	....	33	33⅛	33	33⅛	42	42⅛
	900	1.656	....	....	38	38⅛	38	38⅛	44	44⅛
	1,500	2.265	....	....	44½	45⅛	44½	45⅛	48	48⅝
14	150	0.688	....	....	27	27½	27	27½		
	300	0.875	....	....	....	....	30	30⅝		
	400	1.033	....	....	32½	32⅝	32½	32⅝		
16	150	0.750	....	....	30	30½	30	30½		
	300	1.000	....	....	....	....	33	33⅝		
	400	1.155	....	....	35½	35⅝	35½	35⅝		
18	150	0.875	....	....	34	34½	34	34½		
	300	1.125	....	....	....	....	36	36⅝		
	400	1.250	....	....	38½	38⅝	38½	38⅝		
20	150	1.000	....	....	36	36½	36	36½		
	300	1.250	....	....	....	....	39	39¾		
	400	1.357	....	....	41½	41¾	41½	41¾		
24	150	1.250	....	....	42	42½	42	42½		
	300	1.500	....	....	....	....	45	45⅞		
	400	1.625	....	....	48½	47⅞	48½	48⅞		

[a] Minimum casting thickness in throat between end flanges and body.
[b] Permissible tolerance $+\frac{1}{16}$ in.
[c] Flange and facing proportions and drilling templates shall conform to ASA B16e (see pp. 636–678).
[d] Flange and facing proportions and drilling templates shall conform to API Standard 5G-3, (see p. 1125).
[e] Valves in the next higher pressure class shall be furnished in this size.
[f] These sizes have same face-to-face dimensions as those given in ASA B16.10 for ferrous flanged-end wedge-gate valves (see p. 557).

American Standard Code for PRESSURE PIPING

Section 3

OIL PIPING SYSTEMS

ASA B31.1-1942

Abstracted[1]

Scope.—Section 3 of this Code covers mandatory requirements and recommended practices for the design, manufacture, test, and installation of oil piping systems.[2]

Classification.—The requirements apply particularly to oil piping, refinery gas piping, and piping for gasoline recovery plants. They may be applied also to piping other than oil, such as steam, air, water, gas, etc., used in the distribution system and process areas of the plants devoted principally to the processing and handling of oils, but they shall not be used for steam piping in such plants within the limits of the boiler-house or boiler-plant areas, nor for boiler-feed-water piping; piping for these services shall be in accordance with requirements of Section 1 on Power Piping. Piping for producing operations is not covered by the Code for Pressure Piping.

Oil lines and all vapor lines are classified according to pressure and temperature:

	Pressure, psi	Temperature, F
Class 1 for conditions not exceeding*..	125	350
Class 2 for conditions not exceeding*..	250	450
Class 3 for conditions above**.	300	500
Class 4 for oil at high pressure.	High pressure	100

* Refinery gas lines in receiving houses or within 50 ft of stills and other high temperature equipment shall follow specifications not lower than for Class 1.

** In general, pressures exceeding 300 psi and temperatures above 500 F are covered by Class 3 requirements but reference should be made to the pressure-temperature ratings noted below under "Standards and Specifications."

Standards and Specifications.—The pressure-temperature ratings for American Standard steel valves and fittings, ASA-B16e, which are applicable to oil piping are given in Chap. IV, pages 628 to 631). The pressure ratings and specifications for cast-iron pipe and fittings also are given in Chap. IV. For pressure ratings for valves see pages 1116 to 1127, inclusive.

Materials.—Materials shall be in accordance with the latest revisions of the respective specifications enumerated in the Code. Materials not covered in Tables XXIII to XXVII, inclusive, may be used, provided the allowable stress

[1] For full information or latest revisions, reference should be made to the Codes Copies of the complete code may be obtained from the American Standard. Association, 29 West 39th St., New York 18, N.Y.

[2] For an example of practices followed in refinery piping, see "Extensive Piping Systems Needed for the Production of Aviation Gasoline," by N. S. Bosworth, *Heating, Piping, and Air Conditioning*, January, 1945, pp. 9 to 15.

is set with reference to the relative strength at the operating temperature of the new material and the Code material most closely resembling it.

Cast-iron Pipe within Refinery Limits.—Cast-iron pipe may be used for oil, refinery gas, brine, foamite, etc., for underground service where the metal temperature of the pipe line is less than 300 F. No operating pressure limit is prescribed, except as limited by the specified test pressure and by the pipe wall thickness formula (16*e*) given on page 42. Cast-iron pipe may be used for oil or refinery gases aboveground for pressures not in excess of 150 psi where the metal temperature does not exceed 300 F. Cast-iron pipe may be used in oil condensers and coolers for service pressures up to 150 psi where the pipe is covered by water not above 212 F and where the metal temperature of the pipe will not exceed 450 F. (It is recommended that oil vapor temperatures limited to 525 F and oil temperatures to 650 F to ensure that the 450 F pipe wall temperature will not be exceeded.)

Cast-iron pipe, when used for gas or oil within refinery limits *above*ground shall have the following actual thicknesses when installed [see formula (16*e*), pages 42, and Table XXVI, page 1160, for *under*ground pipe lines]:

1. *When Cast Vertically (Pit-cast) in Dry-sand Molds*

Pit-cast pipe shall have a nominal wall thickness not less than specified by Table 4 of ASA A21.2 for Laying Condition "C" with 3½ ft of cover, and for an arbitrary design pressure 50 psi greater than the actual working pressure. No interpolation between pressures shown in Table 4 is permitted, and the thickness for the tabulated 50 psi pressure increment next above the arbitrary design pressure shall be used. (Consideration is being given this wording as a substitute for obsolete requirements of the 1942 code.)

2. *When Centrifugally or Horizontally Cast*

Service pressure not over 100 psi (Sizes 4 to 24 in., inclusive)	Thickness as specified in F.S.E.C. WW-P-421, Class 150
Service pressure over 100 but not over 150 psi (Sizes 4 to 24 in., inclusive)	Thickness as specified in F.S.E.C. WW-P-421, Class 250

(For abstracts of the foregoing specifications, see pages 420 to 448.)

Cast-iron pipe, when used for gas or oil piping within refinery limits, shall be capable of withstanding a minimum hydrostatic test pressure of three times the service pressure for which the line is designed or under which it will operate, and in no case less than 300 psi.

Cast-iron Pipe outside Refinery Limits.—Cast-iron pipe may be used for oil piping outside of refinery limits where the temperature does not exceed 300 F. When vertically cast in dry sand molds, the metal thickness shall not be less and the service pressure more than specified in ASTM A44 and ASA A21.2, for the class of pipe used (see pages 432 to 437). When centrifugally or horizontally cast, in sizes 3 to 24 in., inclusive, the metal thickness shall not be less than specified in FSEC WW-P-421 for Class 150. The pressures shall not be greater than allowed in that specification for the class of pipe used.

Centrifugally or horizontally cast cast-iron pipe may be used for higher service pressures than tabulated in FSEC WW-P-421, provided the metal thicknesses are not less than allowed by formula (16*e*), page 42, when p equals the service pressure plus the allowance for water hammer tabulated therewith, $S = 8{,}400$ psi based on a tensile strength of 30,000 psi, and $C = 0.14$.

In addition to above limitations, cast-iron pipe lines shall, when installed, be suitable for a hydrostatic test pressure of not less than twice the designed service pressure and in no case less than 150 psi test pressure. The test pressures shall be as given in ASTM and Federal Specifications referred to above for the various

All cast-iron pipe shall have either bolted flange connections, threaded joints, sleeve or gland couplings, or gland-packed joints made integral with the pipe or special joints such as rubber or flexible rings held tight by pressure and properly retained, or bell-and-spigot pipe connections.

When cast-iron pipe is used above ground with joints of the packed-gland type, or others offering no positive mechanical connection between the joined sections, the pipe shall be anchored in such a way as to prevent the joints pulling apart.

Flanges.—The dimensions of all line flanges shall be at least equal to the requirements of the standards referred to for the respective pressures and temperatures.

Valves.—All valves, wherever practicable, except as noted below, 2½ in. in size and larger, shall be of the outside-screw, rising-stem type. Under 2½ in., the valves may be of the inside-screw, rising-stem type. Valves used on lines containing acid shall be preferably of the inside-screw, nonrising-stem type.

Valves on cold oil trunk line service may be either of the rising-stem or nonrising-stem type. Valves to be operated under conditions where it is unnecessary to know whether they are open or closed from a safety standpoint may be either of rising or nonrising-stem type. It is recommended that all nonrising stem valves, except valves smaller than 2½ in., be equipped with indicators to show position of gates or disks. Plug valves may be used instead of gate, globe, or other types. See Special Valves and Plug Valves for provisions.

Body metal thickness of steel valves shall not be less than that of fittings for corresponding material, size, and rating. Every valve shall be plainly marked in accordance with the requirements of MSS Standard Practice SP-25 (see abstract, page 687).

Valves covered by Classes 1 and 2 in sizes 4 in. and larger shall have bolted bonnets. Valves smaller than 4 in. may have screw bonnets, preferably of the union type. It is recommended that valves 4 in. and larger have flanged ends.

Under Classes 3 and 4, valves 2½ in. and larger shall have bolted bonnets excepting flow-line gate valves with steel union bonnets. Valves under these classes, smaller than 2½ in., may have screw bonnets of the union type, but it is recommended that 1½ in. valves and larger under these classes have bolted bonnets. It is recommended that valves under these classes, 2½ in. and larger, shall have flanged ends when used at temperatures above 450 F. Valves with welding ends may be used and shall be welded directly into the pipe line in accordance with Sec. 6, Chap. 4 of the Code.

Brass or bronze trimming or yoke bushings shall not be used in valves where the service temperature is greater than 400 F. For higher temperatures, bushings and trim may be of an alloy steel or other material having a melting point of 2000 F or over.

Check Valves.—Ball check valves of the straight-through type are recommended for Class 3 for temperatures above 750 F. When ball check valves are used in horizontal positions, the balls shall be guided so as to move in as near a horizontal line as practicable. Ball check valves with globe type bodies are not recommended where the service temperature may lead to coke deposits.

Special Valves and Plug Valves.—All special valves, such as plug valves, relief valves, etc., shall conform to the material and strength specifications required for the class of service in which they are used.

For high temperature service, where plug valves are used, they shall be so designed as to prevent galling, either by making the plugs of a different material from the body of the plug valve or by treating the plug to ensure different physical properties, or by special mechanical designs. Where plug valves are used for services covered by Class 3, special consideration shall be given to their design, so as to ensure that they can be readily operated under hot oil conditions. Plug valves which are not gear or worm operated shall either have their operating levers attached to the stem, or the levers shall be maintained in locations readily accessible for emergency operation.

Relief Valves and Automatic Control Valves.—All automatic control valves shall be by-passed or otherwise installed in the line so as to permit manual operation in the event of failure of the automatic valves. Relief valves on equipment where the service pressure is more than 7½ psi shall be connected by piping so as to discharge at places where the danger from flashing will be a minimum.

Where equipment, within which pressure can be generated because of service conditions, is protected by a relief valve, there shall be no block valves permitted between the vessel and its relief valve, except for inspection or repair purposes as permitted below. Further exceptions are in cases where multiple relief valves are provided and the block valves are constructed in such a manner that they cannot be operated so as to reduce the pressure relieving capacity below that required.

Where equipment, within which pressure originates from an outside source exclusively, is protected by a relief valve, there may be block valves between the vessel and its relief valve which need not be locked open if the block valve also closes the vessel from its source of pressure. Any block valve between the relief valve and the vessel which does not close the vessel from its source of pressure shall meet the requirements given in the previous paragraph.

Where equipment, which is normally in operation for indefinite periods at temperatures below 200 F, is protected by a relief valve, it may have a block valve between it and its relief valve for use in inspection and repair. In this case the block valve shall be arranged so that it can be locked open at all times, except for short periods when it is necessary to remove the safety valve for purpose of repair.

The use of gate valves as block valves is not permitted, and globe valves as permitted above shall be installed with the pressure under the disk so as to open with the pressure, except that this limitation need not apply to block valves closing the vessel from its source of pressure.

It is recommended that relief or vent valves be provided on piping between block valves where thermal expansion of the contents of the line may cause a hazard.

Double Valving and Drips and Drains.[1]—All oil piping connections to vessels or units of equipment which may be isolated from operating units to be entered by men for cleaning or repairs shall be installed with double valves and bleeders, or else so arranged that they can be disconnected or blinded. When double

[1] This requirement is not intended to apply to lines connected to "oil tankage" at atmospheric pressure.

block valves are used, one of the valves shall be placed as close as possible to the equipment to which the pipe connects so as to preclude any pipe or nipple between this valve and the equipment.

Suitable drips or drains shall be installed between these double block valves. The minimum size of drip connections for lines operating at a temperature of 600 F or lower shall be ½ in. The minimum size of drip connections for lines operating at over 600 F shall be ¾ in. The valves and fittings used on drip and drain lines shall be of the same class as the valves and fittings on the main line to which the "drip" connects. Valves for use in drain lines should preferably be designed with concentric ports through body and plug or disk, and with an absence of pockets, etc., tending to impair drainage.

Fittings.—It is recommended that within refinery limits the use of steel screwed fittings and cast-iron or malleable-iron screwed fittings and screwed flanges for oil or refinery gas lines be limited to the maximum nominal sizes given in Tables XXI and XXII. For conditions coming between those given in these tables, interpolations may be made. It is recommended that screwed joints on steel fittings for pressures of 300 psi or over or at temperatures of 750 F or over be seal welded. The weld metal should cover all exposed pipe threads. Fittings shall be marked in accordance with the requirements of MSS Standard Practice SP-25.

Special Fittings.—Where special fittings differing in over-all dimensions from American Standards are required, they shall conform to those standards with

TABLE XXI.—RECOMMENDED MAXIMUM NOMINAL SIZES IN INCHES FOR CAST OR FORGED STEEL SCREWED FITTINGS

Temperature F	Service pressures, psi					
	150	300	400	600	900	1,500
Atmospheric	12	12	12	10	6	4
200	12	10	8	6	4	3
300	10	8	6	4	4	3
400	8	6	4	4	3	3
600	6	4	3	3	2	2
900	4	3	2	2	1½	1½
950	3	2½	2	1½	1½	1½

TABLE XXII.—RECOMMENDED MAXIMUM NOMINAL SIZES IN INCHES FOR CAST- OR MALLEABLE-IRON SCREWED FITTINGS AND FLANGES

Temperature F	Service pressures, psi						
	100	250	400	600	800	1,000	1,500
Atmospheric	16	14	12	10	8	6	4
300	8	6	6	4	3	2½	2½
400	6	6	3	3	3	2½	2½

respect to minimum wall thickness and flange dimensions. All materials shall conform to the requirements for standard fittings for the particular pressure and temperature ratings.

Welded steel fittings such as elbows, tees, etc., and welded assemblies such as built-up manifolds, segmental elbows, etc., may be used if made in accordance with the requirements of the Sec. 6, Chap. 4 of the Code.

The dimensions, strength, and marking of factory-made welding fittings shall comply with American Standard for Steel Butt-welding Fittings (B16.9) or American Standard for Steel Socket-welding Fittings (B16.11[1]), where applicable, and the material shall conform to ASTM Specifications A216, A217, or A234. Special fittings or welded assemblies fabricated in shop or field shall conform to Sec. 6, Chap. 5, of this code. The material of special welding fittings shall conform to ASTM Specifications A216, A217, or A234. For the purposes of this section, special fittings or welded assemblies shall be construed to embrace cast, forged, rolled, or extruded fittings of special dimensions, and built-up manifolds, segmental elbows, fabricated swages, orange-peel bull plugs, or any similar construction not covered by American Standards B16.9 or B16.11 for factory-made welding fittings.

Bolting.—Commercial-steel or wrought-iron machine bolts may be used to make flange connections for Class 1 and 2 services which involve pressures of 125 and 250 psi and temperatures of 350 and 450 F, respectively. They may be used also for Class 4 service which covers cold-oil trunk lines at high pressures.

Alloy-steel bolts and bolt-studs shall be used for Class 3 service conditions which embrace temperatures above 500 F and pressures above 300 psi. Bolting shall be in accordance with Class B or C of ASTM A96 or ASTM A193. Alloy-steel bolts and bolt-studs shall be marked to distinguish them from carbon steel.

Carbon-steel bolts may have square heads of "Regular Series" or hexagonal heads of "Heavy Series" in accordance with American Standard for Wrench Head Bolts and Nuts, ASA B18.2 (see abstract, page 550).

Alloy-steel bolts are recommended to have hexagonal heads of the "Heavy
Bolt-studs threaded at both ends are strongly recommended for high-temperature services.

Bolts and bolt-studs ½ to 1 in., inclusive, shall be threaded with the coarse-thread series, 1⅛ in. and larger to the 8-pitch series of the American Standard Screw Threads, ASA B1.1 (see abstract, page 546). See also American Standard Screw Threads for High-strength Bolting, ASA B1.4, abstracted on page 548. Dimensions of nuts shall be in accordance with the "Heavy Series" of ASA B18.2. Material shall be in accordance with ASTM A194 (see abstract, page 544).

Pipe Threads.—Pipe shall be threaded in accordance with API specifications for Line Pipe, API 5L (see abstract, page 1136) or American Standard Pipe Thread, ASA B2.1, (see abstract, page 481). NOTE.—With the exception of the 2-in. size which has about ¼ in. longer thread, the dimensions of pipe threaded in accordance with API 5L are identical with ASA B2.1. Threads on valves and fittings shall be suitable for the corresponding pipe threads (see API Standard 5F, abstracted on page 1115).

Pipe Joints.—Flanges shall be attached to ferrous pipe by threading, expanding, lapping, welding, or other approved method described in Sec. 6, Chap. 5 of the Code. It is recommended that the hubs of all threaded steel flanges on

[1] Before the ASA for approval.

lines operating at 300 psi pressure or over, or 750 F or over, be seal welded to the pipe. The welds should cover all exposed pipe threads.

Plain or raised facings shall not be used for temperatures over 750 F. Male and female facings shall not be used for pressures over 300 psi when the temperature exceeds 900 F.

Cast-iron flanges used at the ends of pipe bends which are subjected to the strains of expansion and contraction shall not be lighter than the 250-lb American Standard B16b on the 150-lb American Standard B16e, even where on other parts of the line 125-lb fittings are permissible.

Valves, fittings, and flanges shall be attached to nonferrous pipe or tubing by threading (when pipe is made to iron-pipe-size dimensions), by compression type, or flared, flanged, or lapped connections. Socket brazed type joints also may be used provided the brazing alloy contains not less than 60 per cent copper and has a melting point above 1000 F. The brazed joint shall extend the full depth of the socket. Fillet brazed joints are not acceptable.

Gasket Material.—Gaskets for piping operating at over 250 F or for piping adjacent to a hot oil line shall be metallic, asbestos, or other noncombustile material. Compressed asbestos gaskets are not recommended for temperatures above 750 F. Gaskets shall be as thin as the finish of the contact surfaces will permit. For temperatures of 750 F and over where the operating pressure is 400 psi or over, except where ring joints or Sargol or Sarlun joints are used, metal or metallic asbestos-filled gaskets shall be used. Rings for ring joints shall be of a metal not injuriously affected by the fluid conveyed. (See abstracts of ASA B16e, page 626, and API 5G-3, page 1125.) For information on gasket compressions necessary to maintain tight joints, see page 511.

Thermal Expansion.—Provision for thermal expansion shall be made in accordance with Sec. 6, Chap. 5 of the Code (see page 832). Changes in direction of high-pressure and -temperature oil and oil-vapor lines shall preferably be made with seamless pipe bends rather than with fittings. Where expansion slip joints are used, they shall be slip guided with long stuffing bores and shall be provided with limit stops. The trim shall, in general, conform to that of valves for the same class of service, except that special precaution against "freezing" shall be taken.

For pressures not exceeding 125 psi and temperatures below 350 F, swivel joints or swing joints made up of elbows and short lengths of threaded pipe may be used. This type of joint is not recommended for higher pressure and temperatures.

Expansion and contraction of cold oil trunk lines, Class 4 of the Oil Piping classification, shall be provided for by laying the line with sufficient "slack" or curvature. Where expansion joints are used on underground lines, suitable covers shall be provided to permit their functioning properly. (See also pages 765 to 769.)

Hangers, Supports, and Anchors.—Hangers and supports shall be designed in accordance with Sec. 6, Chap. 5 of the Code (see page 722). They shall be of non-combustible material and, where located so that they are liable to destruction by fire, shall be insulated to eliminate chance of failure of supports which might cause a break in the oil line. Care shall be taken to properly anchor and guide pipe where slip expansion joints are used so that pipe can move only in a direction parallel to the center line of the pipe (see also page 721).

Pipe Covering.—Insulation of high-temperature oil or vapor lines shall be protected with a fire-resistant covering of proper mechanical strength to resist

TABLE XXIII.—ALLOWABLE STRESS (S VALUES) PSI FOR PIPE IN OIL PIPING SYSTEMS WITHIN REFINERY LIMITS[4]—LAP-, BUTT-, AND FORGE-WELDED STEEL AND WROUGHT-IRON PIPE (Tables 24 and 25 ASA B31.1)

Specification	Lap-welded steel ASTM A53 ASTM A106 API 5L	Butt-welded steel ASTM A53 API 5L	Forge-welded steel ASTM A136		Wrought iron, API 5L	
Temperature deg F	TS = 45,000 psi YP = 25,000 psi	TS = 50,000 psi YP = 30,000 psi	Grade A TS = 45,000 psi YP = 24,000 psi	Grade B TS = 50,000 psi YP = 27,000 psi	Lap welded TS = 42,000 psi YP = 24,000 psi	Butt welded TS = 42,000 psi YP = 24,000 psi
	Silicon killed[1,5] and low silicon killed[2,5]					
100	11,250	10,800	11,500	12,950	10,800	7,200
125	11,100	10,650	11,400	12,800	10,600	7,100
150	11,000	10,500	11,250	12,700	10,400	7,000
175	10,850	10,350	11,150	12,650	10,250	6,850
200	10,750	10,200	11,050	12,400	10,050	6,750
225	10,600	10,050	10,900	12,300	9,850	6,650
250	10,500	9,900	10,800	12,150	9,700	6,550
275	10,350	9,750	10,700	12,000	9,500	6,450
300	10,250	9,600	10,600	11,900	9,300	6,350
325	10,100	9,450	10,450	11,750	9,100	6,250
350	10,000	9,300	10,350	11,600	8,950	6,150
375	9,850	9,150	10,250	11,450	8,750	6,050
400	9,700	9,000	10,150	11,350	8,550	5,950
425	9,600	8,850	10,000	11,200	8,400	5,850
450	9,450	8,700	9,900	11,050	8,200	5,750
475	9,350	8,550	9,800	10,900	8,000	5,650
500	9,200	8,400	9,650	10,800	7,850	5,550
525	9,100	8,250	9,550	10,650	7,650	5,450
550	8,950	8,100	9,450	10,550	7,500	5,300
575	8,800	7,950	9,350	10,400	7,300	5,200
600	8,700	7,800	9,200	10,250	7,100	5,100
625	8,550	7,650	9,100	10,150	6,900	5,000
650	8,450	7,500	9,000	10,000	6,750	4,900
675	8,350	7,400	8,900	9,850	6,550	4,800
700	8,200	7,200	8,750	9,600	6,350	4,700
725	8,050	6,950	8,550	9,300	6,200	4,600
750	7,850	6,700	8,350	8,950	6,000	4,500
	Rimmed[3]					
675	8,250	7,350	8,800	9,800		
700	8,000	7,100	8,500	9,500		
725	8,050	6,750	8,200	9,000		
750	7,300	6,400	7,800	8,500		

TS = tensile strength; YP = yield point.

[1] Silicon 0.10 to 0.25 per cent.

[2] Silicon below 0.10 per cent.

[3] Not "killed steel," values below 650 F same as silicon killed.

[4] See Table XXVII for S values for oil piping outside of refinery limits.

[5] While it is possible to purchase pipe to this analysis, the silicon content shown is not in accordance with the specification listed.

The above values are to be used in formulas for determining the minimum allowable pipe wall thickness (see pp. 42 and 45).

In all cases, for temperatures above 650 F, if the silicon content of the pipe material is not known to be within the limit specified herein for "silicon killed" or "low silicon killed" steel, the values given above for "rimmed" steel shall be used.

TABLE XXIV.—ALLOWABLE STRESS (S VALUES) PSI FOR PIPE IN OIL PIPING SYSTEMS WITHIN THE REFINERY LIMITS[6]—ELECTRIC-WELDED PIPE
(Table 23 of ASA B31.1)

Specification	Electric fusion-welded steel[4] ASTM A 155			Electric fusion-welded steel[4] API 5L			Electric resistance-welded steel[4,5] API 5L		
Temperature deg F	Grade A TS = 45,000 psi YP = 24,000 psi	Grade B TS = 50,000 psi YP = 27,000 psi	Grade C TS = 55,000 psi YP = 27,500 psi	Grade A TS = 48,000 psi YP = 30,000 psi	Grade B TS = 60,000 psi YP = 35,000 psi	Grade C TS = 75,000 psi YP = 45,000 psi	Grade A TS = 48,000 psi YP = 30,000 psi	Grade B TS = 60,000 psi YP = 35,000 psi	Grade C TS = 75,000 psi YP = 45,000 psi
	Silicon killed[1,7] and low silicon killed[2,7]								
100	13,000	14,600	14,950	14,400	16,800	21,600	15,300	17,850	23,000
125	12,900	14,450	14,800	14,200	16,600	21,300	15,050	17,600	22,650
150	12,750	14,300	14,700	14,000	16,400	21,000	14,850	17,400	22,300
175	12,650	14,150	14,600	13,750	16,150	20,700	14,600	17,150	22,000
200	12,500	14,000	14,500	13,550	15,950	20,400	14,400	16,900	21,700
225	12,400	13,850	14,350	13,300	15,700	20,100	14,150	16,700	21,350
250	12,250	13,700	14,250	13,100	15,500	19,800	13,900	16,450	21,050
275	12,100	13,500	14,150	12,900	15,300	19,500	13,700	16,200	20,700
300	12,000	13,400	14,000	12,650	15,050	19,200	13,450	16,000	20,400
325	11,850	13,200	13,900	12,450	14,850	18,900	13,200	15,750	20,050
350	11,750	13,100	13,800	12,200	14,600	18,600	13,000	15,500	19,750
375	11,600	12,900	13,650	12,000	14,400	18,300	12,750	15,300	19,400
400	11,450	12,750	13,550	11,800	14,200	18,000	12,500	15,050	19,100
425	11,350	12,600	13,450	11,550	14,000	17,700	12,300	14,800	18,800
450	11,200	12,450	13,300	11,350	13,750	17,400	12,050	14,600	18,450
475	11,100	12,300	13,200	11,100	13,500	17,100	11,800	14,350	18,150
500	10,950	12,150	13,100	10,900	13,300	16,800	11,600	14,100	17,850
525	10,800	12,000	12,950	10,700	13,100	16,500	11,350	13,900	17,500
550	10,700	11,850	12,850	10,500	12,850	16,200	11,100	13,650	17,150
575	10,550	11,700	12,750	10,250	12,650	15,900	10,900	13,450	16,850

TABLE XXIV.—(*Concluded*)

Specification	Electric fusion-welded steel[4] ASTM A 155			Electric fusion-welded steel[4] API 5L			Electric resistance-welded steel[4,5] API 5L		
Temperature deg F	Grade *A* $TS =$ 45,000 psi $YP =$ 24,000 psi	Grade *B* $TS =$ 50,000 psi $YP =$ 27,000 psi	Grade *C* $TS =$ 55,000 psi $YP =$ 27,500 psi	Grade *A* $TS =$ 48,000 psi $YP =$ 30,000 psi	Grade *B* $TS =$ 60,000 psi $YP =$ 35,000 psi	Grade *C* $TS =$ 75,000 psi $YP =$ 45,000 psi	Grade *A* $TS =$ 48,000 psi $YP =$ 30,000 psi	Grade *B* $TS =$ 60,000 psi $YP =$ 35,000 psi	Grade *C* $TS =$ 75,000 psi $YP =$ 45,000 psi
	Silicon killed[1,7] and low silicon killed[2,7]								
600	10,400	11,550	12,650	10,000	12,400	15,600	10,650	13,200	16,550
625	10,300	11,400	12,500	9,800	12,200	15,300	10,400	13,000	16,200
650	10,150	11,250	12,400	9,600	12,000	15,000	10,200	12,750	15,900
675	10,050	11,100	12,150	9,500	11,800	14,700	10,100	12,500	15,650
700	9,850	10,800	11,700	9,250	11,400	14,100	9,800	12,100	15,000
725	9,650	10,450	11,250	9,000	11,000	13,300	9,550	11,700	14,150
750	9,400	10,100	10,700	8,700	10,400	12,400	9,250	11,050	13,200
	Rimmed[3]								
675	9,900	11,000	......	9,400	......	......	10,000		
700	9,600	11,650	11,700	9,100	11,400	14,100	9,650	12,100	15,000
725	9,250	10,100	11,000	8,700	10,700	13,200	9,200	11,350	14,000
750	8,750	9,600	10,350	8,250	9,950	12,100	8,750	10,600	12,850

TS = tensile strength; YP = yield point.

1 Silicon 0.10 to 0.25 per cent.

2 Silicon below 0.10 per cent.

3 Not "killed steel," values below 650 F same as silicon killed.

4 In all cases for temperatures above 650 F, if the silicon in the pipe material is not known to be within the limits specified herein for "silicon killed" or "low silicon killed" steel, the values given for "rimmed" steel shall be used.

5 For electric-resistance-welded pipe for applications where the temperature is below 650 F and where pipe furnished under this classification is subjected to supplemental tests and/or heat treatments as agreed to by the supplier and the purchaser, and whereby such supplemental tests and/or heat treatments demonstrate the strength characteristics of the weld to be equal to the minimum tensile strength specified for the pipe, the S values equal to the corresponding seamless grades may be used.

6 See Table XXVII for S values for oil piping outside of refinery limits.

7 While it is possible to purchase pipe to this analysis, the silicon content shown is not in accordance with the specification listed.

NOTE.—The above values are to be used in pipe wall thickness formulas for determining minimum allowable pipe wall thickness (see pp. 42 and 45).

TABLE XXV.—ALLOWABLE STRESS (S VALUES) PSI FOR PIPE IN OIL PIPING SYSTEMS WITHIN REFINERY LIMITS—SEAMLESS PIPE
(Tables 22 and 26 ASA B31.1)

	Seamless steel pipe						Seamless alloy pipe			
Specification	API 5L ASTM A 106 ASTM A 53**		API 5L ASTM A 106* ASTM A 53**		API 5L		ASTM A 158		ASTM A 206	
Temperature deg F	Grade A TS = 48,000 psi YP = 30,000 psi		Grade B TS = 60,000 psi YP = 35,000 psi		Grade C TS = 75,000 psi YP = 45,000 psi		Ferritic 5 chromium TS = 60,000 psi YP = 30,000 psi	Austenitic 18 Cr-8 Ni TS = 75,000 psi YP = 30,000 psi	Carbon molybdenum TS = 55,000 psi YP = 30,000 psi	Temperature deg F
	Silicon killed[1,5]	Rimmed[3]	Silicon killed[1,5]	Rimmed[3]	Silicon killed[1,5]	Rimmed[3]				
100	18,000		21,000		27,000		18,000	18,000	18,000	100
125	17,700		20,700		26,600		17,850	17,900	17,800	125
150	17,450		20,450		26,250		17,700	17,750	17,650	150
175	17,150		20,150		25,850		17,600	17,650	17,500	175
200	16,900		19,900		25,500		17,450	17,550	17,350	200
225	16,600		19,600		25,150		17,300	17,400	17,200	225
250	16,350	Same as silicon killed	19,350	Same as silicon killed	24,750	Same as silicon killed	17,200	17,300	17,000	250
275	16,100		19,100		24,400		17,050	17,200	16,850	275
300	15,800		18,800		24,000		16,900	17,100	16,700	300
325	15,550		18,500		23,650		16,800	16,950	16,500	325
350	15,300		18,250		23,250		16,650	16,850	16,350	350
375	15,000		18,000		22,900		16,500	16,700	16,200	375
400	14,700		17,700		22,500		16,400	16,600	16,050	400
425	14,450		17,450		22,100		16,200	16,500	15,900	425
450	14,150		17,200		21,750		16,100	16,400	15,700	450
475	13,900		16,900		21,400		15,950	16,300	15,550	475
500	13,600		16,600		21,000		15,800	16,150	15,400	500
525	13,350		16,350		20,650		15,700	16,000	15,200	525
550	13,100		16,100		20,250		15,550	15,900	15,050	550
575	12,800		15,800		19,900		15,400	15,800	14,900	575

600	12,550		15,550		19,500		15,700	15,300	14,750	600
625	·12,250		15,250		19,150		15,600	15,150	14,600	625
650	12,000		15,000		18,750		15,500	15,000	14,400	650
675	11,850	11,750	14,750		18,400		15,350	14,900	14,250	675
700	11,550	11,400	14,250		17,650		15,250	14,800	14,100	700
725	11,250	10,850	13,750	13,350	16,650	16,650	15,150	14,700	13,950	725
750	10,900	10,300	13,000	12,450	15,500	15,100	15,000	14,600	13,800	750
775	10,500	9,700	12,250	11,450	14,000	13,450	14,900	14,450	13,600	775
800	10,000	9,050	11,400	10,400	12,600	11,600	14,750	14,300	13,500	800
825	9,300	8,250	10,350	9,200	11,150	9,700	14,400	14,150	13,300	825
850	8,550	7,300	9,250	7,950	9,750	8,100	14,000	14,000	13,150	850
875	7,800	6,550	8,150	6,750	8,350	6,750	13,400	13,800	12,800	875
900	7,000	5,500	7,000	5,500	7,000	5,500	12,500	13,600	12,250	900
925	5,850	4,350	5,850	4,350	5,850	4,350	11,200	13,300	11,250	925
950	4,750	3,250	4,750	3,250	4,750	3,250	10,000	12,850	10,000	950
975	3,600	......	3,600	......	3,600	......	8,750	12,300	8,150	975
1,000	2,500	......	2,500	......	2,500	......	7,300	11,600	6,250	1,000
1,025	......	......	......	......	......	......	6,000	10,650	4,500	1,025
1,050	......	......	......	......	......	......	4,800	9,300	3,000	1,050
1,075	......	......	......	......	......	......	3,700	8,400	2,100	1,075
1,100	......	......	......	......	......	......	2,750	7,500	1,500	1,100
1,125	......	......	......	......	......	......	2,200	6,600	......	1,125
1,150	......	......	......	......	......	......	1,800	5,800	......	1,150
1,175	......	......	......	......	......	......	1,400	5,100	......	1,175
1,200	......	......	......	......	......	......	1,000	4,500	......	1,200

TS = tensile strength; YP = yield point.
* Silicon killed only.
** Not recommended for temperatures above 750 F.
[1] Silicon 0.10 to 0.25 per cent.
[2] Silicon less than 0.10 per cent, values 750 F and below same as silicon killed, above 750 F same as rimmed.
[3] Not "killed" steel.
[4] See Table XXVII for S values for oil piping outside of refinery limits.
[5] While it is possible to purchase pipe to this analysis, the silicon content shown is not in accordance with the specification listed.

In all cases, for temperatures above 650 F if the silicon in the pipe material is not known to be within the limits specified hereon for "silicon killed" or "low silicon killed" steel, the values given for "rimmed" steel shall be used.

NOTE.—The above values are to be used in pipe wall thickness formulas (see pp. 42 and 45).

destruction by water from a fire hose. For typical selection of insulation for oil-refinery piping, see Table VI, page 718.

Proximity of Cold-oil Lines to Hot Lines.—Cast-iron valves and fittings shall not be used on cold-oil lines located in the proximity of hot-oil lines or equipment where failure of the valves or fittings would permit splashing of oil on hot surfaces thus creating a fire hazard.

Inspection and Tests.—All valves, fittings, and pipe shall be capable of withstanding the hydrostatic test pressures specified for the service pressure rating in the applicable ASA and API standards.

After Erection.—All assembled piping, except high-pressure trunk lines, shall be capable of withstanding, without injury or leakage, a hydrostatic or oil test pressure of not less than 1½ times the maximum working pressure allowed by this Code for the weakest piping element pipe, valve, or fitting or connected equipment included in the piping system. The test pressure for steel pipe, flanges, valves, and fittings shall be determined as follows:

Pipe.—One and one-half times the maximum allowable pressure at 100 F as determined by the formulas on pages 42 and 45 and S values of Tables XXIII to XXVII, inclusive.

Flanges, Valves, and Fittings.—One and one-half times the primary service pressure rating but in no case less than one and one-half times the actual working pressure.

High-pressure trunk pipe lines shall be capable of withstanding one and one-half times the service pressure except when this test pressure would result in stressing the material above its elastic limit.

Thickness of Pipe.—The minimum thickness of pipe wall required for different pressures at temperatures not exceeding those given in Tables XXIII to XXVII, inclusive, shall be computed by means of formulas (16*e*) and (16*f*) on page 42. The manufacturing tolerances on wall thickness contained in the respective standard pipe specifications must be added to the minimum thickness so determined to obtain the nominal wall thickness to use in ordering pipe.

It is recommended that no threaded pipe lighter than standard threaded API line pipe shall be used. Where two or more weights of standard threaded

TABLE XXVI.—ALLOWABLE STRESS (S VALUES) PSI FOR PIPE WITHIN REFINERY LIMITS—SEAMLESS BRASS AND COPPER PIPE, SEAMLESS COPPER TUBING, AND CAST-IRON PIPE
(From Table 25 ASA B31.1)

Brass and Copper	S Values at 350 F and Below
Seamless brass pipe, ASTM B 43	7,000
Seamless copper pipe, ASTM B 42	5,400
Seamless copper tubing, ASTM B 75, B 88	5,400
Cast-iron Pipe	**S Values for Underground Pipe Lines, 300 F and Below**
When casting vertically in dry sand molds	4,000
When centrifugally or horizontally cast	6,000

pipe are obtainable in any diameter, the heavier or heaviest is referred to in this recommendation.

Table XXVII.—Allowable Stress (*S* Values) Psi for Pipe in Oil Piping Systems Outside of Refinery Limits (Table 27 ASA B31.1)

Material	Specification	*S* value at 100 F, psi
Seamless steel:		
Grade *A*	API 5L	25,500
Grade *B*	API 5L	29,750
Grade *C*	API 5L	38,250
Grade *A*	ASTM A 106	25,500
Grade *B*	ASTM A 106	29,750
Grade *A*	ASTM A 53	25,500
Grade *B*	ASTM A 53	29,750
Electric-fusion-welded steel:		
Grade *A*	API 5L	20,400
Grade *B*	API 5L	23,800
Grade *C*	API 5L	30,600
Electric-fusion-welded steel (high pressure—high temperature service):		
Grade *A*	ASTM A 155	18,350
Grade *B*	ASTM A 155	20,650
Grade *C*	ASTM A 155	21,200
Electric-resistance-welded steel:[1]		
Grade *A*	API 5L	21,700
Grade *B*	API 5L	25,300
Grade *C*	API 5L	32,500
Lap-welded steel	API 5L	15,950
	ASTM A 106	15,950
	ASTM A 53	15,950
Butt-welded steel	API 5L	15,300
	ASTM A 53	15,300
Forge-welded steel:		
Grade *A*	ASTM A 136	16,300
Grade *B*	ASTM A 136	18,350
Lap-welded wrought iron	API 5L	15,300
Butt-welded wrought iron	API 5L	10,200
Seamless brass pipe	ASTM B 43	10,000
Seamless copper pipe	ASTM B 42	7,500
Seamless copper tubing	ASTM B 75	7,500
	ASTM B 88	7,500

[1] For electric-resistance-welded pipe for applications where pipe furnished under this classification is subjected to supplemental tests and/or heat treatments as agreed to by the supplier and the purchaser, and whereby such supplemental tests and/or heat treatments demonstrate the strength characteristics of the weld to be equal to the minimum tensile strength specified for the pipe, the *S* values equal to the corresponding seamless grades may be used.

NOTE.—The above values of *S* are to be used in the pipe wall thickness formulas given on pages 42 and 45.

In addition to providing the thickness required by the actual pressure, the minimum thickness of pipe used shall not be less than the thickness required for mechanical strength as determined by the best engineering data available. Where seamless pipe is commercially available, it shall be used for temperatures above 750 F. For larger sizes where seamless is not commercially manufactured,

welded pipe may be used, provided the following joint efficiencies are taken into account in determining the thickness.

Joint efficiencies to be used in connection with determinations of S values for pipe materials not covered in the tables:

Seamless	1.00
Fusion welded, ASME rules (Par. U 68)	0.90
Resistance welded	0.85
Fusion welded	0.80
Forge welded	0.80

GLOSSARY OF TERMS PECULIAR TO OIL PIPING

Casing.—Pipe used to line or "case" a well. It is furnished in many different styles, each style having a distinctive scale of graduated sizes or types of thread. The thread is almost always finer than standard pipe thread.

Casing Fittings.—Fittings which are threaded with casing threads and which are usually listed in odd sizes, such as 3¼, 6⅝, 8¼ in., corresponding to the nominal inside diameter of the casing to which they connect.

Casing Head.—A fitting having several outlets and placed on top of the casing in order to separate the oil and gas or to provide pumping connections.

Drill Pipe.—Used where wells are drilled by the rotary method. Has distinctive threaded joint and is usually furnished with an interior upset to withstand the stresses incidental to its being rapidly revolved when drilling.

Drilling Valve.—The master block valve placed at the well head through which the drilling tools pass.

Drive Pipe.—Pipe which is driven into the hole to stop caving or flooding when subterranean water courses are encountered. This pipe is threaded and coupled in such manner that the ends of connecting pipe will butt against each other in order to relieve the strain on the thread during driving.

Go Devil.—An instrument having expanding sides which is placed in pipe lines in order to remove or locate obstructions. It is carried along by the oil flowing in the line and, owing to its design, is given a whirling motion thus scraping the walls of the pipe. In underground lines, the noise of its passage can be heard so that, should there be an obstruction which it could not clear, the exact location of the obstruction can be determined.

Grief Pipe.—That section of drill pipe which is clamped in the rotary machine and to which the other lengths of drill pipe are coupled. So called because of the severe service which it performs.

Line Pipe.—Pipe similar to standard pipe but with longer threads and recessed couplings (see API Specification 5L).

Receiving Houses or Tail Houses.—Buildings to which the streams of distillate from the several condensing worms are carried. The streams pass through "look boxes" where the color and size of the stream may be observed and samples taken, and then pass to a manifold by means of which the operator may direct the distillate to the "run-down pan" used for that product.

River Clamp.—A heavy iron casting made in two pieces and bolted over couplings in sections of pipe lines which are run under water. They stiffen the joint while the pipe is being pulled across the river bed during the process of laying and also furnish additional weight to hold it in place in the river bed.

Run-back Lines.—Lines which carry reflux from the fractionating equipment back to the still.

Run-down Lines.—Pipe lines carrying distillate from the receiving house to the working tanks or run-down pans.

Run-down Pans.—Low tanks located near receiving houses which receive the product from the receiving house. They are used as separators to settle out the water which is present in the oil due to condensation of steam used in processing. The water is drawn off periodically from the bottom of these tanks. From these tanks the product is transferred to storage tanks or for further processing.

Run-in Lines.—Pipe lines carrying distillate from condenser worms to the receiving house.

Screen Pipe or Screen Casing.—A perforated pipe that is placed on the bottom of a string of casing to exclude gravel and other foreign matter injurious to the pumping equipment and valves and also to prevent pulling the oil so rapidly as to break down the oil-sand structure and stop off the well.

Tail Houses.—See Receiving Houses.

Tank Farm.—That section of an oil refinery used for the storage of petroleum products.

Working Tanks.—See Run-down Pans.

CHAPTER XV

GAS PIPING

In America the distribution of *illuminating gas* through pipes began in 1816 as a public utility for lighting buildings and streets in Baltimore, Md. After a development along these lines for nearly a century, the trend of events caused a change in use so that what had been illuminating gas then became *fuel gas* for domestic cooking and for heating in home and industry. The new applications were accompanied by a corresponding change in the composition of the gas, with heating value as the basis of charge rather than candle power. In the meantime, natural gas came more and more into the picture, often being transmitted through long cross-country lines from its origin in the oil fields to remote distributing centers in the big cities.

The year 1825 marked the first commercial use of *natural* gas in the United States. At Fredonia, N.Y., gas was piped through small lead pipes. In 1870 the first pipe line of any consequence was laid to Rochester, N.Y. Two years later, in 1872, the first long iron pipe line in the United States was laid in the vicinity of Titusville, Pa. The natural gas industry developed slowly for several years, as the use of natural gas was restricted to industries in the immediate vicinity of producing wells. Between 1872 and 1890, however, several corporations were organized to transport and sell natural gas to customers in the industrial communities of Pennsylvania, West Virginia, New York, and Ohio. These lines usually were constructed of wrought iron joined with screwed couplings. They seldom exceeded 8 in. in diameter and were operated at a pressure of approximately 80 psi. Since then the distribution of both manufactured and natural gas has become an industry of large proportions, and one that utilizes vast quantities of pipe at pressures ranging from a few ounces to several hundred pounds in different applications.

The author is indebted to the Michigan Consolidated Gas Company for many suggestions followed in formulating this chap-

ter and for much of the material herein presented. The importance of the subject demands that gas and gas piping be accorded considerable attention in other parts of this handbook, as well as in this chapter. Information on the properties and laws of gases and on the flow of gas and air through pipes will be found on pages 142 to 200 in Chap. II on Fluids—Properties of Fluids. Dimensional standards and material specifications for pipe, valves, and fittings suited for gas service will be found among other products in Chap. IV on Pipe, Valves, and Fittings (pages 348 to 693). Special fittings for gas transmission and distribution systems, together with typical details and design recommendations, are included in the present chapter. Code requirements for gas and air piping will be found at the close of the chapter.

GAS TRANSMISSION AND DISTRIBUTION PRESSURES

The following quotation from the "Gas Engineers' Handbook"[1] furnishes an authoritative statement of accepted usage in the gas industry:

> The transmission and distribution of gas involve the design, construction, and operation of transmission mains, compressor plants, and storage facilities; also the selection and installation of such accessories as meters and governors. The distinction between a transmission main and a distribution main is indeterminate, depending upon local usage. In general, a transmission line is characterized by length, high pressure, high-velocity flow, and few taps and fittings. It is customary to use the Weymouth formula in computing flow in this type of line. A distribution main has low pressure, low velocities, and a large number of taps and fittings. The Spitzglass formulas give excellent results with this type of line.

Authors' Note: The Weymouth formula will be found on pages 186 and 194 and the Spitzglass formula on page 184. Other formulas in common use for the flow of high- and low-pressure gas are discussed on pages 165 to 194.

Low-pressure Distribution.—The usual low-pressure distribution system for direct supply to consumers without further regulation or reduction of pressure operates at a few inches of water column above atmosphere. Practice has demonstrated that with *low-pressure distribution* not more than 50 per cent variation either way from an *average* distribution pressure is advisable if stable

[1] Prepared by Gas Engineers' Handbook Committee of The Pacific Gas Association, reviewed by a special subcommittee of the American Gas Association, and endorsed by the directors of same. Published by McGraw-Hill Book Company, Inc., New York, 1934.

combustion conditions are to be maintained. For example, with a distribution pressure of 6 in. of water column, the variation above or below this average should not exceed 3 in. The minimum pressure satisfactory for low-pressure distribution is about 2½ in. of water column, while the maximum pressure is around 20 in. A minimum pressure of at least 2 in. at outlets is required for good performance of appliances and pilot lights.

Intermediate-pressure Distribution.—The usual intermediate-pressure transmission or distribution system before regulation or reduction of pressure for gas service to consumers operates at pressures from a *minimum* of about 20 in. of water volume to a *maximum* of around 15 psi gauge which is said to be about the highest initial pressure for obtaining the best regulation. Pressure of this order usually is produced with either a centrifugal- or a positive-type blower, or else the intermediate-pressure system is fed in turn through a pressure reducing valve from a high-pressure system. Where much pressure is involved, it should be stepped down and controlled with a *regulator* inside, or as it enters, the customer's premises.

High-pressure Distribution.—Sometimes a pressure higher than "intermediate" is desirable for transmission or distribution. This again requires regulation or reduction of pressure before delivery to customers, which involves either stepping down into intermediate-pressure systems and then into low-pressure systems, or else going direct to low-pressure systems for service to customers. High-pressure distribution is especially suited for suburban gas-supply systems where operating pressures range from about 10 psi gauge minimum to around 100 psi gauge maximum. The higher pressures usually are created with reciprocating compressors, while positive-type blowers can be used for the lower pressures. Although customers' services can be taken off high-pressure mains through regulators, the action of the latter is not so satisfactory where the initial pressure is too high. Hence such services are used ordinarily only in sparsely settled neighborhoods where the extra cost of an intermediate- or low-pressure distribution system is not warranted.

Cross-country Transmission Lines for Natural Gas.—These usually operate at pressures ranging from 100 to 500 psi gauge or higher, depending on the length of the line, the frequency of recompressing stations, and other design considerations (see pages 1200 to 1202). Pressure drops between recompressing sta-

tions may run from 100 to 200 psi or more in the higher pressure lines. Pressure of this order requires the use of reciprocating compressors.

Bracing against Pressure.—With piping systems containing bell joints, Dresser couplings, or similar connecting devices where the pipe is free to slip, consideration should be given to preventing the complete disengagement of the parts under pressure. Whereas earth pressure alone may serve the purpose with buried lines under a pressure of only a few ounces of water column, some positive means should be provided with exposed piping or with buried pipes where the pressure is of consequence (see pages 1053 to 1056 in Chap. XII on Water-supply Piping). It is necessary, therefore, to brace such piping where there are turns in the line or dead ends that would give the pressure a chance to act. In buried lines, timber, concrete, or rock bracing is customarily used at such points to transmit the pressure thrust to an earth area sufficiently large to support the load. In lines aboveground mechanical stops of some sort or tie rods are required.

GAS DISTRIBUTION SYSTEMS

Design, Materials, Construction.—The design of gas distribution systems and the selection of materials and construction to suit various applications are discussed in this section. The materials specifications and dimensional standards for items common to other services will be found in Chap. IV on Pipe, Valves, and Fittings, or in Chap. XIV on Oil Piping. Descriptions of items peculiar to gas piping are given in this section in so far as practicable. Starting from the customers' premises and working back toward the source, the usual selections of materials and construction are somewhat as follows:

CUSTOMERS' PREMISES

Basic standards governing the installation of house piping and gas appliances will be found in an AGA publication entitled "Requirements and Recommended Practice for House Piping and Appliance Installation."[1] The following information pertaining

[1] Copies of AGA publications may be obtained from the American Gas Association, 420 Lexington Ave., New York 17, N.Y. Where complete standards are required, or revisions of some later date, reference should be made to the publications of the AGA. Attention also is called to the existence of "American Recommended Practice for the Installation, Maintenance and Use of Piping and

to piping is abstracted from the second edition, issued in 1941. This is divided into mandatory requirements that must be observed and recommended practice, the observation of which is optional.

Although these requirements are designed to deal primarily with piping and appliance installations for domestic use operating at a few inches of water pressure above atmosphere, they are also intended to serve in principle as a guide in industrial and commercial installations although no attempt has been made to cover the wide variety of conditions and applications in this field. These requirements supplement the American Standard Gas Safety Code, ASA K2-1927, and their various provisions agree with the principles established by it.

In installing gas piping and appliances in customers' premises of any sort, due cognizance should be taken of any state or municipal safety regulations and the rules of the local gas company.

AGA REQUIREMENTS AND RECOMMENDED PRACTICE FOR HOUSE PIPING AND APPLIANCE INSTALLATION

Abstracted[1]

Mandatory Requirements

The *piping* shall be constructed and installed so as to be durable, substantial, and gas tight. Piping shall be of a size (see Table I for recommended practice) and so installed as to provide a supply of gas sufficient to meet the maximum demand (see Table II for recommended practice) without undue loss of pressure between the street service and the appliances. Piping shall be so installed as to prevent an accumulation of condensation from interrupting the flow of gas. No device shall be inserted into gas piping that will reduce the cross-sectional area or otherwise obstruct the free flow of gas. (*Authors' Note:* In this connection it is advisable before installing to see that all scale and dirt are blown out of the pipe and the ends reamed where necessary.)

All gas *appliances* and accessories should meet the American Standard Approval or Listing Requirements[2] as to safety, efficiency of operation, and durability of construction. Appliances are any device for utilizing gas, including luminaries.

Appliances shall be adequately supported and so connected to the piping as not to exert undue strain on the connection. Appliances equipped with a

Fittings for City Gas, ASA S27-1933" which is sponsored by the National Fire Protection Association. Copies of the latter can be obtained from the American Standards Association, 29 West 39th St., New York 18, N.Y.

[1] See footnote, p. 1167.

[2] American Standard Listing Requirements for some 35 different appliances and accessories have been issued as separate bulletins which space does not permit referencing here. Application for a publication covering any desired item may be made direct to the American Gas Association Testing Laboratories, 1032 East 62d St., Cleveland, Ohio.

control valve or valves which permit complete shut-off of the gas supply shall not be connected with flexible tubing. Where a flexible gas tubing connection is made, a gas valve always must be provided at the end where the tubing is attached to the house piping. Flexible gas tubing shall be of adequate capacity, gas tight, and so designed as to permit attachment of the nozzles of appliances and to valves connected to the house piping.

Semirigid gas tubing shall be of adequate capacity (see Table III for recommended practice) and so designed as to permit of secure attachment to the inlet connections of appliances and valves. The use of flared tube connectors is recommended. Attention is called to the existence of American Standard Listing Requirements for Semi-rigid Gas Appliance Tubing and Fittings, ASA Z21.24-1941.[1]

Valves controlling several appliances or lighting fixtures shall be placed at an adequate distance from each other so that they will be readily distinguishable. A gas valve or shut-off, which constitutes the only means of gas control, shall be easily accessible and within convenient reaching distance when lighting the burner. No water heating appliance shall be installed in a closed system of water piping unless a water pressure relief valve is provided. When air or oxygen under pressure is used in connection with the gas supply, effective means shall be provided to prevent the air or oxygen from going back into the gas piping. No device or attachment shall be installed on any appliance which may in any way impair combustion.

Recommended Practice

Before proceeding with any job of gas piping, it is first necessary to decide how the piping should be installed in order to meet the requirements of safety

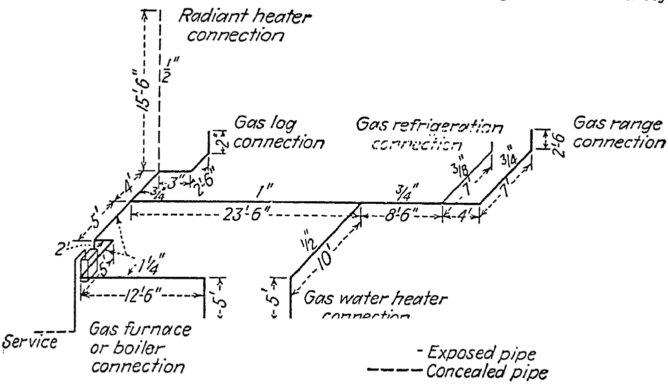


FIG. 1.—Design sketch for gas piping in a house. (*From AGA "Requirements and Recommended Practice for House Piping and Appliance Installation."*)

and service. In those cases where a whole system of house piping is involved, not merely a short extension, it sometimes is desirable to make a piping sketch

[1] For complete information reference may be made to the American Standard (see note, p. 1167).

showing the location of piping and the sizes of the different portions of the system. Such a sketch is shown in Fig. 1.

Size of Piping.—The size of gas pipe necessary to install depends on the following factors:

(*a*) Length of pipe and number of fittings.

(*b*) Maximum gas consumption to be provided for.

(*c*) Allowable loss in pressure from service pipe to appliance.

(*d*) Specific gravity of the gas.

(*e*) Diversity factor, *viz.*, number of appliances that will operate simultaneously.

TABLE I.[1]—CARRYING CAPACITIES OF PIPES[2] OF DIFFERENT DIAMETERS AND LENGTHS IN CUBIC FEET PER HOUR OF LOW-PRESSURE GAS, WITH PRESSURE DROP OF 0.3 IN. AND SPECIFIC GRAVITY OF 0.60

Length of pipe, feet	Nominal diameter of pipe, inches									
	½	¾	1	1¼	1½	2	3	4	6	8
15	76	172	345	750	1,220	2,480	6,500	13,880	38,700	79,000
30	52	120	241	535	850	1,780	4,700	9,700	27,370	55,850
45	43	99	199	435	700	1,475	3,900	7,900	23,350	45,600
60	38	86	173	380	610	1,290	3,450	6,800	19,330	39,500
75	..	77	155	345	545	1,120	3,000	6,000	17,310	35,300
90	..	70	141	310	490	1,000	2,700	5,500	15,800	32,250
105	..	65	131	285	450	920	2,450	5,100	14,620	29,850
120	..	...	120	270	420	860	2,300	4,800	13,680	27,920
150	..	...	109	242	380	780	2,090	4,350	12,240	25,000
180	..	...	100	225	350	720	1,950	4,000	11,160	22,800
210	..	...	...	205	320	660	1,780	3,700	10,330	21,100
240	..	...	...	190	300	620	1,680	3,490	9,600	19,740
270	..	...	...	178	285	580	1,580	3,250	9,000	18,610
300	..	...	...	170	270	545	1,490	3,000	8,500	17,660
450	..	...	...	140	226	450	1,230	2,500	7,000	14,420
600	..	...	...	119	192	390	1,030	2,130	6,000	12,480

[1] From AGA Requirements and Recommended Practice for House Piping and Appliance Installation, 2d ed., 1941.

[2] For Schedule 40 (standard-weight) steel pipe. Approximately equal flows will be obtained with Type *K* copper tubing of the same nominal size.

Authors' Note: Table I can be used in the following manner to solve problems involving over-all pressure drops other than 0.3-in. of water.

Example.—What will be the flow through 300 ft of 2-in. pipe corresponding to an over-all pressure drop of 3.0 in. of water?

Solution.—The length of 2-in. pipe corresponding to a pressure drop of 0.3 in. of water, for the same rate of flow can be determined by proportion as follows: $L/300 = 0.3/3.0$, from which $L = 300(0.3/3.0) = 30$ ft. Referring to Table I, the flow through 30 ft of 2-in. pipe is 1,780 cu ft per hr with a pressure drop of 0.3 in. of water. Hence 1,780 cu ft per hr will flow through 300 ft of 2-in. pipe with a pressure drop of 3.0 in. of water.

In deciding upon the maximum gas consumption to be provided for, adequate allowance should be made for any probable change in the specific gravity or heating value of the gas supply. The carrying capacity of different sizes and lengths of pipe for low-pressure gas having a specific gravity of 0.60 referred to air and with a pressure drop of 0.3 in. of water column are given in Table 1. To convert the figures in this table to any other gravity, multiply the value given by $\sqrt{0.6/s}$ where s is the specific gravity of the gas for the particular condition. By adopting 0.3 in. pressure drop for Table I, due allowance was made for the effect of an ordinary number of fittings.

The volume of gas of a given heating value required to meet the maximum demand of some common appliances may be computed from Table II. (*Author's Note:* The consumption in cubic feet per horsepower for various types of gas engines will be found on page 162.)

TABLE II.[1]—MAXIMUM DEMAND FOR SOME COMMON GAS APPLIANCES

(For accuracy, the manufacturer's Btu rating appearing on name plate should be used)

Appliance	Maximum Demand, Btu per Hr[2]
Domestic gas range (four-burner top)	62,500
Domestic gas range (six-burner top) with extra oven	107,500
Domestic circulating water heater	25,000 to 37,500
Domestic hot plate or laundry stove (per burner)	12,500
Gas steam radiator (per section)	2,000
Domestic room heater—radiant heater:	
Per single radiant	2,000
Per double radiant	4,000
Automatic instantaneous water heaters:	
Capacity 4 gpm	150,000
Capacity 6 gpm	225,000
Capacity 8 gpm	300,000
Automatic storage water heaters:	
Slow recovery	2,500 to 10,000
Quick recovery	15,000 to 70,000
Conversion burner	80,000 to 400,000
Unit heater	50,000 to 900,000
Refrigerator	1,900 to 3,900
Gas boilers	65,000 to 5,000,000
Warm air furnaces	40,000 to 500,000
Floor furnaces	15,000 to 80,000

[1] From AGA Requirements and Recommended Practice for House Piping and Appliance Installation, 2d ed., 1941.

[2] Divide the maximum demand in Btu per hour by the heating value of the local gas in order to get the demand in cubic feet per hour.

The carrying capacities of some of the more common sizes and lengths of semirigid gas tubing are shown in Table III. Where semirigid gas tubing is used to connect an appliance to the house piping, fittings of the flared-tube type should be used.

TABLE III.[1]—CARRYING CAPACITY OF SEMIRIGID TUBING OF DIFFERENT DIAMETERS AND LENGTHS IN CUBIC FEET PER HOUR WITH PRESSURE DROP OF 0.3 IN. AND SPECIFIC GRAVITY 0.60

Length of tubing, feet	Diameter of tubing, inches						
	O.D.	¼	⅜	½	⅝	⅞	1⅛
	I.D.	0.152	0.277	0.402	0.527	0.755	1.005
5	....	4.5	22.5	60	125	305	680
10	....	3.0	15.0	41	85	215	470
15	....	...	12.0	33	68	170	370
20	....	...	....	28	59	140	320

[1] From AGA Requirements and Recommended Practice for House Piping and Appliance Installation, 2d ed., 1941.

Materials.—Black steel *pipe* having threaded ends and a thickness conforming ...edule 40 (standard weight) of ASA B36.10-1935 (see pages 361 to 371) ...ay be furnace welded, resistance welded, or seamless, ... low pressure the first named is usual. The most ...e requirements of good practice are threaded black malleable iron, ASA B16c-1939 (see pages 344, 573 to 580). Pipe and fitting *threads* should comply with the American Standard for Taper Pipe Threads, ASA B2-1942, or latest revision thereof approved by the ASA.

Replacement of Defective Pipe or Fittings.—In no case is it permissible to repair defects in pipe or fittings but, having been located, the defective pipe or fitting must be removed and replaced. In case defects are found on any part of the system containing unmetered gas, the gas company should be promptly notified.

Testing for Tightness.—Before any system of gas piping is finally put in service, it should be carefully tested to ensure that it is gas tight. Where any part of the system is to be enclosed or concealed, this test should precede the work of closing in. The piping must stand a pressure of air at least equal to that exerted by a column of mercury 6 in. high for a period of 10 min without showing any drop in pressure. In the case of small exposed additions or extensions to existing piping, a "turn-on test" supplemented by a careful inspection will generally serve to establish whether the piping is gastight. Following the air test and before permanently turning gas on, it always is necessary to apply a "turn-on test" carefully, as prescribed under Mandatory Requirements in the AGA publication. (See also "Testing Services" on page 1183 and "Testing of New Construction" on page 1197.)

Design Details.—Pipe should not be run on outside or vestibule walls. Where this is unavoidable, it should be protected against sudden changes in temperature. A drip from which liquid condensate can be removed should be provided at any point in the line where condensate would collect. Piping should be pitched back to the meter, *viz.*, horizontal pipes should drain to the riser, and from the riser to the meter. Where condensation in house piping is excessive, a drip should

be provided at the outlet of the meter. This drip should be installed so as to constitute a trap wherein an accumulation of condensate will stop the flow of gas before condensate will run back into the meter. Drips should not be located where the condensate is likely to freeze. In order to avoid sags and to ensure adequate support, pipes should be hung or fastened with pipe hooks, bands, or hangers at proper intervals.

A tee with the bottom outlet plugged or capped should be used instead of an elbow at the bottom of all risers and drops, except ceiling or lighting-fixture drops. Concealed piping should be located in hollow rather than in solid partitions and should not be in contact with corrosive materials. In general, piping should be installed so as to be accessible with the least damage to the building structure. Where it is necessary to notch beams or joists, piping should not be more than 24 in. from the wall or other supports.

Pipe should not be installed in chimneys, flues, or ventilating ducts and shafts. Where there are overhanging kitchens or other rooms built beyond foundation walls in which gas appliances are installed, the supply piping should not be run in the open where it will be exposed to extreme changes in temperature. In all such cases the piping should be brought up inside the building proper and run around the sides of the rooms in the most practicable manner. If pipe is to be run from one building to another, it should be placed underground and the size should be the next larger than that specified in Table I (see page 1170) but in no case less than 1 in.

All branch lines should be taken from the top or sides of running lines and not from the bottom. Outlets should extend 1 in. through finished ceilings and walls, and either the outlet fitting or the pipe should be securely fastened to the building structure. Appliance connections to concealed piping should be made with ground-joint unions and the nut punched to prevent loosening by vibration. Swing joints made by combinations of fittings should not be used in concealed piping. Reducing fittings are recommended instead of bushings. Every gas cock or valve should be readily accessible for operation or repair. In a building supplied by a master valve, an individual control valve should be provided for each separate house line.

Bending of gas pipe should not be done except in special cases, and then with extreme care to see that the seam in the pipe has not been opened or weakened or the pipe otherwise damaged. White lead or other pipe joint compounds or dope is to be used sparingly and applied to the male threads only. With liquefied petroleum gases (propane, butane, or butane-air gases), the compounds used in making up joints should be resistant to the action of such gases.

Meters and Regulators.—Piping systems that are supplied by separate meters should not be interconnected. A pressure regulator or governor requiring access to the atmosphere for successful operation and controlling the gas supply to a building, should be equipped with a vent pipe leading to the outer air. Means should be provided to prevent water from entering this pipe, and also to prevent stoppage by insects or foreign matter.

Good practice requires that house piping be brought to a meter location which is readily accessible for examination, reading, or replacement, and where escaping gas will not collect in a confined space. A meter should not be located where it will be exposed to mechanical damage or excessive corrosion.

Conversion Burners.—American Standard Requirements for Installation of Conversion Burners in House Heating and Water

Heating Appliances, ASA Z21.8,[1] sponsored by the AGA, contains piping requirements similar to the foregoing which were abstracted from AGA Requirements and Recommended Practice for House Piping and Appliance Installation.

Power and Heating Boilers.—Attention also is called to the existence of American Standard Requirements for Installation of Gas Burning Equipment in Power Boilers, ASA Z21.33.[1] According to this standard a power boiler shall be considered as one furnishing steam or hot water for either heating or power purposes, the installation of which is not covered by American Standard Requirements for Installation of Conversion Burners in House Heating and Water Heating Appliances, ASA Z21.8.[1] Neither standard applies to boilers designed for gas fuel only where the necessary burners and accompanying accessories are installed at the factory and shipped as an integral part of the boiler.

In addition to furnishing general installation instructions for gas burners, regulators, and controls, ASA Z21.33 provides rules for installing gas piping and meters much like those in the AGA Requirements and Recommended Practice for House Piping and Appliance Installation, abstracted on pages 1168 to 1173. The following additional requirements are noted:

A manually operated main shutoff valve of the lubricated plug type and with handle permanently attached shall be installed at each boiler to shut off its entire gas supply except pilots in cases of emergency. It shall be readily accessible and shall clearly indicate the "on" and "off" positions or direction of rotation to open or close. It is recommended that provision be made for shutting off the gas supply to boiler or boilers by a valve or equivalent means immediately outside of the boiler room.

Each burner or group of vertical burners shall be equipped with a plug valve with stops at the open and closed positions. It shall be clearly marked to indicate the open and closed positions and the handle shall be permanently attached. Where burner valves operate at pressures above ½ psi, the use of lubricated plug valves is recommended. To avoid leakage of gas when the boiler is shut down, it is recommended either that a double valve and bleeder system be installed, or that a blind flange be provided to permit disconnecting and blanking off the line.

Boilers should be supplied by a gas line direct from the meter, independent from other plant uses. Piping shall be of a size and so installed as to provide a supply of gas sufficient to meet the maximum demand without undue loss of pressure between the point of delivery and the burners. In deciding upon the maximum gas consumption to be provided for, adequate allowance shall be made for any probable change in the specific gravity or heating value of the gas

[1] For complete information, reference may be made to the American Standard (see note, p. 1167).

supply. It is suggested that the capacities of different sizes and lengths of pipe may be computed by the Spitzglass formulas to be found on page 184.

Soldered pipe joints are not permitted in gas lines to power boilers.

Where high-pressure air is premixed with gas, a back-pressure regulator or other effective means shall be provided to prevent air from passing back into the gas piping.

After the piping and meter have been checked, all piping receiving gas through the meter shall be fully purged at the end of the pipe line to the outside of the building. If it is not practicable to purge to the outside air, other means of safe disposal shall be followed. It is recommended that, where possible, inert gases be used, or that purging be done through a water seal.

SERVICE PIPES

The service pipe that connects the street main with the meter in the customer's premises usually is installed by the gas company. Pipe diameters range from a minimum of $\frac{3}{4}$ in. nominal size to whatever is needed to supply the expected maximum demand, either present or future. Black steel pipe with malleable-iron fittings still is the common construction, although thin-walled copper tubing is being used to an increasing extent in soils where its superior corrosion-resisting properties offset higher first cost. Since the cost of the piping alone is only a fraction of the total cost of installing a service, there is an advantage in using more expensive materials if they will last longer and thus avoid too frequent renewals.

Corrosion Problems.—According to Dr. Scott Ewing,[1] services are the most vulnerable to corrosion and mechanical damage of any part of the distribution system owing to their small size and relatively thin walls. Furthermore, the corrosive conditions to which they are exposed are apt to be more severe than those for street mains. Points where services pass under street or house gutters, or where they enter a basement wall or rise out of the ground under a building without a basement, are particularly vulnerable to attack. Hence the need for considering the behavior of different service-pipe materials in various soils so as to select the material best suited to the particular soil condition encountered or to provide protective coatings, if and as required.

Gas companies report that cinders used for surfacing streets, driveways, or parking lots, or used as a base under cement side-

[1] See "Soil Corrosion and Pipe Line Protection," by Dr. Scott Ewing. Copies are obtainable from the American Gas Association, 420 Lexington Ave., New York 17, N.Y. While connected with the U.S. Bureau of Standards as AGA Research Associate, Dr. Ewing made an extended investigation of the tendency of different pipe materials to corrode in a variety of soils (see p. 1258).

walks, are the most frequent cause of corrosion. Sulphur in the cinders is dissolved by rain water and eventually finds its way downward to the pipe. Under these conditions bare steel pipe may not last more than 1 to 5 years, whereas under favorable conditions it is expected to last 30 years or longer. Copper pipe or tubing is not immune to corrosion when exposed to cinders or when laid in salt, marshy soil and, like steel pipe, should be given a protective coating when laid in such soil. Packing sand or clay around pipe buried in corrosive soil will help to prolong the life of either steel or copper services, whether bare or coated.

Although protective coatings cannot be said to give complete protection from corrosion in all soils, they do tend to slow up the process. For instance, one gas company reports that in certain localities on its system soil conditions are so corrosive that bare steel pipe lasts only 3 to 4 years, whereas coated lines laid in these localities are still in good condition after many years. Bare pipe often fails at frequent intervals, say about once every 10 ft, whereas coated pipe may not fail oftener than about once every 100 ft. Under the latter condition, repairs in the field are economical and gas-company engineers feel that covered steel pipe may last indefinitely if occasional repairs are made. The value of pipe coating is dependent upon materials used and care of application. A poor material, or a good one carelessly applied, may cause accelerated local corrosive action. This may happen also if the coating is damaged while handling.

The extra cost of galvanizing either pipe or fittings is considered unwarranted for gas service for the following reasons: (1) No protection is needed for the pipe interior. (2) The exterior of piping aboveground can be kept painted at less expense than galvanizing. (3) Galvanizing is considered ineffectual for resisting corrosion with buried pipes and the only useful purpose it serves is to seal pinholes that otherwise might leak. The protective coatings ordinarily considered for buried gas pipes consist of one or more layers of wax, coal tar, or bitumastic enamel wrapped on the outside with burlap or heavy kraft paper to protect against damage in handling. Where soils are known to be unusually corrosive, some companies have found it desirable to encase the entire underground structures in cement or fine concrete. The nature of such coatings is described in more detail in Chap. XVII on Corrosion. External protective coatings may be applied at a commercial treatment plant specializing in such work or in the shop of the local gas com-

pany. Special equipment is available for coating large-sized pipes for transmission lines or distribution systems on the job.

Steel Construction.—The joints in steel service pipes may be either threaded or made up with plain ends and compression couplings. Despite the low pressures obtaining in gas services, Schedule 40 (standard weight) pipe made to ASA B36.10[1] dimensions (see page 361) is used in preference to lighter weights in sizes up to 6 in. so as to have sufficient metal thickness for mechanical strength and for resisting corrosion. Although this has been true in the past partly because of the weakening effect of threading the pipe ends, Schedule 40 pipe still is specified where plain ends are used underground with either compression couplings or welded joints. Until more adequate protective coatings are developed, fully this much metal seems required for small steel pipes buried in the ground in order to withstand corrosion for a reasonable length of time. Couplings for threaded pipe used underground should be recessed to protect the threads.

Malleable-iron screwed fittings for gas services should conform to the American Standard for 150-lb. working pressure ASA B16c[1] (see pages 573 to 580) which are available in sizes up to 6 in. For the larger sizes, however, *welded construction* or *compression couplings* are to be preferred. Where welding is done, it should conform to the requirements of the American Standard Code for Pressure Piping, ASA B31, and to any existing local regulations. Oxyacetylene welding is used extensively on account of the portability of the equipment for field work. Malleable-iron compression couplings and fittings are available for pipe sizes from ¾ to 2 in., inclusive (see Fig. 2), which take up with a threaded packing nut and are equal in strength to the 150-lb malleable-iron screwed fittings. For larger sizes, bolted compression couplings like those illustrated in Fig. 7 on page 1193 should be used.

Copper Construction.—In order to hold down costs with an expensive material, thin-walled copper tubing, usually not lighter than type K, however, is used instead of copper or brass pipe of "iron pipe size." Although use of Class M copper tubing is permitted under the ASA Code for Pressure Piping, Class K is considered preferable for underground services in order to have a sufficiently thick wall for mechanical strength and resisting corrosion, combined with high ductility to avoid rupture from settle-

[1] Copies of American Standards can be obtained from the American Standards Association, 29 West 39th St., New York 18, N.Y.

ment. Connections may be made with compression couplings or soldered-joint fittings.

Brass compression couplings and fittings suitable for use with copper tubing are available. Their general appearance resembles the malleable-iron couplings and fittings illustrated in Fig. 2. Malleable-iron compression couplings intended for steel pipe can be used with copper tubing by inserting a rubber adapter bushing which serves as an electrical insulator against electrolysis and at the same time fills up the gap in the packing gland to the dimensions of steel pipe. Soldered-joint fittings, where used, should conform to American Standard ASA A40.3 (see pages 589 to 592).

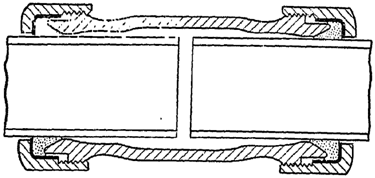


FIG. 2.—Malleable-iron compression coupling with threaded packing nuts for general use on underground piping—gas, water, oil, or other services. (*Courtesy of Dresser Manufacturing Company.*)

For further information on the use of copper pipe and brass fittings in gas service, reference may be made to technical papers[1] and to manufacturers' catalogues.

Steel vs. Copper.—Bare steel pipe should be used for buried gas lines only where soil conditions are known to be noncorrosive, or for temporary work where relatively short life is all that is expected. In corrosive soil coated steel pipe has much to offer in competition with copper tubing in the following respects: (1) With gas lines, corrosion is primarily external as distinguished from underground water lines where corrosion is both external and internal. (2) Bare steel pipe can be shoved with a pipe-pushing machine between pits dug a considerable distance apart (see pages 1184 to 1186) or coated pipe can be inserted in a hole washed or bored between pits, more readily than is the case with copper tubing. (3) The material

[1] (*a*) "Use of Copper Pipe for Gas Services, Drips, etc.," by H. L. Gaidry, *A.G.A. Proc.* 1936, pp. 601–602.

(*b*) "Developments in the Use of Copper Tubing for Gas Service," by E. A. Munyan, (Manager Gas Dept., Union Gas and Electric Co., Cincinnati, Ohio) *Gas Age-Record*, Vol. 75, pp. 43–48, Jan. 19, 1935.

cost of copper tubing with brass fittings is 1½ to 2 times that of coated steel pipe with malleable-iron fittings.

On the other hand, several large gas companies renew corroded steel services by pulling a reduced size of copper tubing through the existing steel pipe without doing much excavating. One company installs Class *K* annealed copper tubing as follows: First a fish wire is pushed through the old service followed by a ¼-in. steel cable. To the end of the cable is attached a tool and steel brush for cleaning the inside surface of the pipe. Finally, the copper tubing is crimped over the end of a fitting attached to the cable and pulled through. Twenty-five to thirty minutes is needed to pull copper tubing and the gas service is out of commission for not more than 2 hr under normal conditions. Where copper is exposed at the ends of the service next to the main or stop valve, the copper is coated and wrapped with standard materials to minimize electrolysis. The expected life of copper tubing inside steel pipe is assumed to be at least as good as coated steel pipe. This gas company renews practically all services with copper tubing, and several other large companies are known to be following similar practice.

Copper pipe for new services can be installed without a complete excavating job by pushing, boring, or washing a hole between pits (see pages 1184 to 1186), inserting a steel pipe through which the copper tubing is pulled, and then removing the steel pipe.

Cast Iron.—The use of cast-iron pipe for services is usually confined to sizes larger than 2 in. and follows the practice described on pages 1192 to 1196 for distribution mains. Owing to the possibility of damage from settlement, cast-iron pipe should not be installed through a basement wall or foundation. The use of steel pipe is recommended here instead of cast iron.

Installation Details.—Local gas companies have their own details for installing services, and practice may vary between municipalities served by the same company owing to differences in local ordinances. Then, too, such various conditions are encountered that it is impracticable to try to cover them all. A typical sketch of a simple low-pressure installation using compression couplings with threaded packing nuts is furnished in Fig. 3 and serves to show a street tee and street ell at the main, an outside stopcock with curb box, an inside stopcock, and the meter. Piping should pitch back from the meter to the street main with a gradient of 1 in. in 10 ft.

Where the street main operates at a pressure above a few inches of water column, a regulating valve is needed ahead of the meter as shown in Fig. 4. Regulators have different capacities depending on the diameter of internal orifice provided in bodies having the same size of pipe connections and should be selected by reference to manufacturers' catalogues.

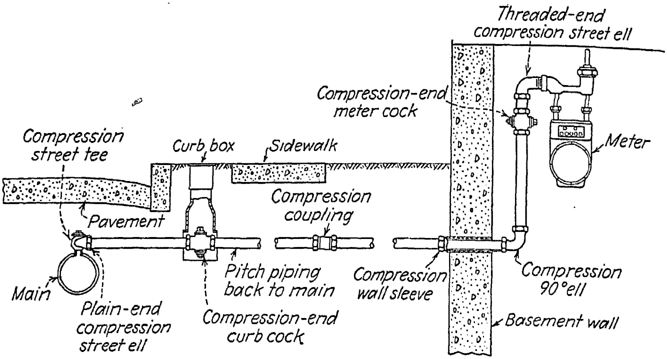


FIG. 3.—Sketch of steel gas service using compression couplings with threaded packing nuts.

One of the large city gas companies has the following specifications for stopcocks:

1. Outside cocks for services supplied from low- or intermediate-pressure mains shall be iron-body brass-plug service cocks, tested for at least 50 psi working pressure and having compression-type ends.

2. Outside cocks for services supplied from high-pressure mains (suburban system transmission or city system 40 psi mains) shall be lubricated plug cocks, tested for at least 40 psi working pressure and having compression-type ends, either with or without bolts.

3. All outside cocks shall have heads of standard dimensions specified. Service boxes shall be standard three-piece cast-iron boxes of the 3 in. or 6 in. size, as specified for each location.

4. Inside cocks shall conform to the requirements for outside cocks except that they shall have standard screwed ends.

5. Bolts, flanged ends, and bodies of all cocks shall be factory dipped with a specified corrosion-resistant coating.

In connection with outside stopcocks it should be noted that practice varies as to whether they are required or furnished on any and all small-size services. Whereas it is impracticable to assign any hard and fast rule, it would seem good practice to provide an

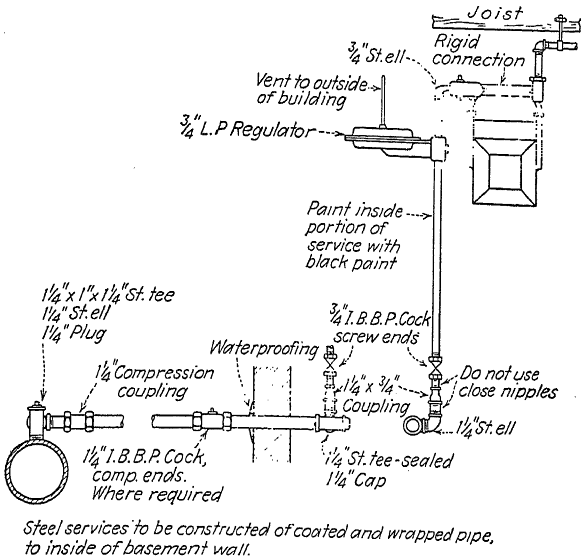


FIG. 4.—Gas-meter installation with pressure regulator.

outside stop on all services over 1½ in. in size. Typical lubricated curb cocks and meter cocks are shown in Fig. 5.

Connections to Mains.—One large city gas company has the following practice for making service connections to street mains. Connections to low-pressure steel mains, 2 to 8 in. in size, are made with service clamps using duprene rubber gaskets, except that Thread-o-lets (threaded welding pads) are used on jobs where a welder is available. Malleable tees are cut into mains smaller than 2 in. When making service connections by means of service clamps or Thread-o-lets, holes are drilled into the mains after having installed the clamps or Thread-o-lets. Thread-o-lets instead of clamps are used on the 40 psi high-pressure suburban system and on the 10 to 12 psi city transmission mains. Steel mains above 8 in. in size are tapped in the same manner as cast-iron

mains for ¾- or 1-in. services, and welded connections or clamps are used for larger sizes.

Size permitting, cast-iron mains are tapped for street tees; otherwise split sleeves are applied to large mains, or tees cut into the

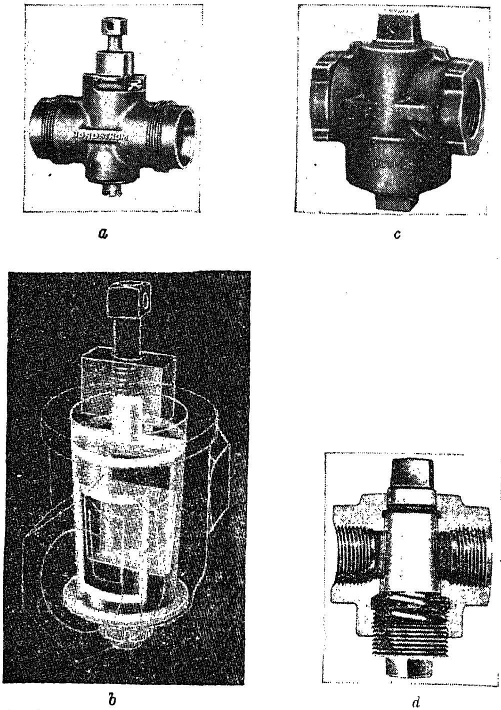


FIG. 5.—Gas-service plug cocks, lubricated type: (*a*) external view of curb cock with ends for compression couplings; (*b*) transparent view of screwed-end curb cock with lubricating ducts emphasized; (*c*) external view of screwed-end meter cock; (*d*) sectional view of screwed-end meter cock.

line for small mains. According to "The American Gas Handbook"[1] the maximum size of service tap that should be made in cast-iron mains is as shown in Table IV.

[1] Published by the American Gas Journal, Inc., 53 Park Place, New York, N.Y.

TABLE IV.—MAXIMUM SIZE OF SERVICE TAPS IN CAST-IRON MAINS

Size of main, in.	24	30
Maximum size of tap, in.	[illegible]	[illegible]

When not to be used for a service, a 1¼-in. hole may be tapped in a 4-in. main and a 1-in. hole in a 3-in. main.

After tapping, the service connection is made by screwing into the main, clamp, Thread-o-let, or split sleeve, a street tee into which a gas service stopper (expandable rubber plug) is inserted to stop the flow of gas while the service is being constructed. The use of a street ell, in combination with a street tee, provides a swing joint that will permit the connection to be made at any desired angle without strain. A mechanical coupling near the connection at the main affords additional flexibility and helps to relieve external strain that may cause leaks and breaks in underground pipes.

Protection against Frost.—The service pipe from the street main to the meter should be graded up from the main with a rise of 1 in. in 10 ft so as to drain back to the main. Each driven pipe installed with a pushing machine (see pages 1184 to 1186) should be tested for traps by pouring a measured quantity of water through it. If the first drive is unsuccessful, a second attempt should be made. If the second attempt is unsuccessful, a third attempt should be made only if there is a reasonable assurance that it can be done successfully. Lacking this assurance, boring should be resorted to.

The depth of bury should be at least 2 ft 6 in. at the shallowest point, as a protection against both frost and mechanical injury. Where there is liable to be trouble from frost, it is well to use no service of less than ¾-in. size no matter how short it may be. In extremely cold climates this often is increased to 1 in. even for a single appliance. Buried services smaller than ¾ in. are not recommended under any circumstances. Some gas companies make it a practice to install no steel service pipes for low pressure smaller than 1½ in., no matter how small the connected load. This provides for future load growth and at the same time makes it possible to renew the line by pulling through a copper pipe when the steel pipe corrodes.

Testing Services.—Service pipes should be tested according to the requirements of Sec. 223 (see page 1214) of the American

Standard Code for Pressure Piping before the trenches are backfilled. In some cases, however, it has been the practice to give services only the "turn-on" test at the operating gas pressure and inspect all accessible joints for leaks with soap suds. Where joints are inaccessible, that portion of the line should be given a static pressure test of at least 10 psi gauge for 10 min without showing any drop in pressure. Inaccessible joints in high-pressure services should be tested with a static pressure at least equal to the maximum pressure to be carried by the system.

Pipe Pushing and Boring.—In the installation of gas service branches and in some other kinds of underground piping, it frequently is advantageous to push the pipe through the soil rather than to dig a trench. The chief advantages of pipe pushing are the saving in cost, the avoidance of traffic obstructions, and the reduction of damage to lawns and pavements. The distance through which a pipe can be pushed depends upon the method used and the character of the soil, which must be free from large stones and preferably of a sandy or loamy type. The accuracy of direction also is dependent upon the condition of the soil and the obstructions encountered. Coated and wrapped pipe should not be pushed through the soil directly owing to the likelihood of damaging the covering by abrasion. It can be inserted safely by hand, however, in oversized holes that have been washed or driven with bare pipe which is withdrawn before the coated pipe is installed.

Jacking.—The most common method of pushing pipe is to use a mechanical or hydraulic jack. The mechanical jack is preferred by some gas companies because of its low cost and light weight. It is capable, under reasonably favorable soil conditions, of pushing ¾- to 2-in. pipe. The hydraulic jack, which is preferred by others, requires less effort to operate but is somewhat more costly and less easily portable. Some jacks can be reversed so as to withdraw a pilot pipe which is used for punching a hole through which a copper or coated steel pipe can be inserted later. A hydraulic jack set up for operation is shown in Fig. 6*a*. A main trench 2 ft wide and somewhat longer than the steel base of the jack is dug to the proper depth. A cross trench to take the blocking timber against which the jack thrusts also is required. The length of pipe that can be pushed without a joint depends, of course, upon the length of the main trench. For the tip of the pipe, a coupling and pipe plug are used in the smaller sizes and a reducer and plug for the larger sizes (over 1½ in.). There is no advantage in using

a sharper point since it is more easily deflected by stones. As to distance, a 4-in. pipe can be pushed 50 to 75 ft, in fairly good soil, with reasonable accuracy. Two-inch pipe can be pushed 200 ft or more, but for any such distances it is advisable to check the direction at intervals by digging cross trenches.

Washing or Jetting.—When fine gravel or sand is encountered, it is advantageous to use water, at city hydrant pressures, to wash a hole ahead of the pipe. In a jetting operation the work should be carefully timed and the pipe advanced in step so as to avoid washing out too large a hole. Usually, when pushing a pipe of ¾- to 2-in. size with the aid of a jet in suitable soil, the pipe can

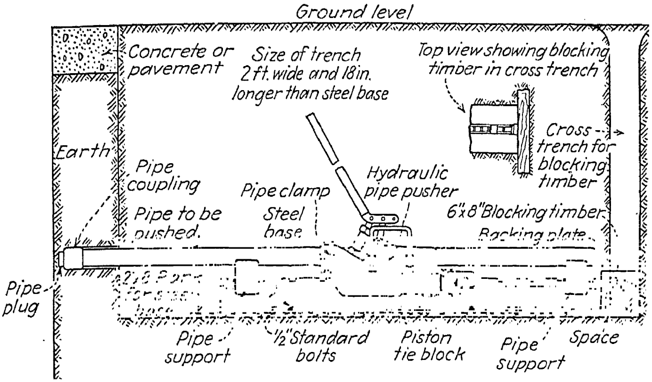


FIG. 6*a*.—Hydraulic pipe pusher set up for operation in a trench. (*Courtesy of the Greenlee Tool Company, Rockford, Ill.*)

be advanced by one man without the use of any mechanical device. With 4- to 6-in. pipe a jack is used or, under some conditions, a cable attached to the rear end of the pipe and pulled by a tractor is satisfactory. After once started the washing or jetting operation must be continuous or sand will set around the pipe, binding it to such a degree that it can be moved only with extreme difficulty.

Boring.—For soil conditions where pushing the pipe or jetting is impracticable, and for larger pipes, a combined mechanical and hydraulic auger such as that shown in Fig. 6*b* is often used to bore a hole. The boring bar is then withdrawn and the pipe inserted in the hole. A machine such as this can bore holes from 2½ up to [illegible] in. in diameter. It uses compressed air as the motive power

and water at high pressure for the jet that issues from the tip of the boring bit. The carriage is advanced along the spur track by means of a lever and ratchet. The accuracy is quite good if the machine is properly aligned, although it depends somewhat upon the skill of the operator. It is preferable to bore slightly uphill.

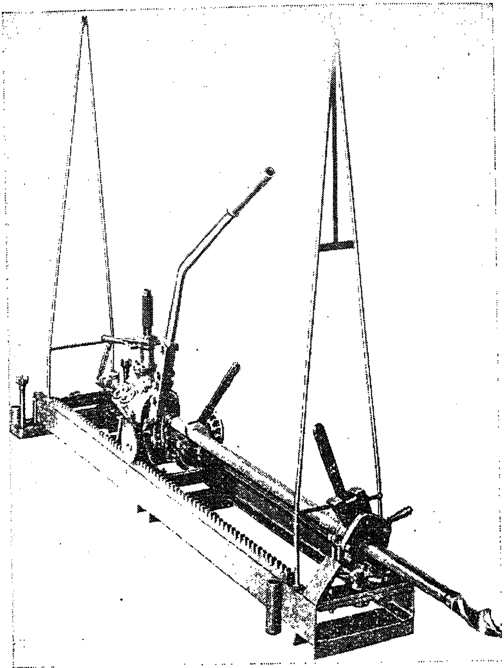


FIG. 6b.—Hydraulic boring machine. (*Courtesy of the Hydrauger Corporation, Ltd., San Francisco, Calif.*)

Pushing Large Pipes.—For large pipes, over 15 in. in diameter, heavy-duty pushing jacks are available. The end of the pipe is left open and the soil removed from the inside as in tunneling work. Such operations naturally require considerable experience.

Carrying Capacity of Service Pipes.—The carrying capacity of gas lines in general is presented on pages 165 to 200. There are certain aspects of sizing service pipes that warrant specific treatment here. Low-pressure services for single dwellings or commercial

buildings should be sized with respect to the connected load, allowing a reasonable pressure drop that will not disturb the stability of combustion conditions. Where gas is distributed at a pressure approximating 6 in. of water column, a satisfactory pressure drop from the main to the meter would be in the order of 0.3 in. of water at maximum flow. The carrying capacities of different lengths and diameters of standard-weight steel pipe corresponding to a pressure drop of 0.3 in. of water column are given in Table I. Where higher pressure drops are permissible, reference may be made to the *Authors' Note* under Table I or to formulas in the section on Flow of Gas and Air in Pipes, pages 165 to 194.

If several customers are supplied through the same service, as in an apartment house, a certain amount of diversity in use can be expected so that it should not be necessary to supply the entire connected load at any time. Table V, from the 1938 American Gas Catalogue and Handbook, gives the size of low-pressure service that has been found by experience to be usually satisfactory for apartment houses, assuming gas of 0.6 specific gravity at 6 in. of water column, and 0.3 in. water pressure drop.

TABLE V.—SIZE OF LOW-PRESSURE GAS SERVICE USUALLY NECESSARY FOR APARTMENT HOUSES

No. of apartments	Nominal size of pipe[1] for given length of service, feet																	
	20	40	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200
4	2″	2″	2″	2″	2″	2″	2″	2″	2″	2″	2″	2″	2″	2″	2″	2″	2″	2″
8	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2
10	2	2	2	2	2	2	2	2	2	2	2	2	2½	2½	2½	2½	2½	2½
12	2	2	2	2	2	2	2	2	2½	2	2½	2	2	2	2½	2	2	2½
14	2	2	2	2	2	2½	2	2	2½	2	2	2	2½	2	2½	2	2	2½
16	2	2	2	2	2½	2	2	2½	2½	2	2	2	2½	2	2	2	2½	2½
20	2	2½	2	2	2½	2	2	2	2½	2	2	2	3	3	3	3	3	3
24	2	2	2	2½	2½	2	2½	2	3	3	3	3	3	3	3	3	3	3
30	2	2½	2	2	2½	3	3	3	3	3	3	3	3	3	3	3	3	3
40	2	2½	3	3	3	3	3	3	4	4	4	4	4	4	4	4	4	4
48	3	3	3	3	3	4	4	4	4	4	4	4	4	4	4	4	4	4

[1] For Schedule 40 (standard-weight) steel pipe.

With *intermediate* and *high-pressure* distribution systems it is customary to make all new service pipes at least ¾-in. nominal size, although ½-in. copper tubing sometimes is pulled through existing ¾-in. steel services which have failed through corrosion.

Table VI-A.—Intermediate-pressure Gas-carrying Capacities[1] in Cubic Feet per Hour[2] of Pipes[3] of Different Diameters and Lengths

(Carrying capacity in cubic feet per hour measured at standard conditions[2])

Length, feet	Nominal pipe size, inches									
	¾	1	1¼	1½	2	2½	3	4	6	8
Capacities based on 2 psi gauge[4] initial pressure and 1 psi drop through line[5]										
50	1,170	2,270	4,620	6,950	13,550	21,850	38,800	80,300	240,000	499,000
100	825	1,570	3,280	4,930	9,600	15,500	27,450	56,900	169,100	352,500
150	675	1,285	2,660	4,020	7,830	12,600	22,400	46,400	137,000	288,000
200	585	1,110	2,320	3,490	6,790	10,920	19,450	40,200	119,800	249,500
300	475	907	1,890	2,850	5,530	8,910	15,900	32,800	97,600	204,000
400	413	786	1,640	2,470	4,800	7,720	13,780	28,400	84,700	176,300
500	370	704	1,460	2,210	4,280	6,900	12,300	25,400	75,600	157,900
1,000	262	497	1,035	1,560	3,040	4,880	8,700	18,000	53,600	111,300
Capacities based on 3 psi gauge[4] initial pressure and 2 psi drop through line[5]										
50	1,680	3,260	6,640	10,000	19,450	31,400	55,800	115,300	344,000	716,000
100	1,190	2,260	4,710	7,080	13,800	22,200	39,450	81,700	243,000	506,000
150	969	1,850	3,830	5,780	11,240	18,130	32,200	66,600	198,200	414,000
200	840	1,600	3,330	5,010	9,740	15,700	28,000	57,700	172,100	358,000
300	685	1,303	2,710	4,100	7,940	12,810	22,800	47,100	140,300	292,500
400	594	1,130	2,360	3,540	6,890	11,100	19,780	40,750	121,900	253,000
500	530	1,010	2,100	3,170	6,150	9,900	17,670	36,400	108,800	226,500
1,000	374	713	1,485	2,240	4,350	6,990	12,480	25,000	76,800	160,000
Capacities based on 5 psi gauge[4] initial pressure and 3 psi drop through line[5]										
50	2,145	4,170	6,490	12,780	24,900	40,100	71,300	147,500	440,000	915,000
100	1,520	2,880	6,020	9,060	17,640	28,400	50,500	104,500	311,000	648,000
150	1,240	2,360	4,900	7,400	14,390	23,200	41,200	85,200	253,500	529,000
200	1,074	2,040	4,250	6,410	12,460	20,100	35,700	73,800	220,000	458,000
300	875	1,665	3,470	5,240	10,150	16,380	29,200	60,200	179,400	374,000
400	760	1,445	3,010	4,540	8,810	14,200	25,300	52,100	155,900	324,000
500	679	1,290	2,680	4,050	7,870	12,690	22,600	46,600	139,000	290,000
1,000	479	911	1,892	2,860	5,560	8,950	15,970	32,950	98,100	205,000
Capacities based on 10 psi gauge[4] initial pressure and 5 psi drop through line[5]										
50	3,060	5,950	12,100	18,220	35,400	57,200	101,600	210,500	627,000	1,304,000
100	2,165	4,110	8,580	12,900	25,150	40,500	71,900	149,000	443,000	923,000
150	1,770	3,370	6,980	10,530	20,500	33,100	,700	121,300	362,000	754,000
200	1,530	2,910	6,060	,130	17,780	28,600	900	105,100	313,500	652,000
300	1,245	2,380	4,950	,460	14,470	23,350	41,600	85,700	256,000	533,000
400	1,080	2,060	4,290	,460	12,560	20,200	36,000	74,200	222,000	462,000
500	968	1,845	3,830	,780	11,220	18,100	32,200	66,400	198,100	413,000
1,000	684	1,304	2,710	,080	7,830	12,780	22,800	46,900	140,000	292,000

Table VI-*A*.—(*Concluded*)

Length, feet	Nominal pipe size, inches										
		1	1¼		2	2½					
Capacities based on 15 psi gauge[4] initial pressure and 7.5 psi drop through line[5]											
50	4,050	7,870	16,020	24,100	47,000	75,700	134,600	279,000	830,000	1,727,000	
100	2,865	5,450	11,350	17,100	33,300	53,600	95,200	197,300	586,000	1,222,000	
150	2,340	4,460	9,240	13,940	27,150	43,800	77,600	160,700	479,000	996,000	
200	2,030	3,860	8,020	12,100	23,500	37,900	67,400	139,200	415,000	864,000	
300	1,650	3,145	6,540	9,880	19,170	30,900	55,000	113,400	339,000	705,000	
400	1,430	2,730	5,680	8,550	16,620	26,800	47,700	98,400	294,000	611,000	
500	1,280	2,440	5,070	7,650	14,860	23,900	42,600	88,000	262,000	547,000	
1,000	904	1,585	3,580	5,400	10,490	16,900	30,100	62,100	185,300	387,000	
Capacities based on 20 psi gauge[4] initial pressure and 10 psi drop through line[5]											
50	5,000	9,720	19,780	29,800	58,000	93,500	166,200	344,000	025,000	2,135,000	
100	3,540	6,720	14,010	21,100	41,100	66,300	117,500	243,700	725,000	,510,000	
150	2,890	5,500	11,400	[illegible],200	33,550	54,100	95,900	198,200	591,000	,231,000	
200	2,510	4,760	9,910	14,930	29,050	46,800	83,200	172,000	513,000	,068,000	
300	2,040	3,885	[illegible]	12,200	23,650	38,200	68,000	140,200	418,500	871,000	
400	1,768	3,370	7,010	10,560	20,500	33,100	59,000	121,500	363,000	755,000	
500	1,5[illegible]0	3,010	6,260	9,450	18,350	29,550	52,600	108,500	324,000	676,000	
000	1,115	2,130	4,430	6,[illegible]	12,960	20,880	37,200	76,600	229,000	478,000	

[1] Computed by Weymouth formula (see page 187) for a specific gravity of 0.7 and a flow temperature of 60 F. To convert the figures in this table to any other specific gravity and/or flow temperature, multiply the values given by $\sqrt{0.7/s}$, or by $\sqrt{520/T}$, as the case may be, or combine them into one adjustment which becomes $19.08/\sqrt{sT}$.

[2] Measured under standard conditions of 14.7 psi abs at 60 F.

[3] Schedule 40 (standard-weight) pipe. Approximately equal flows will be obtained with Type K [illegible].

[4] For convenience in use, the values are grouped according to initial gauge pressure and line pressure drop, whereas the tabular values were computed from the initial and final absolute pressures, *viz.*, gauge [illegible] 14.7.

[5] See text and Table VI-B [illegible] to solve for other lengths and over-all pressure

Table VI-*B*.—Length Ratios for Use with Other Pressure Drops Than Those Shown in Table VI-*A*

Initial pressure, psi gauge	Pressure drop, psi												
	1	2	3	4	5	6	8	10	12	14	16	18	20
	Length ratio												
2	1.000	0.515											
3	1.940	1.000	0.686										
5	2.820	1.45	1.000	0.761	0.630								
10	4.58	2.32	1.60	1.23	1.000	0.853	0.672	0.564					
15	6.33	3.39	2.28	1.76	1.41	1.22	0.946	0.785	0.684	0.611			
20	8.61	4.40	2.97	2.28	1.63	1.56	1.21	1.000	0.859	0.764	0.694	0.641	0.601

For information on regulators and control valves for stepping down distribution pressures to what the customer can use, reference may be made to manufacturers' catalogues and to a series of articles on "Gas Pressure Controls," by Erick Larson.[1]

Where gas is distributed at intermediate pressure, a considerable friction loss can be allowed through the service pipe between the street main and the pressure regulator in the customer's premises. Carrying capacities of different diameters and lengths of services computed by the Weymouth formula (see page 186) to suit a variety of distribution pressures are furnished in Table VI-*A*. This table can be used also for determining the carrying capacity of intermediate-pressure mains where relatively large pressure drops are involved.

All sections of Table VI-*A* can be used for approximate solutions of problems involving lengths and over-all pressure drops other than those shown in the section headings by applying the proper length ratios read from Table VI-*B*. These ratios were computed according to the relations of the Weymouth formula given on page 187. The application of the length ratios is illustrated in the following examples which involve solving the same problem from different angles:

Example 1.—Given a 2-in. service pipe with an initial pressure of 3 psi gauge and a desired delivery pressure of 2 psi gauge, what flow can be expected through an equivalent length of 200 ft?

Solution.—It is first necessary to determine the length of 2-in. pipe which will have 2 psi pressure drop for the same rate of flow that would produce 1 psi pressure drop in 200 ft of 2-in. pipe. This is obtained by multiplying the given length of 200 ft by the length ratio read from Table VI-*B*, which is 1.94 for the assumed conditions. Thus the length corresponding to 2 psi pressure drop is $200 \times 1.94 = 388$ ft. Interpolating in Table VI-*A* for 388 ft, the corresponding flow is found to be 7,016 cu ft per hr.

Example 2.—A flow of 7,000 cu ft per hr is required through 200 ft of 2-in. service pipe. What will the discharge pressure be if the initial pressure is 3 psi gauge?

Solution.—Referring to Table VI-*A*, the length of 2-in. pipe corresponding to a flow of 7,000 cu ft per hr and 2 psi pressure drop is 389.4 ft by interpolation. From which the length ratio is $389.4/200 = 1.947$. Referring to Table VI-*B* the pressure drop for an initial pressure of 3 psi gauge and a length ratio of 1.940 is 1 psi. Hence the discharge pressure is $3 - 1 = 2$ psi gauge.

Example 3.—An initial pressure of 3 psi gauge is available for delivering gas through an existing 2-in. service pipe. A flow of approximately 7,000 cu ft per hr is desired at a discharge pressure of 2 psi gauge. How far can the point of delivery be from the service tap?

[1] See *Amer. Gas J.* throughout the year 1935, Vols. 142 and 143.

Solution.—Interpolating for 7,000 cu ft per hr in Table VI-*A*, the length could be 389.4 ft if the allowable pressure drop were 2 psi according to the basic conditions of that table. The allowable pressure drop is only 1 psi, however, for which the length ratio read from Table VI-*B* is 1.94. Hence the maximum permissible length is 389.4/1.94 = 200 ft.

DISTRIBUTION NETWORK OR STREET MAINS

Distribution system flow capacity usually is designed on the basis of a peak-hour load factor derived from the gas send-out rather than by attempting to make a summation of the rated connected load and applying a diversity factor to it. The gas send-out figures are readily available from plant records and the customers' monthly statements without having to contact the customer or make a field survey. On the other hand, design loads computed by the peak-hour factor method are not very flexible and cannot be used safely on small consumer groups.[1]

The peak-hour factor is derived by computing the ratio of the maximum hourly works send-out to the maximum monthly works send-out. Since the severity of winter seasons will vary from one year to the next, it is best to compute the average ratios over a 5-year period and apply an adequate safety factor. As an example, Hoff reported that the gas send-out of the Los Angeles Gas and Electric Corp., during the maximum hour, was $\frac{1}{310}$ of the maximum monthly send-out. The factor then was changed to $\frac{1}{240}$ to provide the necessary safety factor.

Network Flow.—If gas may flow from the source of supply to the point of demand by more than one path, as in the case of a network of street mains, it is obvious that the flow will divide between available routes so as to obtain the same over-all pressure drop by each. This problem is more complicated than that of the "Divided Circuit or Loop" discussed on pages 90 to 95, and illustrated by the example on pages 196 to 198, because not all the gas enters at one end of the system and leaves at the other. In fact, gas may be supplied to any one section of the network at two or more points and be fed out through branches or customers' services at many points.

[1] Abstracted from Distribution System Design, by Norman L. Hoff, in "Gas Engineers' Handbook," McGraw-Hill Book Company, Inc., New York, prepared by a committee of The Pacific Coast Gas Association, reviewed by a special committee of the American Gas Association, and endorsed by the directors of the latter association. See also "American Gas Practice," Vol. II, Distribution and Utilization of City Gas, by Prof. Jerome J. Morgan, published by Jerome J. Morgan, Maplewood, N.J., 1935. This book contains an extensive bibliography and will be found useful for reference.

A simple illustration is the case of a distribution main supplying one city block, which is fed from both ends of the block and has, say, 20 services taken off at intermediate points. Another simple illustration of a different condition that has to be met would be the same city block of main still supplying the same 20 services, but fed from one end only and passing gas out at the opposite end [illegible] of the network

When, as in the foregoing cases, not all of the gas which enters at one end of a main is discharged at the other end, computation of the carrying capacity of the main by any one of the usual end-to-end flow formulas (see pages 165 to 194) is impossible. Under these circumstances, it is recommended that the capacity of the main, or the quantity of gas discharged, as computed by one of the end-to-end flow formulas, be increased through multiplication by appropriate factors. Space does not permit setting forth here the various modifying factors required to effect such a solution. Those having occasion to solve problems of this sort can refer to the section Gas Distribution by Hoff in the "Gas Engineers' Handbook," or to Prof. Cross's solution[1] of the network problem in water distribution systems which it is said can be extended to gas flow (see also page 90).

Networks may be of every conceivable layout, moreover, so that no firm rules can be established for their calculation. As a means of getting started with such solutions, Hoff suggests assuming points of "no velocity" and making a trial-and-error calculation on this supposition. By point of "no velocity" is meant a point fed from both directions such, for instance, as the service existing somewhere in the aforesaid block of city main which was supplied from both ends. This procedure is continued until the points of "no velocity" are so placed that the pressure drops, calculated by any route from the point of supply to any point of "no velocity," are reasonably equal.

Network Construction.—The design, manufacture, installation, and tests of gas distribution networks or street mains within the corporate limits of cities or villages come within the scope of Division 1 of Sec. 2 on Gas and Air Piping Systems of the American Standard Code for Pressure Piping, ASA B31, which is abstracted on pages 1208 to 1215. Some further discussion of street-main

[1] "Analysis of Flow in Networks of Conduits or Conductors," by Prof. Hardy Cross, *Bull.* 286, November, 1936, Engineering Experiment Station, Univ. of Illinois.

construction seems called for beyond the bare essentials mentioned in the Code; hence the following exposition of current practice.

For many years the conventional construction used for low-pressure street mains was cast-iron bell-and-spigot pipe and fittings conforming to the AGA standard (see pages 441 to 443) and made up with lead or cement joints.[1] Plain-end steel or cast-iron pipe made up with compression sleeve couplings (Dayton or Dresser type) was used to some extent also (see Fig. 7) and still is.

More recently the trend has been to steel pipe with welded joints and to bell-and-spigot cast-iron construction made up with so-called "mechanical joints" owing to limitations of the older lead or

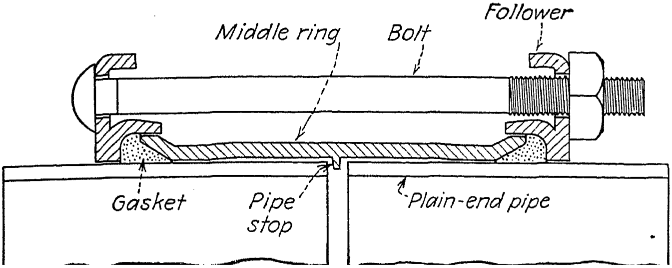


FIG. 7.—Compression sleeve (Dresser) coupling for use with plain-end cast-iron or steel pipe for gas, oil, or water mains. (*Courtesy of Dresser Manufacturing Company.*)

cement joints. Whereas lead, cement, or a combination of lead and cement joints held reasonably tight where saturated gas was distributed at low pressure, they were found to be unreliable at pressures above 10 in. of water column or with dry gas. In many cases, therefore, it has been necessary to repair leaky bell joints, for which purpose a bell-joint clamp (see Fig. 8) has been found satisfactory. This device, like the mechanical joint described in the next paragraph, depends for tightness on compressing a rubber or composition gasket. The American Gas Association has issued "Tentative AGA Requirements for Bell Joint Clamps,"[2] which present minimum standards for satisfactory

[1] For an historical article describing the development of all types of cast-iron pipe joints, reference may be made to "Paper on Cast Iron Pipe Joints," by Martin I. Mix in the report of the AGA Subcommittee on Pipe Joints, *AGA Proc.*, 1932, pp. 722–759.

[2] Copies can be obtained from the American Gas Association, 420 Lexington Ave., New York 17, N.Y.

performance and substantial and durable construction in gas distribution lines, with test requirements intended for demonstrating this.

It is said that a satisfactory bell-and-spigot joint for gas pressures up to 25 psi can be made with rubber rings held in place in the bell by square-braided jute and cement or lead rings. In this case the rubber rings are driven into place and held by the jute and cement. Lead can be used instead of cement, but in that case, the joint will be more expensive. The cement (or lead) acts as a stuffing-box gland to hold the rubber rings firmly inside the central portion of the bell. This effect is accomplished by painting

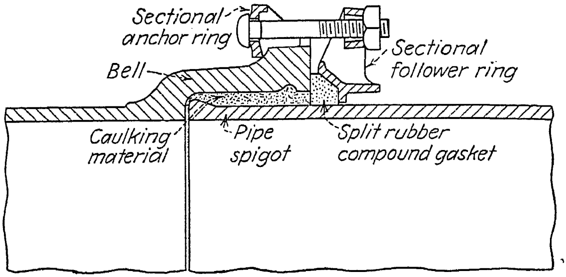


FIG. 8.—Leak clamp for repairing defective bell joints in gas and water mains and other services. (*Courtesy of Dresser Manufacturing Company.*)

the spigot so that the cement adheres only to the inside of the bell and leaves the spigot free to move in and out slightly.

In the absence of a national standard for either *mechanical joints* or *centrifugally cast* cast-iron pipe, manufacturers have their own design of joint and materials specification for this product. Manufacturers' standard dimensions for mechanical joints will be found on page 424. Where users wish to follow a recognized material specification for centrifugally cast pipe and fittings for either leaded or mechanical joints, reference may be made to Federal Specifications Bureau WW-P-421 (see page 443). A typical design of "mechanical" joint is shown in Fig. 9, regarding which the manufacturer has the following to say:

> This is centrifugally-cast cast-iron pipe equipped with a stuffing-box type of joint having a ring gasket and cast-iron gland which enables the pipe to meet the exacting requirements of high-pressure and superservice, particularly in gas and oil transmission and distribution work. It makes a bottle-tight joint under operating pressures in excess of 500 psi gas and 1,000 psi liquid. It allows for

liberal deflection and longitudinal expansion and contraction in the line without danger of leakage.

Rubber gaskets with plain tips, duck tips, or lead tips are said to be suitable for brine, compressed air, gas, oil, sewage, or water service for temperatures up to 175 F. Rubber-impregnated asbestos gaskets are offered for higher temperatures. Rubber gaskets with Thiokol-impregnated duck tips are available for use with petroleum products where desired. Bolts and nuts usually are of high-strength cast iron intended for resisting corrosion, and after installation should be given a protective coating of wax, asphalt, or bituminous products.

The American Gas Association has issued "Tentative AGA Requirements for Mechanical Joints for Cast-Iron Pipe Used for

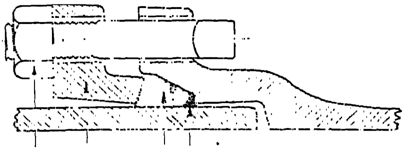


FIG. 9.—Typical mechanical joint used with bell-joint cast-iron pipe for gas, oil, sewer, and water service.

Gas Distribution"[1] which present minimum standards for satisfactory performance and durable construction, with test requirements intended for demonstrating same. The requirements are intended to be sufficiently general to permit originality and improvement of design and should not be construed as a measure of quality beyond compliance therewith.

Other forms of joints are used to some extent for underground cast-iron pipe in gas systems. Among these are the Victaulic coupling, the screw gland, and the Bellmaster joint, all illustrated in Fig. 10.

[1] Copies can be obtained from the American Gas Association, 420 Lexington Ave., New York 17, N.Y. For further explanation of mechanical joints reference may be made to *AGA Proc.*, 1932, p. 717, and to the following papers in the *AGA Proc.* for 1934:

(*a*) "Improvements to Mechanical Pipe Joints and the Prevention of Bolt and Nut Corrosion," by J. A. Perry, pp. 863–881 (well illustrated).

(*b*) "The Distribution of Gasket Pressures in Pipe Joints and Clamps," by George H. Pfefferle, pp. 882–896.

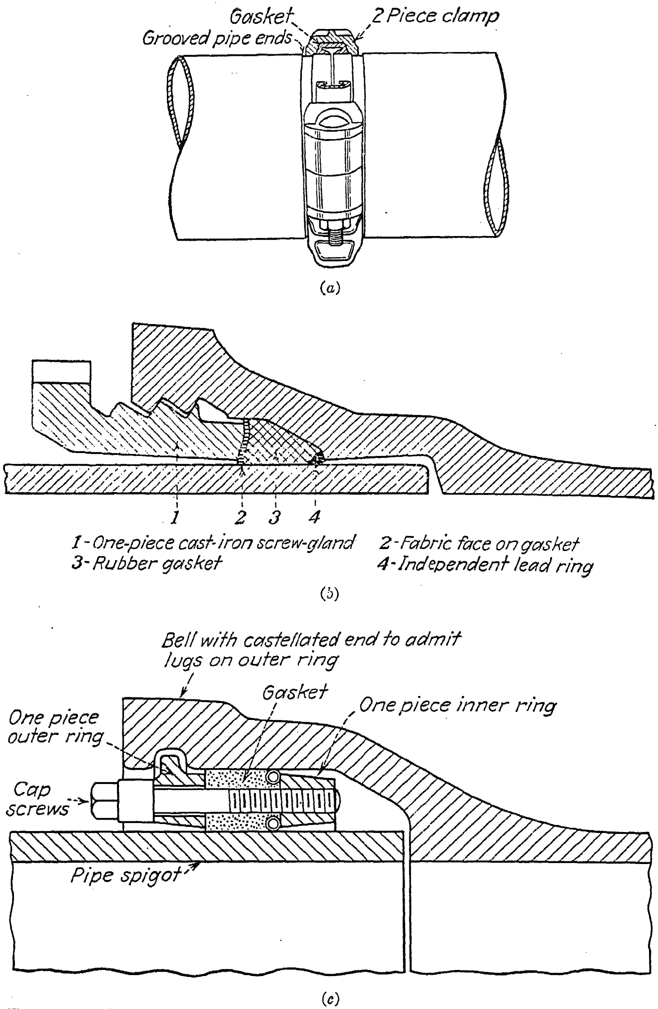


FIG. 10.—Miscellaneous pipe joints: (a) Victaulic coupling; (b) screw-gland joint; (c) Bellmaster joint.

Depth of Bury.—Either cast-iron or steel distribution pipes should be *buried* with at least 30 in. of cover over the top of the pipe in order to minimize traffic jar and lessen the range of temperature changes with ensuing expansion or contraction movement. Sufficient cover also helps to avoid contact with surface moisture, affords protection from mechanical damage at points not under pavement, and reduces the likelihood of damage from frost in cold climates. The strength of buried pipes in resisting earth loads and concentrated loads transmitted through the earth is discussed on pages 430 and 431. See pages 1063 to 1070 for computing Earth Loads on Pipe in Trenches.

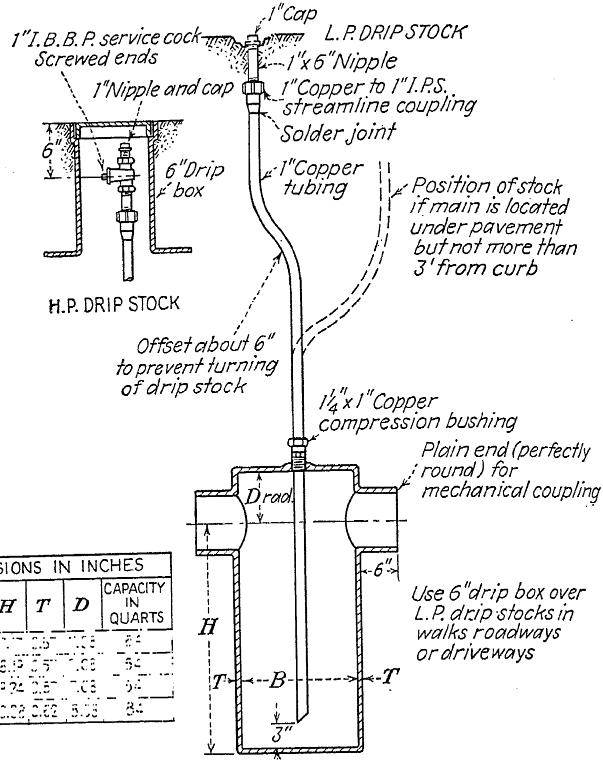
Drainage of condensation inside the pipe to collecting pockets from which it can be removed periodically is essential. The natural contour of the terrain often affords enough pitch to ensure drainage to low points where collecting pockets can be installed. A typical collecting pocket is shown in Fig. 11. If the natural slope is insufficient for good drainage, it may be necessary to vary the depth of excavation enough to give the desired pitch. Whereas a pitch of only 1 in. or so in 100 ft would be sufficient for drainage, especially where gas and condensate are flowing in the same direction, a pitch of 2 to 3 in. per 100 ft is advisable to guard against pockets due to minor irregularities in laying the line.

Testing of new construction is carried out by one large city gas company as follows: Small extensions in the low-pressure system need be tested only with gas pressure (about 6 in. of water column), but if an air compressor is available the low- and intermediate-pressure (operating at 4 psi) mains in the city system are tested with 10 psi air pressure. Large transmission lines and all mains in the suburban system (operating at 10 to 40 psi) are tested with 50 psi air pressure. At the same time the pressure test is being conducted, all joints are tested with soapsuds and any leaks detected are repaired. The air pressure test is accepted as sufficient for any joints that are inaccessible for applying the soapsuds test.

Repairing Leaks.—According to the "American Gas Handbook"[1] the problem presented by increased joint leakage, attendant upon the substitution of dry gas for wet gas, was recognized many years ago and various means were adopted for correcting this

[1] Published by the American Gas Journal, Inc., 53 Park Place, New York, N.Y. See 1942 ed., pp. 126–127. See also "Experience in Leak Proofing Bell and Spigot Joints," by G. E. Hitz, *AGA Proc.*, 1936, pp. 611–622.

condition. By a dry gas is meant one lacking much oil or water vapor. Gases inherently dry are (*a*) natural gas, (*b*) hydrocarbon gases such as propane and butane, (*c*) manufactured gas which



DIMENSIONS IN INCHES					
SIZE	*B*	*H*	*T*	*D*	CAPACITY IN QUARTS
4"	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
6"	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
8"	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
12"	[illegible]	[illegible]	0.62	[illegible]	[illegible]

Coat and wrap all drip stocks and drip connections, including both the copper and steel portions

FIG. 11.—Typical pocket for collecting condensation at low points in a gas-distributing system.

has been compressed for transmission at high or intermediate pressures and then expanded into a low-pressure distribution system, (*d*) mixed manufactured and natural gas.

Although the application of bell-joint clamps and gaskets at each bell-and-spigot joint stops the leaks satisfactorily (see page

1194), this method of repair is apt to be rather costly where it is necessary to excavate to get at the joints. Rehydration and oil fogging have been tried as a means of swelling up the packing in such joints after a change over to dry gas, but these measures are said to be inadequate. An antileak liquid is available, however, which has solvent properties for tar and gum and penetrates and climbs through the packing readily, swelling it approximately 45 per cent by volume and shutting off leakage. The material has been variously estimated by users to have a permanency of at least 10 to 12 years. The most widely used method of applying the antileak is to insert it at the high points in the system and recover for reuse the excess that flows by gravity into the drips.

Leaks of a localized nature can be repaired by any convenient method best suited to the circumstances and type of line construction. Split sleeves (see Fig. 12) are available for bridging over defective joints or breaks in cast-iron mains[1] and similar devices can be had for stopping leaks in steel mains, including split sleeves for welding.[2] (See also Welding on Gas Lines under Pressure, page 1205.) Reference should be made to manufacturers' catalogues for available devices.

Expansion Stresses.—The general subject of expansion stresses in pipes due to change in temperature is discussed in Chap. VII on Expansion and Flexibility (see pages 754 to 860). Where pipes are buried in the ground and subjected to only small changes in temperature, as usually is the case with gas or water distributing systems, expansive movement is resisted by the gripping power of the earth and by branch connections. Under these conditions little, if any, cumulative movement can take place and elongation either is accommodated by slip in the compression couplings or bell joints, or else the tendency is opposed by external resistance which sets up a corresponding stress in the pipe metal (see pages 759 to 763). The latter condition exists in solid welded lines of buried steel pipe which have been found to withstand expansive stresses successfully where the temperature differences are not too great, say in the order of 20 to 30 deg F. With greater temperature differences it may be advisable to include slip joints such as com-

[1] See "Repairs of Breaks in Cast Iron Mains," by W. P. Geldard, *AGA Proc.*, 1936, pp. 602–605.

[2] See "Maintaining High Efficiencies in Pipe Lines," by J. C. Reinbold, *Natural Gas Proc.*, 1937, p. 54 (published by AGA), also available in reprint form from the American Gas Association, 420 Lexington Ave., New York 17, N.Y.

pression couplings between sections of welded line, otherwise excessive stresses set up in the pipe wall may cause the pipe to buckle as a long column (see page 761) or to pull apart at defective welds if such exist.

According to Erick Larson in the "American Gas Catalogue and Handbook,"[1] 1938, page 125:

> The practical temperature differential causing stress would be that caused by the difference between ground temperatures immediately following completion of construction and the subsequent minimum or maximum temperature to which the line would be subjected. When buried at depths of two to three feet, as found in usual practice, the change in temperature is considerably less than that occurring in the atmosphere above the trench.
>
> A number of soil studies appear to indicate that in northern territories the ground temperature at the bottom frost level plus two-feet additional depth does not vary with daily temperature changes and differs between the maximum and minimum temperatures during the whole year only 20 to 25 degrees F. For instance, when the frost penetrates into the ground to a depth of two feet the ground temperature four feet beneath the surface does not vary with daily temperatures but slowly increases and decreases with the seasons. In territories which seldom or never have frost the temperature differential between minimum and maximum is still less than the above at depths of three feet or more.

Where pipe is free to expand on bridges and other exposed places, or in gas-producing plants or compressing stations, the usual means for absorbing thermal expansion are called for as discussed in Chap. VII on Expansion and Flexibility. For a discussion of this problem as it applies to hot and cold gas piping, reference may be made to the AGA paper by L. W. Tuttle on "Pipe Expansion and Expansion Fittings."[2]

CROSS-COUNTRY TRANSMISSION LINES

Design, Materials, Construction.—The transmission of either manufactured or natural gas from the point of origin to where it is turned into the low-pressure distribution network or otherwise delivered to the customers presents interesting problems in engineering and economics. Large-diameter high-pressure transmission lines 1,200 miles or more long have been used for some time for conveying natural gas from remote oil fields to the big-city consuming centers. Numerous shorter lines of high capacity have been in use for many years. As long ago as 1891 a double

[1] Published by the American Gas Journal, Inc., 53 Park Place, New York, N.Y.

[2] See *AGA Proc.*, 1936, pp. 591–600.

8-in. line was laid from Greenstown, Ind., into Chicago, a distance of 120 miles. This line was constructed of wrought-iron screw pipe and employed mechanical compression with an initial pressure of 525 psi. In the course of subsequent development a considerable literature has accumulated on the engineering and economic aspects of design and on the operating problems encountered.[1]

ECONOMIC PROBLEMS

If gas is to be delivered to the consumers at a competitive price, investment costs must be held down to a minimum by striking a proper balance between cost of pipe line and cost of compressing stations required to serve it. Owing to the high pressures used, the long distances traversed, and the relatively small hazard involved in cross-country lines, it is customary to design them for a working stress not far below the yield point of the material in order to keep the initial cost of the line within economic limits.

Considerable study has been given the design of long-distance gas transmission in an effort to find what combination of factors gives the optimum result.[2] The problem resolves itself into a determination of what pipe line and compressor characteristics will deliver the required amount of gas at the lowest annual cost. If, for instance, the diameter of the pipe line is skimped, pressure

[1] For comprehensive general accounts of pipe-line construction practice see:

(a) "Pipe Line Section," *Oil Gas J.*, Sept. 23, and Sept. 30, 1944.

(b) "Tennessee Gas and Transmission Company Pipe Line," *Gas Age*, pp. 35–64 and 90–94, Nov. 16, 1944.

[2] The following references are cited as bearing on the economic angle:

(a) "Long Distance Transmission of Natural Gas," by George I. Rhodes and Edgar G. Hill, *AGA Proc.*, 1930, pp. 288–306, also *Trans. Am. Inst. Chem. Engrs.*, Vol. 25, pp. 58–84.

(b) "An Analysis of Gas Pipe-line Economics," by H. C. Lehn, *Trans. ASME*, Vol. 65, No. 5, pp. 445–460, July, 1943.

(c) "Gas Engineers' Handbook," by Gas Engineers' Handbook Committee of Pacific Coast Gas Association, and AGA, McGraw-Hill Book Company, Inc., New York, 1934, 1st ed., pp. 724–772.

(d) "High-pressure Pipe Line Research," by F. W. Laverty and F. M. McNall, *Trans. ASME*, April, 1944, Vol. 66, No. 3, pp. 215–219.

(e) "Gas Pipe Line Factors Effecting a Minimum Cost," by W. R. Kepler, *Proc. AGA*, 1930, pp. 797–819; *Gas Age-Record*, Vol. 66, 1930, pp. 709–714; and *Bull.* 510, A. O. Smith Corp., Milwaukee, Wis., 1930.

(f) "Economics of Pipe Lines for Natural Gas," *Proc. Engrs.' Soc. West. Penna.*, Vol. 49, 1932.

(g) "American Gas Practice," Vol. II, Distribution and Utilization of City Gas, by Prof. Jerome J. Morgan, 2d ed., 1935, pp. 411–436. Published by Jerome J. Morgan, Maplewood, N.J.

drops will be greater with a corresponding increase of horsepower for compression. This might be justified economically. On the other hand, if the whole level of line pressure was stepped up to offset the reduction in diameter and a heavier wall or stronger pipe used, the over-all result might be still better, or it might be worse. These suggested possibilities serve to indicate the nature of some factors involved in pipe line design.

The total annual cost of the pipe line alone consists of the fixed charges on the initial investment plus the direct operating expenses such as patrolling and maintenance. Since the latter are relatively small, they are often assumed as a percentage of the initial cost. The use of high-tensile-strength seamless or electric-welded steel pipe is said to be a major factor in cutting costs and make feasible the construction of long-distance transmission lines of large diameter into territory previously considered too remote from sources of supply. The cost of laying the line is estimated by some (see references, page 1201) as a percentage of the pipe cost, while others prefer to treat this as a separate item. It is general practice to estimate pipe cost on a straight per ton basis, possibly shading the figures to allow for some slight variation in cost with diameter.

Both the initial and operating costs of *compressor stations* are usually considered as varying with the installed horsepower. It is customary to figure pressure drops in transmission lines by the Weymouth formula (see page 186) and to space compressor stations in relation to the size of the lines so as to permit compression ratios in the order of 1.5 to 2. Formulas, charts, and typical cost figures are available in the references cited on page 1201. The station cost on a line of optimum design is said to be of the magnitude of 25 per cent of the total cost. These stations usually are composed of large gas-engine-driven compressors. Single units of from 1,000 to 1,500 hp each are normally used. On large systems these units are installed in batteries of 5 to 15 engines at intervals along the line, depending to some extent on deliveries to branches between stations and also on the economic capacity of the line. Data on the compression of natural gas will be found on pages 142, 162, and 198.

LINE CONSTRUCTION

The design, manufacture, installation, and tests of cross-country gas transmission lines outside the corporate limits of cities or villages, or in sparsely inhabited areas, are classified under Division

2 of Sec. 2 on Gas and Air Piping Systems of the American Standard Code for Pressure Piping, ASA B31, which is abstracted on pages 1208 to 1215. Coming under Division 2, cross-country transmission lines may be worked at higher stresses than would be permitted the same materials under Division 1. Maximum allowable working pressures for steel pipe in Division 2 are 80 per cent of the mill test pressure, while cast-iron pipe for this service may be designed without the usual allowance for mechanical strength and/or corrosion.

Pipe, Couplings, and Mechanical Joints.—Both steel and cast-iron pipe are used extensively for gas transmission lines. Steel pipe is bought to API Specification 5L for Line Pipe (see page 1131) owing to the extensive choice of wall thickness and tensile strengths available. The two common methods of joining steel line pipe are compression sleeve couplings (see page 1193) and fusion welding. Cast-iron pipe usually is of the centrifugally cast variety having bells suited for mechanical joints, although plain or grooved ends for Dresser or Victaulic couplings are used to a considerable extent (see pages 1194 to 1196). The rubber gaskets used in couplings and mechanical joints must be of special composition to resist oil and remain resilient for the life of the line.

Welding.—Experience has proved that it is feasible to build comparatively large solid-welded lines of steel pipe that have stood up remarkably well with few, if any, interruptions occurring. In California where there is only a small difference between seasonal maximum and minimum ground temperatures at pipe depths, solid-welded lines as large as 26 in. in diameter have been laid successfully. Elsewhere, however, it has not been thought advisable to lay all of the long large-size, high-pressure lines with sufficient slack to warrant solid-welded construction, so a part of such lines have been laid by alternating welds with compression sleeve couplings to relieve the expansive movement caused by temperature variations.

Three methods are in use for field welding of joints in steel line pipe, *viz.*, oxyacetylene fusion welding, oxyacetylene pressure welding, and electric-arc welding. Welded construction is justly growing in favor with advances in the art of welding which tend to ensure getting unbreakable bottletight joints and at the same time reduce the cost of the line. Oxyacetylene fusion-welded joints are convenient for field work owing to the simplicity and portability of the equipment, and excellent results can be obtained par-

ticularly where the work can be rolled during welding. Although electric-arc welding using coated electrodes may be better for overhead fixed-position welding, the engine-driven generator set required for field work is more costly and cumbersome than oxyacetylene equipment.

Pressure welding of field joints is a more recent development and is done by placing the two pipe ends together under a thrust of 150 to 450 psi while the metal is heated with an oxyacetylene gas-fired ring to a temperature where it becomes soft but does not run.[1] When this condition is reached, the thrust is raised to around 750 psi which upsets the metal to about twice the pipe-wall thickness and effects a pressure-welded joint. The actual welding can be finished in about one minute from the time the torches are lighted, and the output of the equipment depends chiefly on the speed with which it can be moved from one joint to the next. It is said that under favorable conditions an average of as high as 16 joints per hour can be welded in 10-in. pipe by this process.

As would be expected, in order to make the highest speed, it is essential to have a good right of way to facilitate moving the heavy equipment required for pressure welding. This equipment consists of a hydraulic press suspended from a boom mounted on a caterpillar-type tractor which carries a hydraulic pump and tows a trailer with oxyacetylene gas tanks, all of which are interconnected with rubber hose lines to supply hydraulic pressure and fuel gas at the joint.

Tapping and Repairing Lines under Pressure.—Gas transmission and distribution lines often have to be tapped or repaired under pressure to avoid inconvenient and costly shutdowns of the system. A variety of bolted and gasketed repair clamps are available for such jobs, as illustrated in manufacturers' catalogues. Welding also is employed as a means of stopping leaks or making connections to steel lines. A survey of this practice was conducted by a committee of the American Gas Association which covered companies serving more than one-half of all meters installed in the United States.[2] Of the 95 companies that replied, 52 reported

[1] For an illustrated description of this process, reference may be made to an article by Paul Reed on "Operation of the Pressure Welding Process," *Oil Gas J.*, Dec. 17, 1942, pp. 35, 45. See also "Pressure Welded Pipe Line," by Elton Sterrett, *The Welding Engineer*, February, 1945, pp. 37–39.

[2] "Report on Welding Gas Lines under Pressure," *Interim Bull.* 2-42, Accident Prevention Committee, AGA. Copies may be obtained from American Gas Association, 420 Lexington Ave., New York 17, N.Y.

that they regularly made repairs by welding with the lines under pressures ranging from 1 psi up to 350 psi, three companies stated that they did so sometimes, and 42 replied in the negative. Some used electric-arc welding and others oxyacetylene. In answer to a question about making "hot taps" in which nipples or saddles are welded onto steel lines under pressure and holes tapped through afterward, 87 companies replied in the affirmative, 3 replied "occasionally," and 5 replied in the negative. Pressures as high as 600 psi were included. The incentive to weld gas lines under some pressure at least is considerable owing to the inconvenience of shutdown and to the necessity of purging a line where no pressure exists in order to avoid the possibility of igniting an explosive mixture of gas and air inside the pipe.

The following section on Electric Welding on Lines under Pressure, taken from a paper[1] by J. C. Reinbold, sets forth the technique used by the Panhandle Eastern Pipe Line Company in making repairs by electric-arc welding. Whereas Reinbold favors arc welding, others probably are equally enthusiastic about gas welding; at least both types are used to about the same extent for outdoor work where gas-welding equipment is more portable.

Electric welding on lines under pressure has eliminated many major shut-outs with their consequent interruption of service and loss of gas. It has perhaps done as much toward maintaining high efficiencies in pipe lines as any other comparatively recent development. It completely and finally takes care of many leak repairs, line connections and reinforcements that formerly were accomplished through temporary means while awaiting opportune times for shut-outs in which to make them permanent.

Electric welding under such conditions is generally position welding. One-eighth and five thirty-seconds inch electrodes have been found satisfactory for this service. The rod is used with reversed polarity and under normal conditions with amperage set at 100 and voltage 25 to 30.

Electric welding under pressure is employed in the installation of repair patches, split sleeves (see Fig. 12*a*), and line taps. Patching is resorted to in the repair of pipe flaws or pits, defective welds and defective seams. In the repair of pipe flaws and pits the patch or shoe is cut to fit and clamped in place during the welding process. In the repair of defective welds the patch is shaped to take care of weld protrusion, and before installation is fitted with a pipe coupling tap of suitable size through which escaping gas may be piped from the bell hole during welding. Rubber gaskets are often necessary under the patch around the leak if escaping gas is excessive. In such instance the patch is clamped in place before welding is commenced. In the repair of defective seams in pipe of

[1] "Maintaining High Efficiencies in Pipe Lines," by J. C. Reinbold, *Natural Gas Proc.*, 1937, p. 54 (published by AGA). Reprints may be obtained from the AGA (see note, p. 1204).

more than 16 in. in diameter, the patch is usually cut to extend longitudinally 18 in. or two feet on each side of the defect and circumferentially eight to ten inches. Heavy clamps are used around the pipe near the ends of the patch for reinforcing purposes and at least two clamps are used in holding the patch in place until such time as their removal is necessary to complete the weld. The welds over the defective seam are not made until the balance of the patch has been welded. As in the repair of defective welds, any escaping gas is piped from the bell hole, and as an added precaution rope asbestos is sometimes placed under the patch between the gasket and welding edge to protect gasket material from possible burns.

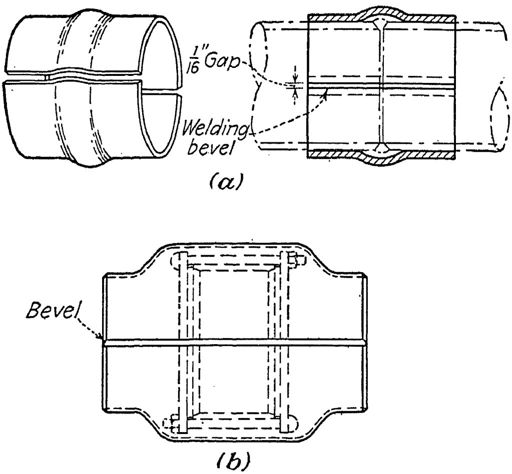


FIG. 12.—Use of split welding sleeves: (*a*) to repair defective line-joint weld; (*b*) to repair leaking coupling.

In the repair of coupling leaks or the installation of split welding sleeves over couplings (see Fig. 12*b*) preparatory to the installation of road casing, several preliminary steps are necessary. A suitable sized pipe coupling is welded to one of the halves of the sleeve for the purpose of carrying off any escaping gas and backup strips are also tacked to one of the halves on the underside of the longitudinal seams to facilitate welding. These are so shaped that interference with a snug fit is avoided. After the pipe surface is thoroughly clean the sleeve is clamped in position over the line coupling. The side seams are first welded with a series of small beads and the weld material is built up approximately one-eighth inch for reinforcement. The ends are then welded to the pipe as in patch welding and when completed a bull plug is screwed into the blow-off coupling and finished with a light sealing bead.

Line taps on the side, bottom, or top are made by saddling the branch to the line and reinforcing with gussets, straps, or welding saddles. Welding saddles are usually placed over the branch before welding same to the line. Where the

length of the branch does not permit this, the welding saddle is split and rewelded around the branch. Welding of the saddle follows and subsequently the tap is made by drilling through with a tapping machine.

In electric welding where the line is under pressure, extreme care must be exercised to confine the heat to a small area. Temperatures should be tested by hand from time to time in the area of the weld and only small beads should be run. Each such bead should be scaled and the pipe allowed to cool before applying the next one. In patch welding three or four small beads are usually sufficient. The final operation is the running of the weaved or lace-up bead in which only enough metal is carried to cover up any small pin holes in the weld. In running the lace-up bead more heat is applied to the patch than to the pipe inasmuch as a short forceful arc is used for this type of weld and care must be exercised to avoid burning into the pipe.

Electric welding on pressure lines, although now somewhat common, carries its risks unless precaution, experience, and forethought are exercised. It should not be employed on any line which is stressed beyond a reasonable factor of safety or which is under exceptional strain. Electric welding on thin wall or excessively corroded pipe which is under pressure should be avoided.

Protective Coatings.—Where soil conditions are corrosive enough to warrant doing so, protective coatings of wax, tar, or asphalt wrapped with felt should be applied to the exterior of steel pipe (see pages 1264 to 1268). In most soils it is advisable at least to paint the outside of the pipe as it is laid with a good asphalt or bitumastic paint. Cast-iron pipe for gas service usually is coated at the manufacturer's works on the outside only with coal-tar-pitch varnish.

Natural Obstacles.—Many and varied natural obstacles have to be overcome in building long-distance transmission lines.[1] In going over mountain ranges it sometimes is necessary to blast solid rock for many miles to make a trench for the pipe. Through narrow mountain passes this work often requires as much rock excavation as is necessary for a railroad or highway. In crossing long swamps, dredges are used and the pipe must be heavily weighted to overcome its natural buoyancy when submerged in a mixture of mud and water. Rivers are crossed sometimes with special pipe-line bridges and in other cases with submerged multiple line crossings connected to manifolds on either bank. Each

[1] For an interesting account of natural obstacles to be overcome with illustrations showing methods of line construction under difficulties, reference may be made to the paper on "Large Diameter Transmission Lines," by Elmer F. Schmidt, *Natural Gas Proc.*, 1932, p. 57 (published by AGA). Reprints also may be obtained from the American Gas Association, 420 Lexington Ave., New York 17, N.Y. See also "Constructing the Lirette-Mobile Natural-gas Transmission Line," by W. B. Poor, *Mech. Eng.*, January, 1943, pp. 9–13.

line has a valve where it attaches to the header so that in case one or more lines are broken during a flood, the valves can be closed and the load carried by the remaining lines.

AMERICAN STANDARD CODE FOR PRESSURE PIPING

Section 2

GAS AND AIR PIPING SYSTEMS

ASA B31.1-1942

Abstracted[1]

Space here does not permit abstracting from the code all of Sec. 2 on Gas and Air Piping Systems. The section embraces piping for conveying air, fuel gas, illuminating gas, and other substantially noncorrosive gases at temperatures not in excess of 450 F as used in city gas distribution systems, in cross-country transportation systems, in gas manufacturing plants, in gas or air compressing stations, and in processing plants. Piping for gas or air at temperatures above 450 F is required to conform to Sec. 3 on Oil Piping Systems, Including Oil Vapor, Refinery Gas, and Gasoline Recovery Plants.

Services to which Sec. 2 *does not* apply:

1. Air piping under pressures 30 psi and below (exempted from code requirements).
2. Equipment or apparatus, or pipe connections which are a part of apparatus. (For requirements that do apply, see American Standard Safety Code for Compressed Air Machinery and Equipment, ASA B19, page 1215, and AGA Codes and recommended practices for the installation of gas-burning appliances and equipment copies of which can be obtained from AGA or ASA headquarters.)
3. Piping lined with firebrick or other refractory material used for conveying hot gases. (No code available.)
4. Piping used for conveying refinery process gas in oil refineries or gasoline recovery plants, and for sewer gas; or ducts for waste gases and ventilation. (For refinery gas piping, see pages 1148 to 1162).
5. Casting, tubing, and pipe when used in gas or oil wells, and natural-gas and oil-field producing and gathering systems outside the boundaries of cities and villages. The gathering system is defined as a piping system which carries gas from wells to a common point in the producing field where the transportation system begins. (No code available.)
6. Residential and commercial building gas piping systems. (For other codes that do apply, see pages 1167–1175.)

The piping systems in Sec. 2 of the Code are grouped in two divisions depending on the hazard involved:

[1] This abstract is based on ASA B31.1-1942. For the complete code or revisions to same, reference should be made to the latest issue of ASA B31.1, copies of which may be obtained from the American Standards Association, 29 West 39th St., New York 18, N.Y.

Division 1 includes all gas and air piping systems which are constructed: (*a*) in mines or in power, industrial, and gas manufacturing plants wherever located, or (*b*) anywhere within the boundaries of cities or villages, except that where cross-country transportation systems extend through sparsely populated or rural territories that are within the legal boundaries of cities or villages, such portions of the cross-country transportation systems shall be included within Division 2.

Division 2 includes all gas and air piping systems which are (*a*) cross-country transportation systems, including the piping of their compressing stations, and (*b*) other systems not included in Division 1.

Distinction is made throughout Sec. 2 where requirements apply to only one of the foregoing divisions or differ between the two. The most important departure is between the formula for computing the "required thickness of pipe" in Division 1 and the corresponding, but different, formula for computing the "maximum allowable working pressure" in Division 2. As might be expected the requirements of Division 1 are more conservative than those of Division 2. The formula for Division 1 bears a general similarity to those used in other sections of the Code (see pages 42 to 47), although in some cases different S values and C factors are employed. The requirements of Division 2 are notably different in that they chiefly depend on relating allowable working pressure to the hydrostatic test pressure applied at the pipe mill. Owing to these significant differences, the complete formulas and S values for both Division 1 and Division 2 of Sec. 2 are reproduced here.

220. Thickness of Pipe. Division 1.—(*a*) For inspection purposes, the minimum thickness of pipe wall required shall be determined by formula (1) for the designated pressure and for temperatures not exceeding 450 F for steel, wrought iron, or cast iron, and 406 F for brass or copper:

$$t_m = \frac{pD}{2S} + C \tag{1}$$

where t_m = minimum pipe wall thickness, in.[1]
p = maximum internal service pressure, psi gauge.[2]
D = outside diameter of pipe, in.[3]
S = allowable stress in material due to internal pressure, at the operating temperature, psi. For S values, see Table VII, page 212.
C = allowance for threading, mechanical strength, and/or corrosion, in.

(*b*) The value of C used in formula (1) shall not be less than that given for the respective material in the following list:

[1] To the minimum pipe wall thickness calculated from formula (1), the manufacturing tolerance, demanded for the pipe considered, must be added to obtain the nominal wall thickness. (See ASA Standard B36.10.)

[2] When computing the allowable pressure for a pipe of a definite minimum wall thickness, the value obtained by this formula may be rounded out to the next higher unit of 10.

[3] For design, the outside diameter of pipe as given in tables of standards and specifications shall be used in obtaining the value of t_m. Where available from pipe on hand or in stock, the actual measured outside diameter and actual measured wall thickness may be used to calculate the maximum internal service pressure.

Type of Pipe	Value of C[1] in Inches
Cast-iron pipe centrifugally cast or cast horizontally in green sand molds	0.14
Cast-iron pipe, pit cast	0.18
Threaded steel, wrought iron, or nonferrous pipe	Depth of thread,[2] or formula (2), whichever is greater
Grooved steel, wrought iron, or nonferrous pipe	Depth of groove, in.
Plain end[3] nonferrous pipe or tube	0.00
Plain end[3] steel or wrought iron	See formula (2)

$$C = \frac{600}{Y} + a \tag{2}$$

where Y = minimum specified yield point for material in question, psi.

a = zero when piping is not subject to internal corrosion and when laid in noncorrosive soil and not subject to electrolytic corrosion or is adequately protected against corrosion. For indeterminate conditions a = 0.025 in., minimum. See Par. 220 (*h*).

(*c*) The value of S in formula (1) shall not exceed that given in Table VII for the respective material and specification.

(*d*) When steel pipe is manufactured under a specification not listed in Table VII, the value of S may be $0.6K$ for service temperatures not in excess of 100 F and $0.52K$ for a service temperature of 450 F (where K is the stipulated minimum effective yield strength), S values for intermediate temperatures to be obtained by interpolation. The value of p allowed under Par. 220(*d*) shall not exceed two-thirds of the mill test pressure for service temperatures not in excess of 100 F and five-ninths of the mill test pressure for a service temperature of 450 F, p values for intermediate temperatures to be obtained by interpolation. (See Par. 222.)

(*e*) Threaded pipe thinner than listed in any of the specifications for threaded pipe enumerated in Table VII shall not be used.

(*f*) The value of p shall not be taken at less than 180 psi for pipe diameters up to and including 24 in., and 100 psi for pipe diameters greater than 24 in., except that this requirement does not apply to cast-iron pipe or to copper tubing.

(*g*) Pit-cast iron pipe shall have a minimum thickness equal to either the AGA Specification or Class *A* of the AWWA Specification. Cast-iron pipe, centrifugally cast or cast horizontally in green sand molds, shall have a minimum thickness equal to Class 50 of ASA A21. Nonferrous tubing shall have a minimum thickness equal to Class *M* of ASTM Specification B88. Additional wall

[1] Except for plain end nonferrous pipe or tube, the values of C stipulated above are such that the actual stress in the wall of the pipe is less than the values of S given in Table VII as applicable in formula (1).

[2] The crest and root of thread are truncated a minimum amount equal to $0.033p$ except for 8 threads per inch which are truncated $0.045p$ at the crest and $0.033p$ at the root. The (basic) maximum depth of the truncated thread, h, is $0.80p$ except for 8 threads per inch which is $0.788p$. (See American Standard for Pipe Threads, ASA B2.1-1942.)

[3] Plain end includes pipe joined by flared compression couplings, lap (Van Stone) joints, and by welding, *i.e.*, by any method which does not reduce the wall thickness of the pipe at the joint.

thickness shall be provided as required to care for corrosion, erosion, or mechanical strength.

(*h*) The value of $a = 0.025$ in. in formula (2) is a minimum value for use where corrosion-free conditions cannot be demonstrated, and is not intended to be adequate for service involving corrosion. In such cases adequate measures shall be taken to insure that the pipe is not rendered unsafe by corrosion. These measures may comprise additional wall thickness, protective coating, cathodic protection, special alloys, etc., at the option of the user.

221. Determination of Effective Yield Strength, with Special Reference to Division 2.—The effective yield strength of steel pipe, designated as factor K in formulas (3), (4), and (6) and in Par. 220(*d*), may be determined as follows:

(*a*) The factor K may be taken as the product of Y, the stipulated minimum yield strength, and E, the efficiency of the joint. When K is taken as YE, Y may be determined by the method employed for determination of yield strength in the individual specifications selected from those for steel pipe listed in Table VII and E shall be taken from footnote 3 to Table VII, but E may be taken as 1.0 for electric-resistance-welded steel pipe provided supplemental tests are made which demonstrate that the strength of the weld is equal to the minimum specified tensile strength of the pipe. In those cases where the specification for the steel pipe is not listed in Table VII, Y may be determined by a method employed for determination of yield strength in any one of the specifications for steel pipe listed in Table VII.

(*b*) Alternatively, the factor K may be determined by internal hydrostatic pressure tests on finished lengths of pipe or on cylindrical samples cut from finished lengths of pipe in accordance with formula (3) as follows:

$$K = \frac{p_y D}{2t} \qquad (3)$$

where K = effective yield strength, psi gauge.

*p_y = pressure in pounds per square inch gauge causing the stress in the metal to reach the yield strength.

t = nominal or specified pipe wall thickness, in.

D = nominal or specified outside diameter of pipe, in.

222. Mill Testing.—(*a*) Every valve and fitting shall be capable of withstanding an internal hydrostatic mill test without showing failure, leakage, distress, or distortion other than elastic distortion at a pressure not less than one and one-half times the maximum working pressure for which the manufacturer guarantees it.

(*b*) Every cast-iron pipe manufactured hereafter for use in piping systems within the scope of this section shall be subjected to and safely withstand an internal hydrostatic mill test without showing failure, leakage, or distress at a pressure not less than provided in the appropriate specification of those enumerated in Table VII, and not greater than would produce a stress equal to one-half the tensile strength.

* The value of p_y may be taken as the pressure required to cause a volumetric offset of 0.2 per cent or as the pressure required to cause a permanent increase in circumference of 0.1 per cent at any point, but other suitable methods of determining that the stress in the steel has reached the yield strength may be used, provided such methods conform in all respects to recognized engineering practices.

TABLE VII.—ALLOWABLE *S* VALUES FOR PIPE IN GAS AND AIR PIPING SYSTEMS[1]
(Table 6 of ASA B31.1-1942)

Material	Specification	Values of *S*, psi for temperatures in deg F not to exceed[2] 100	400	450
Seamless steel:				
Grade *A*	ASTM A 106	18,000	.....	15,650
Grade *B*	ASTM A 106	21,000	.....	18,250
Seamless steel:				
Grade *A*	API 5 L	18,000	.....	15,650
Grade *B*	API 5 L	21,000	.....	18,250
Grade *C*	API 5 L	27,000	.....	23,450
Seamless steel:				
Grade *F*-25	API 5 A	15,000	.....	13,000
Grade *H*-40	API 5 A	24,000	.....	20,800
Grade *J*-55	API 5 A	33,000	.....	28,500
Seamless steel:				
Grade *A*	ASTM A 53	18,000	.....	15,650
Grade *B*	ASTM A 53	21,000	.....	18,250
Seamless steel:				
Ordinary use	ASTM A 120	15,900	.....	13,800
Electric-fusion-welded steel: ASME Unfired Pressure-vessel Code 18 in. and larger (Par. U68)				
Grade *A*	ASTM A 155	12,950	.....	11,250
Grade *B*	ASTM A 155	14,600	.....	12,700
Grade *C*	ASTM A 155	14,950	.....	13,000
Electric-fusion-welded steel: 8 to 30 in.				
Grade *A*	ASTM A 139	14,400	.....	12,500
Grade *B*	ASTM A 139	16,800	.....	14,600
Electric-fusion-welded steel:				
Grade *A*	API 5 L	14,400	.....	12,500
Grade *B*	API 5 L	16,800	.....	14,600
Grade *C*	API 5 L	21,600	.....	18,750
Electric-fusion-welded steel:				
Grade *F*-25	API 5 A	12,000	.....	10,400
Grade *H*-40	API 5 A	19,200	.....	16,700
Grade *J*-55	API 5 A	26,400	.....	22,900
Electric-fusion-welded steel:				
Large (30 in. and larger)	ASTM A 134	0.48 *Y*[3]	.....	0.42 *Y*[3]
Electric-resistance-welded steel:[3]				
Grade *A*	ASTM A 135	15,300	.....	13,300
Grade *B*	ASTM A 135	17,850	.....	15,500
Electric-resistance-welded steel:[3]				
Grade *A*	API 5 L	15,300	.....	13,300
Grade *B*	API 5 L	17,850	.....	15,500
Grade *C*	API 5 L	22,950	.....	19,900
Electric-resistance-welded steel:[3]				
Grade *F*-25	API 5 A	12,700	.....	11,000
Grade *H*-40	API 5 A	20,400	.....	17,700
Grade *J*-55	API 5 A	28,000	.....	24,300
Forge-welded steel 14 to 96 in.:				
Grade *A*	ASTM A 136	11,500	.....	10,000
Grade *B*	ASTM A 136	12,950	.....	11,250

TABLE VII.—(*Concluded*)

Material	Specification	Values of S, psi for temperatures in deg F not to exceed[2]		
		100	400	450
Lap-welded steel: Open hearth or electric furnace	ASTM A 106	12,000		10,400
	ASTM A 53	12,000		10,400
	ASTM A 120	12,000		10,400
	API 5 L	12,000		10,400
Lap-welded steel: Open hearth or electric furnace				
Grade F-25	API 5 A	12,000		10,400
Grade H-40	API 5 A	19,200		16,700
Grade J-55	API 5 A	26,400		22,900
Lap-welded steel: Bessemer	ASTM A 53	14,400		12,500
	ASTM A 120	12,000		10,400
	API 5 L	14,400		12,500
Butt-welded steel: Open hearth or electric furnace	ASTM A 53	9,000		7,800
	ASTM A 120	9,000		7,800
	API 5 L	9,000		7,800
Butt-welded steel: Bessemer	ASTM A 53	10,800		9,400
	ASTM A 120	9,000		7,800
	API 5 L	10,800		9,400
Lockbar joint steel	ASTM A 137	0.6 YE*		0.52 YE*
Riveted joint steel or wrought iron	ASTM A 138	0.6 YE*		0.52 YE*
Lap-welded wrought iron	ASTM A 72	11,500		10,000
	API 5 L	11,500		10,000
Butt-welded wrought iron	ASTM A 72	8,650		7,500
	API 5 L	8,650		7,500
[illegible]	ASTM B 43	7,600	7,000	
Seamless copper pipe	ASTM B 42	7,000	6,000	
Seamless copper tubing	ASTM B 75	7,000	6,000	
	ASTM B 88	7,000	6,000	
Cast iron pipe, centrifugally cast or cast horizontally in green sand molds	F.S.E.C. WW-P-421	6,000		6,000
Cast iron pipe: pit-cast	A.W.W.A., A.G.A.	4,000		4,000

[1] For allowable S values for steel pipe fabricated under specifications other than those listed see Par. 220(d).

[2] Allowable S values for intermediate temperatures may be obtained by interpolation.

[3] Where pipe furnished under this classification is subjected to supplemental tests and/or heat treatments as agreed to by the supplier and the purchaser, and whereby such supplemental tests and/or heat treatments demonstrate the strength characteristics of the weld to be equal to the minimum tensile strength specified for the pipe, the S values equal to the corresponding seamless grades may be used.

* Y = Minimum specified yield strength of the material (but not to be taken as over 80 per cent of the minimum ultimate tensile strength).

E = Efficiency of the joint. The values of E on which the S values in Table VII are based are as follows:

Seamless	1.00	Fusion welded ordinary	0.80
Fusion welded ASME rule (Par. U68)	0.90	Forge welded	0.80
Resistance welded	0.85	Lap welded	0.80
		Butt welded	0.60

Joint efficiencies of mechanical joints to be calculated from dimensions.

(c) Every pipe (other than cast-iron pipe) manufactured hereafter for use in piping systems within the scope of this section shall be subjected to and safely withstand an internal hydrostatic mill test without showing failure, leakage, distress, or distortion other than elastic distortion at the mill test pressure stipulated in the specification under which the pipe is manufactured.

(d) When steel pipe is manufactured under a specification not listed in Table VII (including a specification which differs from one listed in Table VII only with regard to mill test pressure), such specification shall stipulate a minimum tensile strength and a minimum effective yield strength in addition to the mill test pressure. The stipulated mill test pressure shall conform to formula (4) as follows:

$$p_m = \frac{NKt}{D} \tag{4}$$

where p_m = mill test pressure, psi gauge.

*N = a numerical factor usually taken as not less than 0.90 and not greater than 1.60.

K = stipulated minimum effective yield strength, psi gauge.

t = nominal or specified pipe wall thickness, in.

D = nominal or specified outside diameter of pipe, in.

(e) Provided mill testing equipment is available which will permit testing pipe at higher pressure without producing distortion beyond the dimensional tolerances permitted in the specification under which the pipe is manufactured, a value of N in excess of 1.60 may be taken, but in no case shall the value of N exceed 1.80.

(f) The specification shall stipulate the manner of selection and the number of samples to be taken for test and the test methods to be employed to verify compliance with the requirements stipulated as to tensile strength and effective yield strength. Acceptable methods of determining effective yield strength are given in Par. 221, but other methods may be stipulated providing such methods conform in all respects to recognized engineering practices.

(g) The value of K stipulated shall not exceed 80 per cent of the stipulated minimum tensile strength.

223. Pressure Testing after Installation.—(a) Every piping system within the scope of this section shall be capable of withstanding a test pressure of

(1) 150 per cent of the maximum service pressure, for systems within the scope of Division 1.

(2) 50 psi greater than the maximum service pressure, for systems within the scope of Division 2.

(b) A test made after installation may be made with air or gas pressure which for systems within the scope of Division 1 need not exceed 120 per cent of the maximum allowable working pressure, and for systems within the scope of Division 2 shall not exceed 120 per cent of the maximum allowable working pressure.

224. Working Pressure. Division 2.—(a) The maximum allowable working pressure for piping systems within the scope of Division 2 constructed with cast iron pipe shall be 200 psi less than the mill test pressure, but in no case greater than that determined by formula (5):

* The value of N shall not be taken as less than 0.90, but this shall not be construed to require a test pressure in excess of 2,800 psi.

$$\frac{2St}{D} \tag{5}$$

where p = maximum allowable working pressure, psi gauge.
S = 5,000 for pit-cast.
= 7,500 for centrifugally cast or cast horizontally in green sand molds
t = nominal or specified pipe wall thickness, in.
D = nominal or specified outside diameter of pipe, in.

(*b*) The maximum allowable working pressure for piping systems within the scope of Division 2 constructed with pipe which has been mill tested subsequent to the official adoption of this code in accordance with Par. 222(*c*) shall be 80 per cent of the mill test pressure.

(*c*) The maximum allowable working pressure for piping systems within the scope of Division 2 constructed with pipe (except cast iron pipe) which has not been mill tested subsequent to the official adoption of this code in accordance with Par. 222(*c*) shall be, at the option of the user, either 80 per cent of the mill test pressure or that determined by formula (6):

$$p = \frac{1.44Kt}{D} \tag{6}$$

where p maximum allowable working pressure, psi gauge.
K effective yield strength in pounds per square inch gauge, to be determined by the user in accordance with Par. 221.
t = nominal or specified pipe wall thickness, in.
D = nominal or specified outside diameter of pipe, in.

AMERICAN STANDARD SAFETY CODE FOR COMPRESSED AIR MACHINERY AND EQUIPMENT

Section 6

PIPING

ASA B19-1938

Rule 6.1. General.—Piping shall meet the requirements of the American Standard Code for Pressure Piping (ASA B31.1). In its installation, adequate provision shall be made for expansion and contraction, and to counteract pulsation and vibration. Steam and air piping shall be equipped with adequate traps or other means for removing liquid from the lines. Air discharge piping shall be so installed that pockets where oil may accumulate are avoided.

Rule 6.2. Insulation.—Steam piping shall be adequately insulated where it is exposed to contact.

Rule 6.3. Compressor Air Intake.—Provision shall be made to prevent drawing air containing inflammable or toxic gases, vapors, or dusts into the compressor, and to prevent steam, water, or waste of any sort from being blown or drawn into the compressor intake. No valve shall be installed in the air intake pipe to an air compressor with an atmospheric intake.

Rule 6.4. Air Discharge Piping.—The air discharge piping from the compressor to the air receiver shall be at least as large as the discharge opening on

the air compressor. Because the air discharge pipe becomes hot, no wood or other inflammable material shall be permitted to remain in contact with it.

Rule 6.5. Valves.—(*a*) Stop valves in the air line between the compressor and the air receiver are not recommended. In every case where a stop valve is so installed, one or more spring-loaded safety valves shall be installed between the compressor and the stop valve. The total capacity of such safety valves shall be sufficient to limit the pressure in the air discharge piping to 10 per cent above the relieving pressure of the safety valves on the air receiver.

(*b*) Any stop valve which may be installed in the air discharge piping shall be so located that it can be inspected and cleaned at definite regular intervals. Stop valves shall preferably be of the gate type and not of the globe type. If a globe valve is used, it shall be so installed that the pressure is under the seat and that the valve will not trap condensation.

(*c*) Every steam-driven air compressor shall be provided with a manually operable throttle valve in the steam supply line, in a readily accessible location.

(*d*) A stop valve shall be installed between the air receiver and each piece of stationary utilization equipment at a point convenient to the operator thereof.

(*e*) A stop valve shall be installed at each outlet to which an air hose may be attached.

CHAPTER XVI

REFRIGERATION PIPING

Refrigeration piping is generally understood to include refrigerant and brine piping. A refrigerant is defined as a substance used to produce refrigeration by its expansion or vaporization. Brine is defined as any liquid used for the transmission of heat without a change in state and having no flash point, or a flash point above 150 F determined by the method specified in ASTM Specification D56. The term "refrigerant" is customarily understood to apply to any fundamental refrigerating medium such as carbon dioxide, ammonia, methyl chloride, sulphur dioxide, or the "Freon" refrigerants, whereas secondary or indirect cooling mediums such as a calcium chloride and water solution are referred to as brine. Any substance or medium used in mechanical refrigeration wherein the working substance is evaporated, compressed, condensed, or liquefied, and then is reexpanded or vaporized, is termed a refrigerant in contrast to processes where the refrigerant medium is wasted as in melting ice, or in the conversion of carbon dioxide from a solid to a gas.

TYPES OF REFRIGERATION SYSTEMS

The mechanical refrigeration type wherein reciprocating, centrifugal, and rotary design compressors are used is most common. Steam ejector and absorption refrigeration systems are employed to a more limited extent according to their adaptability to the particular conditions of use. For methods of computing refrigeration loads, including those for air conditioning, and sizes of equipment necessary to provide for them, reference should be made to standard texts on refrigeration, the "ASRE Refrigerating Data Book,"[1] or the "ASHVE Guide."[2]

Mechanical Compression.—This method of refrigeration depends on the evaporation of a liquid refrigerant which, in evaporating,

[1] Published by the American Society of Refrigerating Engineers, 50 West 40th St., New York 18, N.Y.

[2] Published by the American Society of Heating and Ventilating Engineers, 51 Madison Ave., New York 10, N.Y.

extracts the heat of vaporization from the substance to be cooled. Refrigerant vapor, saturated or slightly superheated, is then compressed and discharged at a higher pressure and correspondingly higher saturation temperature. The vapor is condensed at this higher temperature by a cooling agent, usually water or air, to which it releases the heat of compression. The liquid refrigerant then flows to the expansion valve, owing to the lower pressure at the expansion valve, where the pressure is further reduced to that

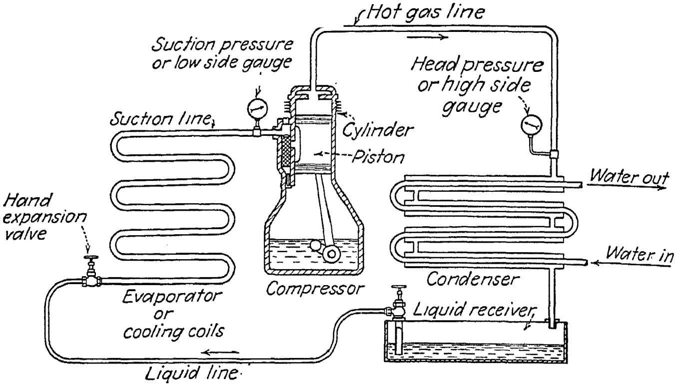


FIG. 1.—Piping diagram for direct-expansion refrigerating system. (*Courtesy of Trane Company.*)

existing in the direct expansion cooling coils. Heat necessary for vaporizing the liquid refrigerant at the lower temperature existing in the coils is extracted from the fluid surrounding the outer surface of the coil. This, of course, results in the desired cooling effect. The cycle of the mechanical compression system, illustrated in Fig. 1, employs a reciprocating compressor, a condenser and liquid receiver, and direct-expansion cooling coils. Reciprocating, centrifugal, and rotary types of mechanical equipment all operate on that cycle, the principal variation being the type of compressor used.[1]

Steam Ejector and Absorption Systems.—The fact that water will vaporize at low temperatures when subject to a high vacuum

[1] For a detailed discussion of the principles of refrigeration, reference should be made to standard texts on refrigeration, or the "ASRE Refrigerating Data Book," *op. cit.*

is used in the steam ejector type of compressor. Steam jet air ejectors of the type used in power plants to remove condenser air will provide the low absolute pressure necessary to induce evaporation of the water. The absorption system is somewhat more complex as regards the nature of the cycle although one of its principal advantages is the absence of moving parts, except for a pump. At present, the most commonly used refrigerant-absorbent combinations are (1) water and ammonia and (2) dichloromonofluoromethane and dimethyl ether of tetraethylene glycol. Both the steam ejector and absorption systems compare most favorably with the mechanical compression system when an inexpensive source of cooling water and steam, or other heat, is available. Absorption-type refrigeration systems using gas are being developed for use in air conditioning.[1] Since the mechanical compression type of refrigeration system is by far the most widely used at present, discussion of piping in this chapter will be confined largely to this system although the principles considered are generally applicable to any refrigeration piping.

PIPING DESIGN

The piping for mechanical refrigerating systems may be divided into three principal parts (see Fig. 1): (1) the hot-gas discharge line from the compressor to the condenser, (2) the liquid line leading from the condenser or liquid receiver to the cooling coil, (3) the suction line connecting the cooling coil to the compressor. The direct and indirect systems commonly used in connection with mechanical compression refrigeration systems with a schematic indication of the auxiliary piping required are illustrated in Fig. 2, as given in the American Standard Safety Code for Mechanical Refrigeration, ASA B9.[2] Important considerations in the design of these several classes of refrigerating piping are discussed in this section.

Design Pressures.—To comply with the American Standard Code for Pressure Piping,[3] ASA B31.1, and the American Standard Safety Code for Mechanical Refrigeration, ASA B9,[2] piping systems for refrigerants must be designed for the pressures shown

[1] See "Gas for Summer Air Conditioning," *Heating & Ventilating*, February, 1944, p. 56–78.

[2] Copies may be obtained from the American Society of Refrigerating Engineers, 50 West 40th St., New York 18, N.Y.

[3] Copies are available from the American Society of Mechanical Engineers, 29 West 39th St., New York 18, N.Y.

in Table I, which are two-thirds of the test pressures given in Table VIII on page 1242.

A safe rule for design of piping systems for water and brine is that the pressure shall be the maximum that will be imposed on the system in normal operation, but should not be less than 100 psi,

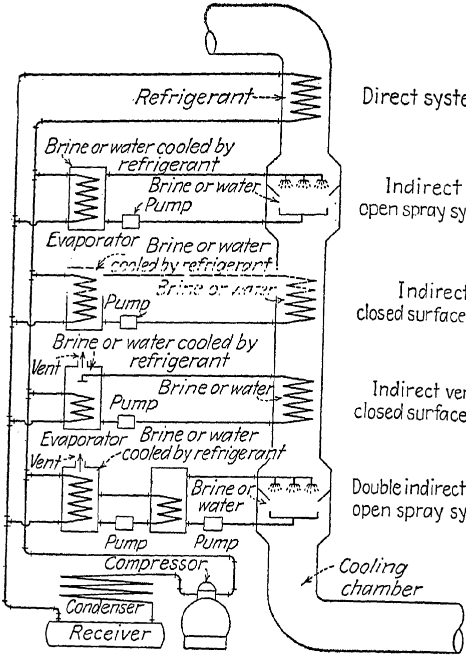


FIG. 2.—Piping diagram for various direct and indirect refrigerating systems. (*Reproduced by permission from the American Standard Safety Code for Mechanical Refrigeration, ASA B.9 published by the American Society of Refrigerating Engineers.*)

including, for cast-iron pipe, the water hammer allowances of Table VIII on page 47.

Pipe Size.—The rate of flow of fluids in pipe lines, whether liquids such as brines or water, or of vapors of volatile refrigerants is governed by a number of factors. A pressure difference gives the fluid the incentive to flow, which is opposed by a resistance that depends on the diameter, length, and interior surface condi-

TABLE I.—DESIGN PRESSURES FOR VARIOUS REFRIGERANTS COMPUTED FROM TEST PRESSURES GIVEN IN TABLE 41, CODE FOR PRESSURE PIPING

Refrigerant	Chemical formula	High-pressure side	Low-pressure side
Ammonia	NH_3	200	100
Butane	C_4H_{10}	60	34
Carbon dioxide	CO_2	1,000	667
Dichlorodifluoromethane (Freon-12)	CCl_2F_2	157	97
Dichlorotetrafluoroethane (Freon-114)	C_2Cl_2F	53	34
Dichloromethane (Carrene No. 1) (Methylene chloride)	CH_2Cl_2	20	20
Dichloromonofluoromethane (Freon-21)	$CHCl_2F$	47	34
Dichloroethylene	$C_2H_2Cl_2$	20	20
Ethane	C_2H_6	733	400
Ethyl chloride	C_2H_5Cl	40	34
Isobutane	$(CH_3)_3CH$	87	50
Methyl chloride	CH_3Cl	143	83
Methyl formate	$HCOOCH_3$	34	34
Propane	C_3H_8	217	140
Sulphur dioxide	SO_2	133	63
Trichloromonofluoromethane (Freon-11)	CCl_3F	34	20

tion of the pipe and on the density, viscosity, and mean velocity of the fluid. These are the same factors that influence the flow of any fluid, as discussed in Chap. II. The pipe size required to meet any specified flow conditions and pressure drops may be calculated by means of the formulas on pages 81 to 137. Data on density of refrigerant vapors and liquids[1] are given in Table II, viscosities of refrigerant vapors and liquids are given in Figs. 3 to 11, and acceptable velocities of flow are discussed in succeeding paragraphs. Viscosities of the Freon gases were obtained at a pressure of 1 atmosphere, except for Freon-113 which was secured at 0.1 atmosphere. Where available, viscosity at the operating pressure should be used. Although viscosity data are incomplete, charts for the more commonly used refrigerants are contained herein.

Discharge Lines.—Pipe lines should be sized so as to keep the pressure drop within limits that are commensurate with reasonable pipe sizes. High pressure drop in discharge lines increases the power consumption of the compressor since the discharge pressure

[1] Complete tables of properties of commonly used refrigerants are contained in the "1942 Refrigerating Data Book" published by the American Society of Refrigerating Engineers, 50 West 40th St., New York 18, N.Y.

must be increased to provide for the high pressure drop to the condenser, if the desired refrigerating capacity is to be maintained.

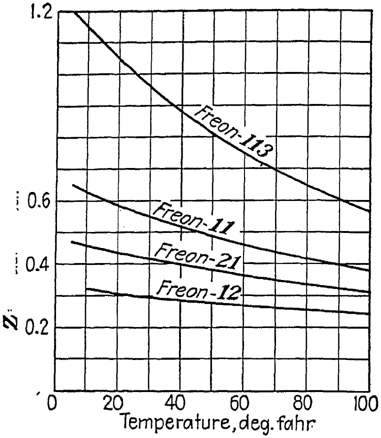


FIG. 3.—Temperature *vs.* absolute viscosity, *z*, in centipoises at saturation pressure of Freon refrigerant liquids. (*Reproduced by permission from "Viscosities of Freon Refrigerants," by A. F. Benning and W. H. Markwood, Jr., Refrig. Eng., Vol.* 37, *April,* 1939.)

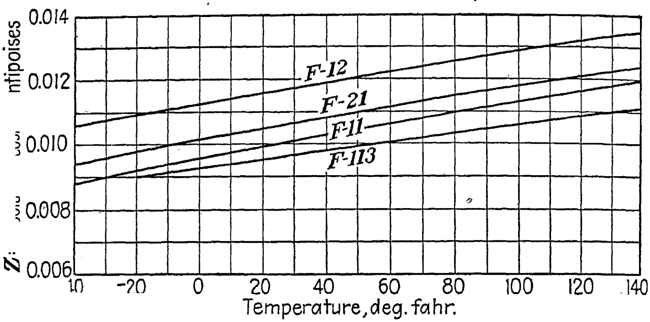


FIG. 4.—Temperature *vs.* absolute viscosity, *z*, in centipoises, of Freon refrigerant gases, at 1 atmosphere pressure. (*F*-113 at 0.1 atmosphere). (*Reproduced by permission from "Viscosities of Freon Refrigerants," by A. F. Benning and W. H. Markwood, Jr., Refrig. Eng., Vol.* 37, *April,* 1939.)

This offsets the reduced investment incident to the smaller line. In general, a total pressure drop of 2 to 4 psi will be found satisfactory in connection with a velocity of flow of 2,000 to 3,500 ft

per min. By total pressure drop is meant the total system loss including that in pipe, valves, fittings, and coils. The loss of head in any piping due to flow through fittings and valves often constitutes an appreciable percentage of the total pressure drop. This pressure drop usually is converted to the equivalent drop in

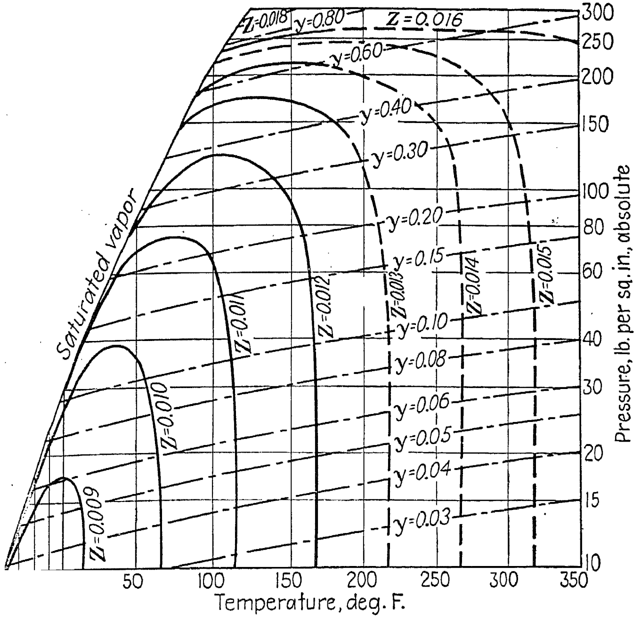


FIG. 5.—Absolute viscosity, z, in centipoises and density, y, in lb per cu ft of ammonia vapor at different pressures and temperatures. (*Reproduced by permission from the "Refrigerating Data Book," published by the American Society of Refrigerating Engineers,* 1942.)

straight pipe of the same size and is added to the length of pipe to secure the total equivalent length of pipe to be used in determining the pressure drop. For pressure drop through valves and fittings, see Table XIV, page 100. For a discussion of pressure drop in hot-gas discharge lines, see Chap. VI of the "Trane Air Conditioning Manual," published by the Trane Company.

Liquid Lines.—A total pressure drop of not to exceed 5 psi may be used to size liquid lines which should be chosen so as to limit

TABLE II.—DENSITIES AND SATURATION PRESSURES OF REFRIGERANT VAPORS AND LIQUIDS*

Refrigerant†	Density, lb per cu ft at saturation pressure‡			Saturation pressure,‡ psi gauge	
	Vapor		Liquid		
	5 F	86 F	86 F	5 F	86 F
Ammonia	0.1227	0.5643	37.20	19.57	154.5
Carbon dioxide	3.741	21.09	37.40	319.7	1024.3
Trichloromonofluoromethane (Freon-11)	0.0815	0.4461	91.38	23.95§	3.58
Dichlorodifluoromethane (Freon-12)	0.6735	2.569	80.63	11.81	93.2
Dichloromonofluoromethane (Freon-21)	0.1095	0.577	84.52	19.25§	16.53
Trichlorotrifluoroethane (Freon-113)	0.0370	0.257	96.96	27.92§	13.93§
Isobutane	0.1560	0.658	34.10	3.3§	44.8
Methyl chloride	0.2209	0.930	55.80	6.19	80.83
Methylene chloride	0.2020	0.150	83.30	27.53§	9.44§
Methyl formate	0.0212	0.141	61.00	26.34§	1.84§
Sulphur dioxide	0.1558	0.844	84.40	5.87§	51.75

* From "Properties and Characteristics of Refrigerants," by R. J. Thompson. Reproduced by permission from *Refrig. Eng.*, November. 1942, published by the American Society of Refrigerating Engineers. For complete data on properties of refrigerants, see the "1942 Refrigerating Data Book" published by the American Society of Refrigerating Engineers, 50 West 40th St., New York 18, N.Y.

† For chemical formulas see Table I. Formula for Freon-113 is $C_2Cl_3F_3$.

‡ Saturation pressures corresponding to standard ton conditions of 5 F evaporator and 86 F condenser temperatures. For further information see references of footnote *.

§ Inches of mercury below atmospheric pressure.

the velocity of flow to 2 to 4 ft per sec. A higher pressure drop will invite the possibility of vaporizing a small part of the liquid refrigerant in the pipe line before it reaches the expansion valve. Although this does not affect the capacity of the refrigeration system, it does decrease the capacity of the expansion valve. A further disadvantage of high pressure drop in the liquid line is that it reduces the pressure differential across the expansion valve and also reduces the capacity of the expansion valve.[1]

If the evaporator is located one or more floors above the liquid receiver, the liquid pressure will decrease in the liquid line riser as the elevation increases owing to the static weight of the liquid. The liquid pressure at any height ahead of the evaporator control valves on any floor, minus the evaporator or low side pressure, equals the pressure differential available across the control valves. The control valves must be selected to have sufficient capacity at

[1] For further data on sizing liquid lines, see "Pressure Drop in Liquid Refrigerating Lines," by E. Gygax and K. S. Willson, *Refrig. Eng.*, February, 1940.

this differential to take care of the contemplated load. As the liquid rises in the liquid riser, a part of it will boil off into gas owing to the decrease in pressure corresponding to the diminished static head. Thus the control valve has to pass both liquid and gas in which the latter may represent as much as 40 per cent of the volume of the mixture. The control valve must be sized so that an adequate quantity of the mixture will be passed to provide for the anticipated load.[1]

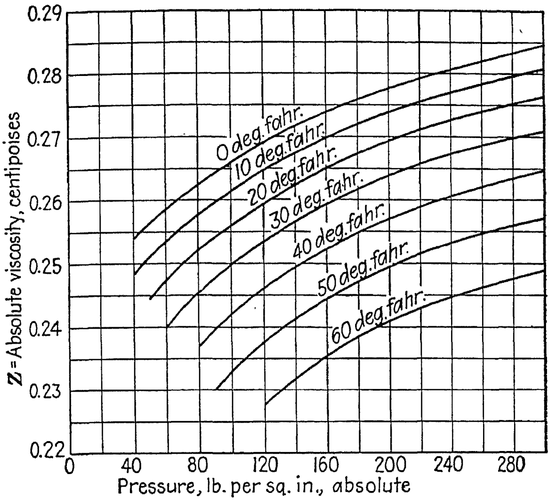


FIG. 6.—Absolute viscosity, z, in centipoises of liquid ammonia at different pressures and temperatures. (*Based on data from the "Refrigerating Data Book," published by the American Society of Refrigerating Engineers*, 1942.)

Where permitted by local codes and if the layout is favorable, the condenser may be located on the top floor and the liquid line run down to evaporators and valves situated below. The refrigerant pressure thus increases in dropping down the liquid riser, which permits use of smaller control valves to secure the same capacity.

Suction Lines.—Proper sizing of the suction line should be carefully considered since excessive pressure drop in this line will decrease the refrigerating capacity of the compressor. The total

[1] See "New Aids in Refrigeration Piping Design," by R. C. Doremus, *Heating, Piping and Air Conditioning*, Vol. 9, pp. 492, 603, 725, 1937; Vol. 10, p. 105, 1938.

pressure drop in the suction line should not, in general, exceed 1 to 3 psi and preferably should be within the lower figure, although

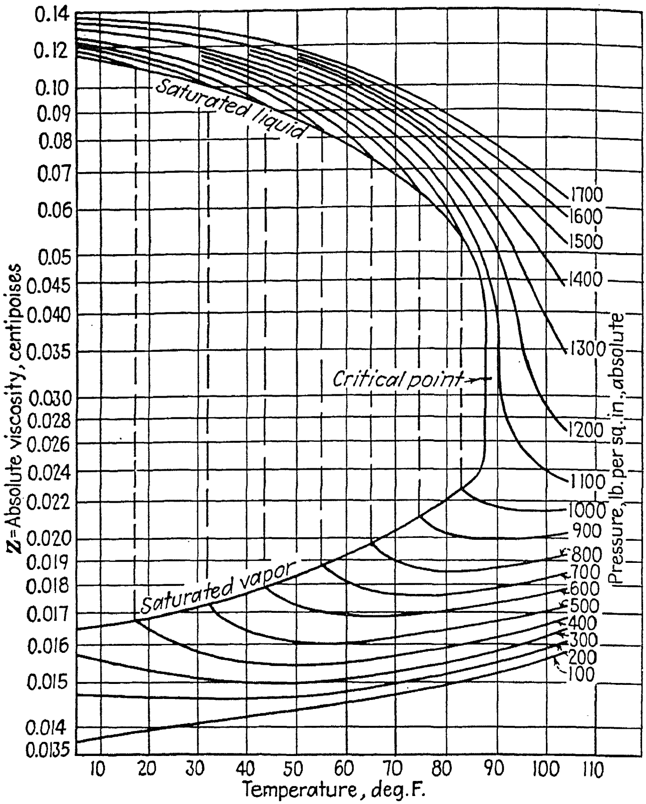


FIG. 7.—Absolute viscosity, z, in centipoises of carbon-dioxide liquid and vapor at different pressures and temperatures. (*Reproduced by permission from the "Refrigerating Data Book," published by the American Society of Refrigerating Engineers,* 1942.)

for small plants where the power requirements are small, a pressure drop corresponding to the higher figure can be tolerated. In using the higher pressure drop, a compressor should be selected that will produce the necessary capacity with the reduced pressure

existing at its suction intake. Sizing suction lines on the small side is unwise because, if the capacity of the compressor proves to be too small owing to the high pressure drop, it is necessary to secure a new compressor or speed up the existing one. The latter alternative may prove expensive in the long run owing to higher power costs and increased maintenance. Use of pipe lines too

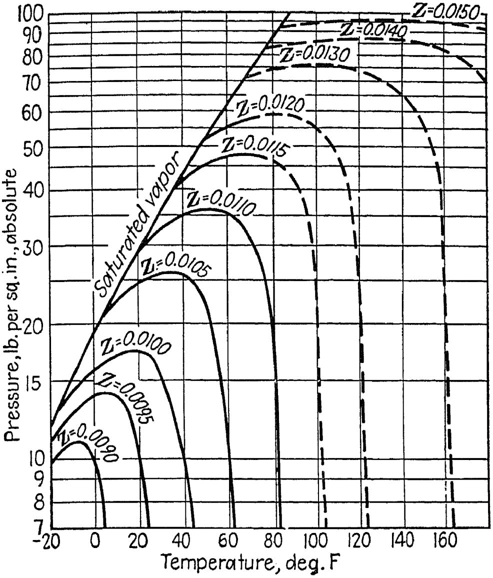


FIG. 8.—Absolute viscosity, z, in centipoises of methyl chloride vapor at different pressures and temperatures. (*Reproduced by permission from the "Refrigerating Data Book," published by the American Society of Refrigerating Engineers,* 1942.)

large should be avoided when using oil-soluble refrigerants because the velocity of the refrigerant may be too low to carry back the entrained oil. This points the need for careful sizing of the suction lines. Velocities of 1,000 to 2,000 feet per minute are common in suction lines, although this varies considerably with the suction pressure, the particular refrigerant, and the diameter of the pipe.[1]

Effect of Lubricating Oil.—It is customary to include lubricating oil with the refrigerant to provide lubrication particularly for the

[1] See "New Aids in Refrigeration Piping Design," *op. cit.*

compressor. The addition of the oil, which is extremely viscous as compared to the refrigerant, will increase the pressure drop of the refrigerant-oil mixture over that shown by a pure refrigerant. The pressure drop of the mixture may be approximated by assuming that the viscosities of the oil and the refrigerant are additive by weight. For accurate results, viscosities should be secured for the oil at the operating temperature and may be converted to centipoises as discussed on pages 118 to 124. In determining the

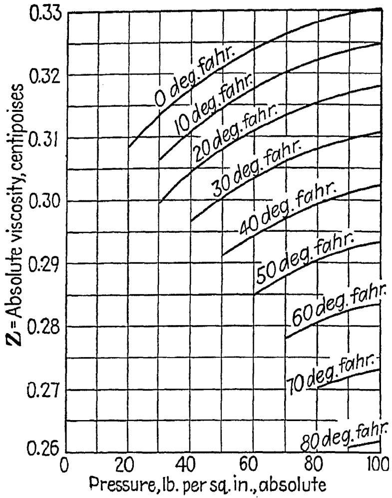


FIG. 9.—Absolute viscosity, z, in centipoises of liquid methyl chloride at different pressures and temperatures. (*Based on data from the "Refrigerating Data Book," published by the American Society of Refrigerating Engineers*, 1942.)

refrigerative effect of a refrigerant-oil mixture, or the volume of the refrigerant in cubic feet per pound, it should be remembered that the oil will have no refrigerating effect.[1] The effect of oil in the refrigerant is important in proportioning liquid lines. In suction lines, because of the increase in volume of the refrigerant on vaporizing, the volume of oil compared to refrigerant gas is small and may safely be neglected as regards pressure drop. In planning the piping for refrigerants in which the lubricating oils are readily miscible, such as the Freons and methyl chloride,

[1] See "Pressure Drop in Liquid Refrigerating Lines," *op. cit.*

adequate provision for recovery of the oil is essential. It is important to avoid traps in .he liquid and suction lines and evaporators which will collect the oil resulting in a decrease in the oil supply in the compressor to an extent where the latter may be damaged as a result of inadequate lubrication.[1]

In the case of ammonia, sulphur dioxide, and carbon dioxide, these refrigerants are relatively immiscible with mineral lubricating

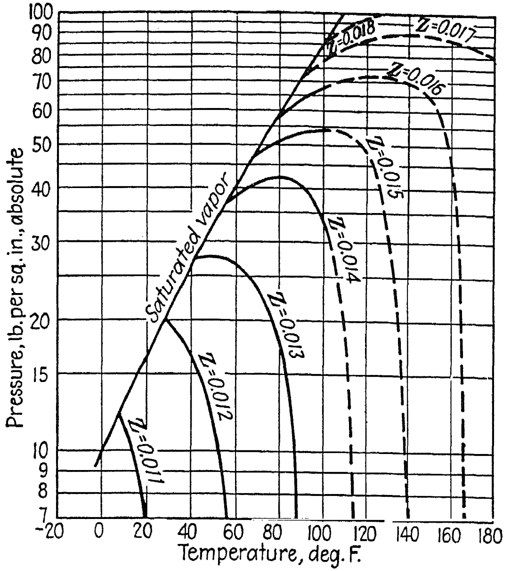


FIG. 10.—Absolute viscosity, z, in centipoises of sulfur-dioxide vapor at different pressures and temperatures. (*Reproduced by permission from the "Refrigerating Data Book," published by the American Society of Refrigerating Engineers,* 1942.)

oils, yet the oil is carried along in both gas and liquid lines. Oil traps are installed to remove as much of the oil as possible from the gas discharged from the compressor. Oil passing this trap will flow through the condenser with the liquid into the receiver where it will separate by gravity. If the oil is not removed periodically, it will pass along with the liquid ammonia to the expansion valves

[1] See "New Aids in Refrigeration Piping Design," *op. cit.*, Vol. 9, pp. 492, 603, 1937.

where it may cause trouble by freezing to a grease and interfering with the free passage of the ammonia through the valves.[1]

As a preliminary guide for determining suitable pipe sizes, Tables III-*A*, III-*B*, and III-*C* are included giving tons of refrigeration that will produce a pressure drop of 1 psi when flowing through a 2-in. Schedule 40 steel pipe 100 ft long, including equivalent lengths. Multipliers to secure the same pressure drop in other sizes of pipe and tubing are contained in Table IV. Correction for other pressure drops may be made by using the factors given in Table V.

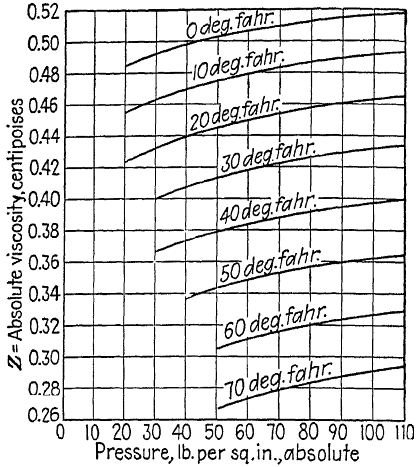


FIG. 11.—Absolute viscosity, z, in centipoises of liquid sulfur dioxide at different pressures and temperatures. (*Based on data from "Refrigerating Data Book," published by the American Society of Refrigerating Engineers,* 1942.)

Example.—As an illustration in the use of these tables, assume that it is desired to know the capacity of a 4-in. Schedule 40 welded-steel pipe 150 ft long carrying saturated ammonia vapor at 40 psi gauge. Total desired pressure drop, 2 psi; six elbows and one globe valve in the line.

Solution.—Total equivalent length in linear feet:

Pipe	150.0
6 elbows at 5.37 (see p. 100)	32.2
1 globe valve at 112 (see p. 101)	112.0
Total	294.2

[1] *Ibid.*, p. 728.

Permissible pressure drop per 100 ft, 2 ÷ 2.942	=	0.680 lb
Multiplier for pressure drop, Table V, by interpolation	=	0.82
Multiplier for 4-in. Schedule 40 pipe, Table IV	=	5.9
Tonnage in 2-in. Schedule 40 pipe for 1-lb pressure drop per 100 lin ft., Table III-*A*, by interpolation	=	41.0 tons
Tonnage in 4-in. Schedule 40 pipe for above conditions, 41.0 × 0.82 × 5.9	=	198.4 tons

By means of a trial-and-error process, the proper pipe size to accommodate a given tonnage requirement may be determined, solving in this case for the pipe size multiplier and thus the pipe size. The above method for sizing of refrigerant piping will give reasonably accurate results commensurate with the variables involved but, where greater accuracy is desired, the pressure drop method described heretofore should be used.

Expansion Valves.—Closely associated with the sizing of liquid lines is providing a suitable expansion valve for regulating the flow of refrigerant so that the evaporator operates at maximum efficiency.[1] Although hand-controlled expansion valves may be used when the load is constant, more accurate control under fluctuating loads is provided by the thermostatic expansion valve which regulates the liquid flow so as to keep the evaporator surface in full use. A remote bulb attached to, or inserted in, the suction line operating on the suction temperature serves to actuate the valve. Expansion valves must be accurately sized, properly installed, and properly adjusted to avoid short cycling of the compressor on one hand or flooding the evaporator with liquid refrigerant entering the suction line as the other extreme. A valve that is too small will produce an excessive pressure drop and a decrease in capacity will result. Too large a valve will open only partly and the consequent wire drawing will damage the seat of the valve. It is considered desirable to select a valve having from 50 to 100 per cent more capacity than the desired amount of refrigeration to provide an adequate opening on normal load and sufficient capacity to handle a reasonable overload. Table VI gives an indication of the quantities of the various refrigerants to be handled by the expansion valve to produce a standard ton of refrigeration. Expansion valves may be secured with screwed or flanged ends and in globe, tee, or angle styles to suit the individual designer's ideas for the particular layout involved.

[1] See "Selection and Application of Thermostatic Expansion Valves for Low-temperature Installations," by R. S. Dawson, *Refrig. Eng.*, October, 1942, p. 222.

TABLE III-*A*.—SATURATED VAPORS. FLOW GIVING 1 PSI PRESSURE DROP PER 100 LIN FT IN 2-IN. SCHEDULE 40 ("STANDARD WEIGHT") STEEL PIPE

Absolute pressure	Ammonia			Freon (F-12)			Methyl chloride			Sulphur dioxide			Carbon dioxide		
	Temp. sat. vapor	Lb per hr	Tons refrig.	Temp. sat. vapor	Lb per hr	Tons refrig.	Temp. sat. vapor	Lb per hr	Tons refrig.	Temp. sat. vapor	Lb per hr	Tons refrig.	Temp. sat. vapor	Lb per hr	Tons refrig.
In. Hg															
5	−83.0	219	8.0												
10	−63.7	310	11.6	......	.....	....	......	.....	....	−26.2	593	6.9			
15	−51.3	380	14.4	......	.....	....	......	.....	....	−12.5	727	8.5			
20	−41.9	439	16.7	−38.0	1,238	4.8	......	.....	....	− 2.0	840	9.9			
25	−34.3	490	18.8	−29.0	1,383	5.4	−18.5	838	10.3	6.6	945	11.1			
Psi															
15	−27.3	540	20.8	−20.8	1,525	6.1	−10.0	927	11.4	14.4	1,040	12.3			
20	−16.6	623	24.2	− 8.2	1,753	7.2	2.9	1,073	13.4	26.4	1,202	14.3			
25	− 8.0	696	27.2	2.2	1,953	8.3	13.3	1,204	15.2	36.3	1,345	16.0			
30	− 0.6	762	30.0	11.1	2,135	9.2	22.3	1,325	16.9	44.8	1,472	17.6			
35	5.9	822	32.5	18.9	2,305	10.1	29.7	1,430	18.3	52.2	1,595	19.1			
40	11.7	877	34.8	25.9	2,465	11.0	36.6	1,535	19.8	58.8	1,707	20.4			
50	21.7	980	39.2	38.3	2,760	12.6	48.4	1,720	22.4	70.4	1,915	22.8			
60	30.2	1,072	43.1	48.7	3,020	14.1	58.6	1,892	24.8	80.3	2,095	24.9			
80	44.4	1,239	50.0	66.3	3,495	16.8	76.1	2,220	29.4	96.9	2,445	....	−67.4	2,320	10.2
100	56.0	1,386	56.2	89.0	3,925	19.3	88.7	2,440	....	110.2	2,750	....	−57.6	2,610	11.6
120	66.0	1,520	62.0	93.4	4,320	....	101	2,655	....	121.5	3,040	....	−49.3	2,865	12.8
140	74.8	1,643	67.2	104.5	4,710	....	111	2,805	....	131.6	3,310	....	−42.0	3,100	14.0
160	82.6	1,760	72.2	114.5	5,070	....	120	2,910	....	......	.....	....	−35.4	3,320	15.1
180	89.8	1,868	....	123.7	5,420	....	128	3,140	....	......	.....	....	−29.4	3,530	16.1
200	96.3	1,970	....	132.1	5,770	....	140	3,325	....	......	.....	....	−23.8	3,730	17.1
250	110.8	2,195	....	......	.....	....	......	.....	....	......	.....	....	−11.6	4,190	19.3
300	123.2	2,390	....	......	.....	....	......	.....	....	......	.....	....	− 1.1	4,630	21.5
400	......	.....	....	......	.....	....	......	.....	....	......	.....	....	16.6	5,470	25.2
500	......	.....	....	......	.....	....	......	.....	....	......	.....	....	31.3	6,240	28.2
600	......	.....	....	......	.....	....	......	.....	....	......	.....	....	43.9	7,030	30.9
700	......	.....	....	......	.....	....	......	.....	....	......	.....	....	55.0	7,830	32.8
800	......	.....	....	......	.....	....	......	.....	....	......	.....	....	65.0	8,670	33.1
900	......	.....	....	......	.....	....	......	.....	....	......	.....	....	74.2	9,670	33.1
1,000	......	.....	....	......	.....	....	......	.....	....	......	.....	....	82.6	11,100	30.0

Reproduced by permission from the "1942 Refrigerating Data Book," published by the American Society of Refrigerating Engineers.
"Lb per hr" is flow in 2-in. Schedule 40 steel pipe with 1 psi pressure drop per 100 lin ft.
"Tons refrig." is based on 86 F liquid up to expansion valve.
Tables III-*A*, III-*B*, III-*C*, were prepared by M. B. Golber and are based on viscosity and density data and formulation appearing in the 1939–1940 "Refrigerating Data Book."

TABLE III-*B*.—SUPERHEATED VAPORS. FLOW GIVING 1 PSI PRESSURE DROP PER 100 LIN FT IN 2-IN. SCHEDULE 40 ("STANDARD WEIGHT") STEEL PIPE

Absolute pressure	Ammonia			Freon (F-12)			Methyl chloride			Sulphur dioxide			Carbon dioxide		
	Temp. comp. vapor	Lb per hr	Tons refrig.	Temp. comp. vapor	Lb per hr	Tons refrig.	Temp. comp. vapor	Lb per hr	Tons refrig.	Temp. comp. vapor	Lb per hr	Tons refrig.	Temp. comp. vapor	Lb per hr	Tons refrig.
In. Hg															
25	...	.....	....	...	.....	....	...	.....	....	8	938	13.1			
Psi															
15	...	.....	....	...	.....	....	...	.....	....	28	1,021	14.1			
20	...	.....	....	...	.....	....	...	.....	....	57	1,144	15.4			
25	...	.....	....	...	.....	....	25	1,203	17.8	81	1,254	16.6			
30	...	.....	....	13	568	12.1	45	1,283	18.6	98	1,325	17.2			
35	7	822	38.6	22	601	12.7	65	1,367	19.5	115	1,413	18.0			
40	22	869	40.4	31	636	13.3	75	1,448	20.4	132	1,522	19.1			
50	47	965	44.0	45	691	14.1	100	1,612	22.1	158	1,667	20.4			
60	69	1,027	46.0	58	747	14.8	125	1,760	23.6	179	1,835	21.9			
80	105	1,153	50.1	78	833	16.0	155	1,990	25.6	215	2,030	23.3			
100	135	1,260	53.3	94	916	16.8	180	2,215	27.5	234	2,210	24.5			
120	160	1,354	56.1	109	987	17.2	200	2,430	29.2	265	2,405	26.0			
140	182	1,440	58.5	121	1,050	17.7	220	2,550	29.8	284	2,550	26.8			
160	201	1,520	60.6	133	1,110	18.0	235	2,680	30.5						
180	219	1,595	62.5	143	1,170	18.1	250	2,805	30.9						
200	235	1,660	64.0	153	1,227	18.2	270	2,940	29.8						
250	270	1,820	67.5												
300	300	1,955	70.1												
400	...	.....	....	...	.....	....	...	.....	....	...	.....	....	28	5,330	49.5
500	...	.....	....	...	.....	....	...	.....	....	...	.....	....	54	5,860	50.1
600	...	.....	....	...	.....	....	...	.....	....	...	.....	....	82	6,430	50.7
700	...	.....	....	...	.....	....	...	.....	....	...	.....	....	98	6,750	48.8
800	...	.....	....	...	.....	....	...	.....	....	...	.....	....	118	7,100	47.0
900	...	.....	....	...	.....	....	...	.....	....	...	.....	....	133	7,500	44.5
1,000	...	.....	....	...	.....	....	...	.....	....	...	.....	....	148	7,750	39.7

Reproduced by permission from the "1942 Refrigerating Data Book," published by the American Society of Refrigerating Engineers.
Temperature of compressed vapor based upon isentropic compression from saturated vapor at 5 F.
Tons of refrigeration based upon liquid at temperature corresponding to pressure of saturated vapor at 5 F.

TABLE III-*C*.—LIQUID FLOW GIVING 1 PSI PRESSURE DROP PER 100 LIN FT IN 2-IN. SCHEDULE 40 ("STANDARD WEIGHT") STEEL PIPE

Fluid	Temp., deg F	Density, lb per cu ft	Flow, lb per hr	Tons[1]
Ammonia.......	86	37.2	14,100	556
Carbon dioxide..	86	37.4	17,900	82.8
Freon (F-12)....	86	80.6	21,000	89.3
Methyl chloride.	86	56.3	17,400	218
Sulphur dioxide		84.4	21,300	251

[1] Tons refrigeration based on 5 F evaporation, saturated gas, and 86 F liquid.

Reproduced by permission from the "1942 Refrigerating Data Book," published by the American Society of Refrigerating Engineers.

TABLE IV.—VALUES OF MULTIPLIERS TO BE APPLIED TO FLOW IN 2-IN. SCHEDULE 40 STEEL PIPE TO FIND FLOW GIVING SAME PRESSURE DROP FOR OTHER PIPE SIZES, WEIGHTS, AND MATERIALS

Nominal size, inches	Steel or wrought iron		Brass		Copper tubing		
	Schedule 40, standard weight	Schedule 80, extra strong	Regular	Extra strong	Type *K*	Type *L*	Type *M*
1/8	0.0041	0.0022	0.0052	0.0022	0.00175	0.0021	0.0021
1/4	0.0090	0.0055	0.0115	0.0059	0.0068	0.0071	0.0077
3/8	0.021	0.0138	0.025	0.0157	0.0138	0.0166	0.0188
1/2	0.039	0.028	0.047	0.032	0.029	0.032	0.036
5/8	....	......	....	......	0.052	0.055	0.060
3/4	0.086	0.064	0.098	0.073	0.075	0.086	0.094
1	0.164	0.127	0.193	0.144	0.162	0.176	0.190
1 1/4	0.33	0.27	0.38	0.31	0.29	0.30	0.32
1 1/2	0.51	0.42	0.57	0.48	0.47	0.49	0.50
2	1.00	0.84	1.13	0.95	0.99	1.02	1.04
2 1/2	1.65	1.37	1.90	1.55	1.77	1.83	1.89
3	2.8	2.4	3.3	2.8	2.8	2.9	3.0
3 1/2	4.2	3.6	4.6	4.2	4.2	4.4	4.5
4	5.9	5.1	6.6	5.8	6.0	6.2	6.3
5	10.7	9.5	12.1	10.6	10.6	11.0	11.2
6	17.5	15.3	19.9	16.9	16.8	17.6	17.9
8	36	32	40	35	35	36	37
10	67	58	71	66	61	64	66
12	108	93	114	......	97	103	104

Reproduced by permission from the "1942 Refrigerating Data Book," published by the American Society of Refrigerating Engineers.

TABLE V.—VALUES OF MULTIPLIERS FOR FLOW TO GIVE PRESSURE DROPS OTHER THAN 1 PSI PER 100 LIN FT

Pressure drop, psi	0.0	0.2	0.4	0.6	0.8
0	0.00	0.42	0.61	0.77	0.89
1	1.00	1.10	1.20	1.29	1.37
2	1.45	1.52	1.60	1.67	1.73
3	1.80	1.86	1.92	1.98	2.04
4	2.10	2.15	2.21	2.26	2.31
5	2.36	2.41	2.46	2.51	2.56

Reproduced by permission from the "1942 Refrigerating Data Book," published by the American Society of Refrigerating Engineers.

Water Piping.—A complete refrigeration system is likely to require water piping for condenser water, cooling tower or evaporator condenser make-up, chilled water, etc. Piping used in connection with air-conditioning systems may convey in addition well water, recirculated water, water supply for air washing and humidifying etc. If a closed piping system is involved and the system is relatively simple, it is customary to size the pipe on the basis of a water velocity of 3 to 10 ft per sec. Where it is necessary to keep within a specified pressure drop, piping must be sized accordingly. Rules for determining pressure drop for various pipe sizes and rates of water flow are given on pages 269 to 289.

TABLE VI.—QUANTITY OF DIFFERENT LIQUID REFRIGERANTS REQUIRED THROUGH REGULATING VALVE[1] PER TON OF REFRIGERATION

Refrigerant	Btu refrigerating effect per pound, standard ton[2]	Pounds refrigerant per minute	Cubic inches refrigerant per minute
Ammonia	474.45	0.4215	19.60
Sulphur dioxide	141.37	1.414	28.90
Methyl formate	189.23	1.056	29.93
Ethyl chloride	134.05	1.492	30.88
Methylene chloride	148.7	1.345	41.70
Freon-21	89.41	2.237	45.73
Freon-11	67.54	2.961	55.98
Freon-113	53.67	3.726	66.46
Freon-12	51.07	3.916	83.90
Isobutane	111.5	1.794	91.10
Carbon dioxide	56.69	3.528	162.80

[1] From "Properties and Characteristics of Refrigerants," by R. J. Thompson. Reproduced by permission from *Refrig. Eng.*, November, 1942, published by the American Society of Refrigerating Engineers.

[2] Standard ton conditions: 5 F evaporator and 86 F condenser temperatures, with 90 F superheat of vapor and 90 F subcooling of liquid. For further information see reference of footnote 1.

For large water piping systems, the piping cost should be evaluated against the annual cost of pumping.

Brine Piping.—For a number of applications, brine may be used for indirect refrigeration in which the brine acts as a heat conveyer. In this case, the brine is applied to the material to be cooled, and the refrigerant in turn removes the heat from the brine.

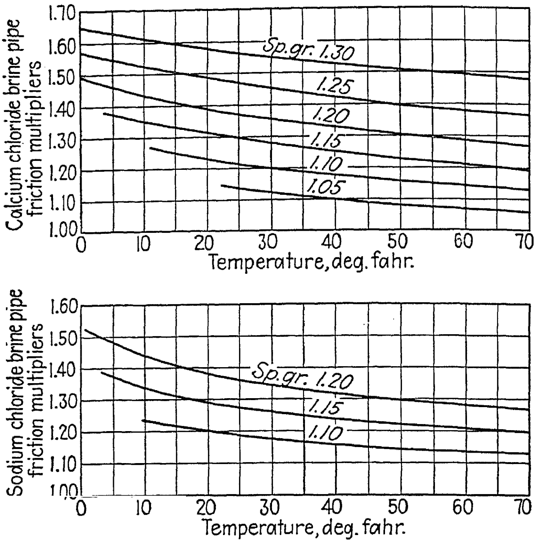


FIG. 12.—Calcium and sodium chloride brine pipe-friction multipliers. (*Reproduced by permission from "Piping for Indirect Refrigeration," by R. C. Doremus, Heating, Piping and Air Conditioning, Vol.* 10, *March and April,* 1938.)

The concentration of the brine solution is determined by the brine temperature desired to secure the necessary refrigerating effect. Usually the brine temperature must be 10 to 20 F below the coldest item to be refrigerated and the refrigerant temperature must be correspondingly lower than the brine. Sufficient concentration may be secured to obtain freezing points of the brine solutions down to -50 F. Gravities and freezing points of brines of various concentrations are available.[1]

[1] See "Piping for Indirect Refrigeration," by R. C. Doremus, *Heating, Piping and Air Conditioning,* March and April, 1938, Vol. 10, pp. 186, 251.

The frictional resistance to flow of sodium and calcium-chloride brines is higher than for water since their densities are greater. A convenient means for estimating brine friction losses for the flow of brines is to apply the brine friction multipliers to water charts and tables such as those given on pages 272 to 284. These factors are given in Fig. 12 based on water at 60 F as unity. For an accurate determination of friction losses, the rational flow formulas of Chap II should be employed. (See pages 81 to 137.) Densities and viscosities for brine solutions are given in Figs. 20 and 21 on pages 129 and 130.

In brine-circulating systems, it is economical to select pipe sizes so as to provide a velocity of 3 to 8 ft per sec. Higher velocities will increase pumping costs because of the higher frictional loss, whereas lower velocities will give large pipe sizes with consequent high initial cost of pipe.

Pipe Wall Thickness.—The proper thickness of pipe wall may be determined by using the wall thickness formula (16*e*) for power piping on page 42 of this handbook, which is the formula desig-

TABLE VII.—ALLOWABLE *S* VALUES TO BE USED FOR PIPE IN REFRIGERATING SYSTEMS
(Table 42 of ASA B31.1-1942)

Material	Specification	Value of *S*,
Steel pipe, seamless, Grade *A*.	ASTM A 53 ASTM A 120 ASTM A 106 or API 5L	12,000
Steel pipe, seamless, Grade *B*.....	ASTM A 53 ASTM A 106	15,000
Steel pipe, lap welded............	ASTM A 53 or ASTM A 106 or ASTM A 120 or API 5L	9,000
Steel pipe, butt welded..............................	ASTM A 53 or ASTM A 120	6,800
Steel pipe, electric resistance welded*....................	ASTM A 135	10,200
Wrought iron, lap welded	ASTM A 72 or API 5L	8,000
Wrought iron, butt welded.	ASTM A 72	6,000
Cast iron pipe, pit cast.	ASTM A 44, ASA A21.2	4,000
Cast iron, centrifugally cast or cast horizontally in green sand molds	FSEC WW-P-421	6,000
Brass pipe, seamless red brass.........................	ASTM B 43	4,700
Copper pipe, seamless	ASTM B 42	4,000
Copper tubing, seamless...........................	ASTM B 88, or B 68	4,000

* Where pipe furnished under this classification is subjected to supplemental tests and/or heat treatments as agreed to by the supplier and the purchaser, whereby such supplemental tests and/or heat treatments demonstrate the strength characteristics of the weld to be equal to the minimum tensile strength specified for the pipe, the *S* value is equal to the corresponding seamless

nated in the Refrigeration Section of the Code for Pressure Piping for securing wall thickness of refrigerating piping. The values of allowable stress for use in this formula are given in Table VII. Factors of safety and allowance for joint efficiency of welded pipe have been taken into account.

In the case of cast-iron pipe, the minimum values of the water-hammer allowance to be added to the working pressure are given in Table VIII on page 47 of this handbook. In no case should the pressure for computing pipe wall thickness be taken less than 100 psi.

The following limitations in use of this formula are specified in the Code for Pressure Piping:

(*a*) Standard iron-pipe-size copper, or red brass (not less than 80 per cent copper) may be used but, if threaded, the wall thickness shall not be less than standard-weight thickness as specified in ASTM Specifications B42 and B43 (see page 473).

(*b*) Where corrosion is not a factor and no metal is removed in making the joint as for flared or soldered type joints, for instance, a zero *C* factor has been assigned. To provide adequate mechanical strength, however, limitations on minimum wall thickness have been designated as follows:

For plain end non-ferrous pipe or tubing, minimum wall thicknesses shall be as follows: for nominal sizes up to 1 in., the nominal thickness for field erection shall not be less than specified for Type *L*, and for factory assembly, not less than specified for Type *M* in ASTM Specification B88; for nominal sizes 1 in. and larger, the nominal wall thickness shall in no case be less than 0.049 in. Additional wall thickness shall be provided as required to care for corrosion, erosion, or mechanical strength. Soft annealed copper tubing used for refrigerant piping erected on the premises shall not be used in sizes larger than ¾ in. nominal size, and shall conform to Types *K* or *L* of ASTM B88. Copper tubing used for refrigerant piping erected on the premises shall conform to ASTM B88, Types *K* or *L* for dimensions, and shall be absolutely free from scale and dirt.

Standard-weight (Schedule 40) steel or wrought-iron pipe may be used for design working pressures not exceeding 250 psi, provided lap-welded or seamless pipe is used for sizes larger than 2 in., and extra-strong (Schedule 80) pipe is used for liquid lines for 1½-in. size and smaller.

Pipe Joints.—According to the Piping Code, joints in *steel or wrought-iron pipe* may be any of the commonly accepted types, screwed, flanged, or welded. The facing of flanged joints, however, is generally limited to the retained gasket type, that is, male-and-female, tongue-and-groove, or ring types. The Code for Pressure Piping limits the use of screwed joints to sizes 4 in. and smaller for brine piping, and 3 in. and smaller for refrigerant piping where the design pressure is 250 psi or below, or 1¼ in. and smaller where the design pressure is above 250 psi. Steel

pipe lighter than standard weight or Schedule 40 of ASA B36.10 shall not be threaded. In all cases where pipe is threaded, exposed threads shall be tinned or otherwise coated to prevent corrosion. Flanges attached to refrigerant pipe in sizes in excess of those given above must be steel and should be welded to the pipe to ensure a pressuretight joint.

Welding of steel pipe joints is displacing screwed and flanged construction to a large extent, particularly for submerged coils or concealed piping where leaks would be difficult to reach. Ammonia pipe fittings made by a number of manufacturers have been available, but the fittings of the several manufacturers are not necessarily interchangeable owing to variations in dimensions between the different makes. An American Standard for flanged ammonia fittings was in process of development a few years ago but was dropped because flanged construction was rapidly being displaced by welded construction, and ammonia was being displaced to a considerable extent by other refrigerants that used copper or brass tubing with flared or solder fittings. Welding fittings of the butt-welding type for the larger sizes and the socket-welding type for the smaller lines facilitate erection of pipe by welding. Dimension of American Standard steel butt-welding fittings are given on pages 502 and 505, and for the proposed American Standard for socket-welding fittings on pages 505 to 507. Dimensions for 150- and 300-lb steel welding-neck and slip-on welding flanges are given on pages 633 to 647.

According to the Piping Code, joints for *nonferrous pipe* may be screwed, flanged, soldered (sweated capillary), or of the flared compression type. Socket-sleeve brazed joints also are commonly used. Threaded joints for refrigerant pipe using copper pipe of standard pipe size are subject to the same size limitations as are steel or wrought-iron pipe. For screwed brass or copper pipe, extra-heavy fittings are required. For service where the temperatures do not exceed 250 F, soldered joints of the socket-sleeve type in accordance with the American Standard Soldered Joint Fittings, ASA A40.3, are acceptable. Dimensions of those fittings are given on page 589. Fittings used in soldered joints for refrigerant pipe are limited by the Piping Code to cast red brass, die-pressed, wrought, or extruded brass or copper. Sweated capillary joints on hard-drawn brass or copper may be made with an alloy having a melting point greater than 1000 F, or with a solder having a melting point below 500 F but above 350 F.

There are several soft solders available having melting points within this range, but a 50-50 lead tin and a 95-5 tin antimony are most widely used for refrigerant piping.

Flared compression fittings for refrigerant pipe in accordance with the American Standard for Brass Fittings for Flared Copper Tubes, ASA A40.2, dimensions for which are given on page 588, may be used for joints on annealed copper tubing for sizes not to exceed ¾ in. in outside diameter, providing all fittings are exposed for visual inspection. Flared-tube fittings for handling refrigerants are made of forged brass and are widely used in domestic and small commercial installations.[1]

Fittings of the socket-sleeve type are acceptable, using a brazing solder in which the copper plus silver shall not be less than 60 per cent and having a melting point above 1000 F. The brazed junction between pipe and sleeve shall extend the full depth of the socket and no fillet brazed joints should be used because of their inferior strength.

Stop Valves.—Extensive requirements for the use of stop valves for refrigerant piping are contained in the Code for Pressure Piping and the American Standard Safety Code for Mechanical Refrigeration, the more important of which are as follows: For refrigerants, stop valves are required to be of the globe, angle, or needle type. Gate and other nonglobe-type valves and cocks shall not be used for refrigerants except in industrial plants which are completely separated from residential, business, and other built-up sections and from public buildings and playgrounds and which have competent operating men constantly in charge. In such locations, gate and other nonglobe types of valves and also constantly lubricated cocks are permitted for vapor and gas pipes of systems using the hydrocarbons or ammonia as the refrigerant.

All refrigerant valves having stuffing boxes or valve stem packing must be of the backseating type to permit repacking under pressure. Stop valves used with soft annealed copper tubing or hard-drawn copper tubing ¾ in. nominal size or smaller must be securely mounted independent of tubing fastenings or supports. Stop valves placed where it is not obvious what they control must be suitably labeled. The latter provision is a wise precaution for any type of piping.

[1] For standard dimensions of refrigeration-type flared-tube fittings, see "SAE Handbook" published by the Society of Automotive Engineers, 29 West 39th St., New York 18, N.Y.

Expansion and Supports.—Although the temperature differentials encountered in refrigeration piping are not of so great magnitude as in some other piping, such as for high-temperature steam lines, expansion and contraction of refrigeration lines cannot safely be neglected. Failure to provide properly for this factor may result in distortion and leaks at fittings as well as possible rupture of fittings owing to excessive stresses. Pipe bends, offsets in the pipe lines, or corrugated expansion joints are commonly used to provide the necessary flexibility. Slip-type expansion joints are not permitted by the Code for Pressure Piping because of the hazard involved in possible leakage of the refrigerant through the joint. Pipe bends made of the same material as the pipe line itself, or of other suitable material, are acceptable. Data on the thermal expansion of various piping materials are furnished in Tables I and II on pages 554 and 559.

Refrigerant piping, like any other piping, should be securely supported by means of metal hangers, brackets, straps, clamps, or pedestals so as to avoid transfer of harmful stresses and vibrations to pipe joints. Codes require the use of rigid metal enclosures for soft annealed copper tubing for refrigerant piping erected on the premises, except that flexible metal enclosures may be used at bends or terminals if not exceeding 6 ft in length. Enclosures need not be provided for connections between the condensing unit and the nearest riser box if such connections are not longer than 6 ft.

Location of Piping.—Both the Code for Pressure Piping and the Safety Code for Mechanical Refrigeration specify that the following rules be adhered to in locating refrigerant piping:

Refrigerant piping crossing an open space which affords passageway in any building shall not be less than seven and one-half (7½) ft above the floor unless against the ceiling of such space.

Refrigerant piping shall not be placed in public hallways, lobbies, stairways, elevators, or dumbwaiter shafts, excepting that such refrigerant piping may pass across a public hallway if there be no joints in the section in the public hallway, and provided non-ferrous tubing of one (1) inch nominal outside diameter and less be contained in a rigid metal pipe.

Refrigerant piping, with or without insulation covering, shall be exposed to view, excepting for mechanical protection herein specified, or when located in the cabinet of a Unit System. This does not apply to refrigerant piping installed outside the building or in a flue vented to the outer air.

Tests.—The following *test pressures* for refrigeration piping are specified in the American Standard Code for Pressure Piping and

Code for Mechanical Refrigera-
f the piping by water and for
be conducted with a dry, inert gas. Carbon
dioxide is used frequently.

(*a*) Every refrigerant-containing part of every system shall be tested and proved tight by the manufacturer at not less than the minimu
shown in Table VIII.

TABLE VIII.—TEST PRESSURE
(Table 41, ASA B31.1-1942)

Refrigerant[1]	Chemical formula	Minimum test pressure, psi	
		High-pressure side	Low-pressure side
Ammonia	NH_3	300	150
Butane	C_4H_{10}	90	50
Carbon dioxide	CO_2	1,500	1,000
Freon-12	CCl_2F_2	235	145
Freon-114	$C_2Cl_2F_4$	80	50
Carrene No. 1	CH_2Cl_2	30	30
Freon-21	$CHCl_2F$	70	50
Dichloroethylene	$C_2H_2Cl_2$	30	30
Ethane	C_2H_6	1,100	600
Ethyl chloride	C_2H_5Cl	60	50
Isobutane	$(CH_3)_3CH$	130	75
Methyl chloride	CH_3Cl	215	125
Methyl formate	$HCOOCH_3$	50	50
Propane	C_3H_8	325	210
Sulphur dioxide	SO_2	170	95
Freon-11	CCl_3F	50	30

[1] For chemical names of Carrene and Freon see Table I, p. 1221.

(*b*) For refrigerants not listed in Table VIII, the test pressure for the high-

115 F.

i gage.

(*c*) Every refrigerant-containing part of every system that is erected on the premises, except compressors, safety devices, pressure gages, and control mechanisms, that are factory tested, shall be tested and proved tight after complete installation and before operation at not less than the minimum pressures shown in Table VIII.

(*d*) No oxygen or any combustible gas or combustible mixture of gases shall be used for testing.

(*e*) For refrigerant pipe, soldered joints in pipe or tubing erected on the premises shall remain mechanically intact when subjected to a pull-apart test equivalent to a pressure of not less than 300 psi with a temperature of not less than 800 F, except that this requirement shall not apply to soldered joints in pipe or tubing ½ in. nominal pipe size or smaller when used in systems containing not more than 20 lb of refrigerant.

It is required that a dated declaration of test, signed by the installer and, if present, the inspector, shall be mounted in the machinery room of a refrigeration system.

No universal method for *detecting leaks* of refrigerants is available so that the individual method best suited for the particular refrigerant used should be employed.[1] Ammonia, in addition to its characteristic pungent odor, may be detected and located by burning sulphur sticks in the presence of the refrigerant, which results in the white fog or fumes of ammonium sulphite, or by the alkaline reaction of ammonia fumes on moistened litmus paper or on phenolphthalein. Presence of sulphur dioxide may be detected and the leak located by exposure of a solution of 28 per cent ammonia in water which again results in the white fog of ammonium sulphite.

Owing to their relatively high chemical stability, leaks of many odorless and flammable refrigerants are not easily found. This necessitates use of other means of detection, such as the application of soap and water solutions at suspected points or the observation of oil spots or leaks. Such methods are limited in their application, however, as they are not satisfactory in dark, confined, or inaccessible locations. Freon-vapor leaks may be located readily by means of a halide lamp wherever such leaks are accessible.

MATERIALS AND DIMENSIONAL STANDARDS

A wide variety of materials is used in the construction and installation of refrigeration piping, individual selection depending to a large extent on the nature of the refrigerant employed. Table IX taken from the American Standard Code for Pressure Piping indicates acceptable specifications for use in selecting materials for refrigerant piping. Abstracts of these specifications will be found in Chap. IV.

Pipe.—Pipe and tubing used for refrigeration service may be steel, wrought iron, brass, or copper. Welded or seamless steel is commonly used for ammonia piping and in the larger sizes for other refrigerants. Standard dimensions for steel pipe in accordance with the American Standard for Wrought Iron and Wrought Steel Pipe, ASA B36.10,[2] are given on pages 361 to 371. Copper

[1] See "Properties and Characteristics of Refrigerants," by R. J. Thompson, *Refrig. Eng.*, November, 1942.

[2] Copies are available from the American Society of Mechanical Engineers, 29 West 39th St., New York 18, N.Y.

and red brass (85 per cent copper, 15 per cent zinc) pipe in accordance with ASTM Specifications B42 and B43 are permitted in iron-pipe size dimensions in both regular and extra-strong thicknesses. Dimensions are shown on page 473. American Standard taper pipe threads may be used although solder type fittings are available for use with copper pipe.

TABLE IX.—LIST OF MATERIAL SPECIFICATIONS
(Table 40 of ASA B31.1-1942)

Material	Specification
Bolts:	
Alloy steel	ASTM A96
Commercial steel (bar stock)	ASTM A107
Wrought iron (bar stock)	ASTM A84
Fittings, valves, and flanges:	
Cast iron	ASTM A126, AWWA
Malleable iron	ASTM A197
Steel (cast carbon)	ASTM A27 or A95
Steel (forged)	ASTM A105 or A181
Brass castings	ASTM B62
Bronze castings	ASTM B61
Pipe and tubing:	
Steel, seamless or welded, for coiling or bending	ASTM A53
Steel, seamless or welded, for ordinary uses	ASTM A120
Steel, seamless or welded	API 5L
Steel, seamless or welded	ASTM A106
Steel, electric resistance welded	ASTM A135
Wrought iron, welded	ASTM A72
Wrought iron, welded	API 5L
Cast iron (pit cast)	ASTM A44, ASA A21.2
Cast iron (centrifugally cast or cast horizontally in green sand molds)	FSEC WW-P-421
Brass (seamless)	ASTM B43
Copper pipe (seamless)	ASTM B42
Copper tubing	ASTM B88

Seamless copper tubing with flared or soldered fittings is commonly used in the manufacture and installation of refrigeration equipment except for large installations where steel pipe with welded joints may prove more economical. The primary con-

siderations are to have the system clean and properly dehydrated and to secure tight joints. Two general classifications of copper tubing are made: (1) small-diameter dehydrated copper tubing and (2) copper "water" tube in three types designated as Types *K*, *L*, and *M* in ASTM Specification B88.

The small-diameter dehydrated tubes are acceptable for connections in and around refrigerating equipment, particularly in the smaller sized machines. Joints customarily are made with flared-tube fittings.[1] This small tubing is dehydrated at the mill and the tube ends sealed to prevent entrance of moisture before it is used. It is furnished in a soft temper usually in coils of 50 to 100 ft although longer lengths on reels are available. The most common sizes are as follows, all of which are provided with 0.035-in. wall thickness:

Outside diameter, in.	1/8	3/16	1/4	5/16	3/8	7/16	1/2	5/8	3/4
Weight, lb per ft	0.038	0.065	0.092	0.118	0.145	0.172	0.198	0.251	0.305

Copper water tubing in accordance with ASTM Specification B88 connected with a solder seal joint is particularly useful for conveying refrigerants such as Freon and methyl chloride. The three types available, Types *K*, *L*, and *M*, vary as regards wall thickness, Type *M* being the thinnest. Dimensional data on copper water tubing are given on pages 474 and 476. The soft tubing is regularly sold in 30- to 60-ft coils up to and including 1¼ in. O.D. Twenty-foot straight lengths are furnished in both hard and soft tempers in all sizes.

Pipe Coils.—Pipe coils are used extensively as a means of providing heat-transfer surface. To eliminate the use of cast or wrought fittings involving threaded or welded joints, continuously welded coils have been fabricated. Pipe coils are made in single, double, and triple flat coils, ovals, trombones, cylindrical, pancake, and numerous other shapes and sizes. In designing pipe coils, the weight should not exceed practical limits for erection, and the radii of bends should not be less than experience has proved desirable for the material used. Minimum recommended diameters for pipe bends are indicated in Table X.

[1] For dimensions of SAE refrigeration-type flared-tube fittings, see "SAE Handbook," published by the Society of Automotive Engineers, 29 West 39th St., New York 18, N.Y.

TABLE X.—LIMITING DIMENSIONS FOR STEEL PIPE BENDS, HAIRPINS, AND COILS
(Dimensions are distance on centers across U bend, in.)

Nominal pipe size, inches	Pipe schedule ASA B36.10	Usual dimension, inches	Limiting dimension, inches
1/4	40	2 1/2	2
	80	2	1 3/4
3/8	40	3	2 1/2
	80	2 1/2	2 1/4
1/2	40	3	2 1/2
	80	2 1/2	2 1/4
3/4	40	3 1/2	3
	80	3	2 3/4
1	40	4	3 1/2
	80	3	2 3/4
1 1/4	40	5	5
	80	4	4
2	40	12	6
	80	8	6
2 1/2	40	20	10
3	40	32	16
3 1/2	40	40	20
4	40	44	24
4 1/2	40	48	28
5	40	54	32
6	40	60	40
7	40	72	50
8	40	86	60

Where steel butt-welded fittings are used in making up pipe coils, center-to-center of return bends may be secured from the American Standard for Steel Butt-welding Fittings, ASA B16.9, given on page 502.

Valves and Fittings.—To comply with the Code for Pressure Piping, valves, flanges, and fittings should be of a design and material recommended by the manufacturer for the particular service and may be made of cast iron, malleable iron, bronze or steel castings, hot-forged or drop-forged steel, wrought copper, bronze, or brass. Where cast iron is used, it shall conform to Class *B* gray iron of ASTM Specification A126 (see page 625).

Valves and fittings should be in accordance with the manufacturer's recommendations for the service but the minimum metal thickness in valve or fitting bodies shall not be less than that

specified for the pressure and temperature in the respective American Standards and the Standard Practices of the Manufacturers' Standardization Society of the Valve and Fittings Industry. Where a manufacturer's standards for valve, flanges, and fittings depart from ASA or MSS dimensions but have been demonstrated through long usage to be acceptable, the manufacturer's standards are permitted by the Code for the particular refrigerant service listed by the manufacturer. However, they must be of at least equal strength and tightness and be capable of withstanding the same hydrostatic test requirements as the American Standard for the pressure and temperature. Dimensions of American Standard flanged and screwed cast-iron fittings, screwed malleable-iron fittings, steel-flanged fittings are contained in Chap. IV.

Gaskets.—Gaskets must have a high resistance to attack by oil and refrigerant. Rubber is unsatisfactory as a base for composition gaskets for most refrigerants because of its tendency to swell when in contact with oil and to dissolve in refrigerants such as the Freons. Neoprene and chloroprene have found wide acceptance because of their ability to withstand contact with oil and refrigerant. Pressed fiber, including asbestos with an insoluble binder, and metallic gaskets are considered acceptable for most refrigerants provided the joint design is suitable. Care should be exercised in the selection of compounded materials for other refrigerant parts such as stuffing box and valve stem packing, cotton-covered motor windings, insulating varnishes, etc., so that the effects of the refrigerant are not deleterious.[1]

Bolting.—Welding of steel pipe and solder-type joints for copper pipe have almost completely displaced flanged joints. In some instances, however, it is desirable to provide flange joints so that valves or fittings may be readily removed from the pipe line. Bolts or bolt studs for flanged joints conform to the requirements in the corresponding American flange standard. For design pressures above 300 psi, it is recommended that alloy-steel bolting in accordance with ASTM Specification A96 be used. For lower pressures, commercial carbon-steel bolts made from hot-rolled open-hearth bar stock, Grade 22 of ASTM Specification A107 or better, is acceptable. Although used only infrequently in current practice, the Piping Code also permits wrought-iron bolts

[1] For further data, see "Properties and Characteristics of Refrigerants," *op. cit.*

conforming to ASTM Specification A84. Threads on carbon-steel bolts, bolt studs, and nuts should conform to the coarse-thread series of the American Standard for Screw Threads, ASA B1.1. Alloy-steel bolts, bolt studs, and accompanying nuts are threaded in accordance with the American Standard Screw Threads for High-strength Bolting, ASA B1.4. Carbon-steel bolts should be provided with American Standard regular square or hexagonal heavy boltheads. Nuts should be in accordance with ASTM Specification A194 for material and should have American Standard heavy hexagonal dimensions.

Insulation.—Insulation of cold pipe lines is discussed in Chap. V on Heat Insulation. Insulating materials commonly used are cork, hair felt, rock wool, and wool felt. Cork and rock wool are available in molded shapes for covering pipe and screwed or flanged fittings. Molded sections or half sections are made in standard lengths 3 ft long and in three standard thicknesses as follows: (1) ice water, ranging from 1¼ to 2 in., (2) brine, from 1¾ to 3¼ in., and (3) heavy or special thickness brine, from 3½ to 4 in. Joints are filled with a waterproof cement and seams or crevices are puttied before applying the final coat of asphaltic paint. Hair felt comes in rolls and wool felt is made up in flat sheets protected by layers of waterproof felt. Typical selections of insulation for low-temperature refrigerating and air-conditioning piping are given in Table IX of Chap. V, page 720.[1]

CORROSION AND BRITTLENESS

Materials for the construction and installation of refrigerating systems should be suitable for the refrigerant as regards corrosion by the fluid conveyed, and no material should be used that will deteriorate significantly owing to chemical action of the refrigerant or entrained oil, or the combination of both. Care should be exercised also in selecting materials that will not become unduly brittle at the low temperatures encountered in modern refrigeration practice.

Corrosion.—It is recognized that, when moisture and/or air are present, many of the refrigerants are corrosive to the usual materials and it is assumed in recommending materials in Table

[1] For further data on insulation of cold pipe lines, see "Thickness of Insulation Required to Prevent Sweating on Cold Pipe Lines," by R. N. Hosey, *Heating & Ventilating*, September, 1942. Also, "Specifications of the Industrial Mineral Wool Institute," *Heating, Piping and Air Conditioning*, July, 1942.

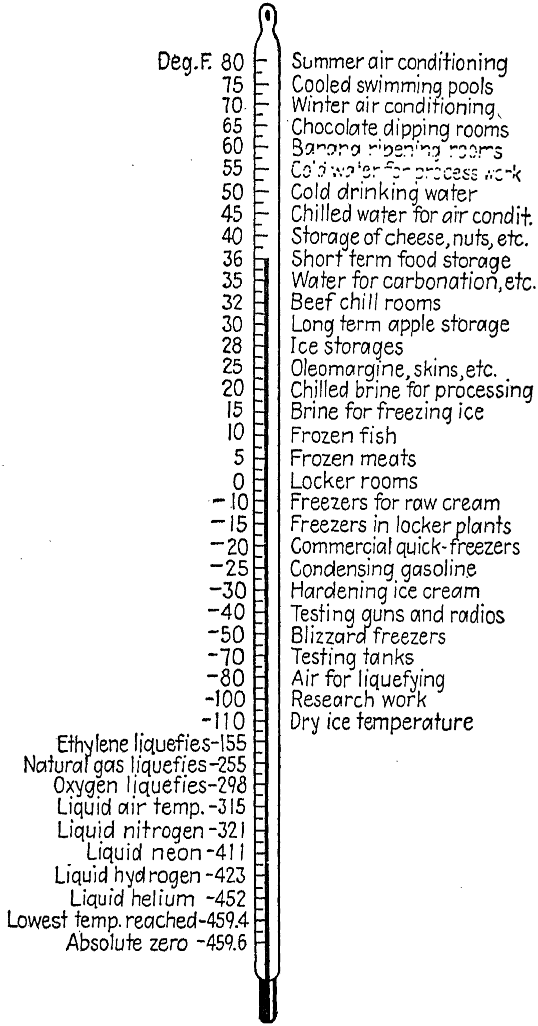


FIG. 13.—Various applications of low-temperature refrigeration. (*Reproduced by permission from "New Demands for Low Temperature Refrigeration" by Cyril Leech, Refrigerating Engineering, November*, 1942.)

IX that refrigeration systems will be charged and operated in accordance with accepted practice to prevent or minimize such corrosion.[1] Care should be exercised to prevent or minimize corrosion by use of materials not significantly corroded by the fluid conveyed or by use of a suitable inhibitor, or both.

Refrigerants themselves are not corrosive to materials to any significant extent but the addition of water vapor and/or air results in a reaction of the refrigerant with water to form a corrosive acid in addition to the corrosive action of water itself. Low solubility of water in a refrigerant is a desirable characteristic of the latter since it is the combination of water with the refrigerant to form an acid or alkali that attacks many of the component metals used in the piping systems. Methyl chloride should not be used with aluminum, zinc, magnesium and its alloys, and die-casting alloys, for instance, as these may be corroded excessively even with but small amounts of water present.[2]

Brines are particularly corrosive, and the use of dissimilar metals such as iron brass, iron copper, etc., should be avoided in the brine piping system since the galvanic action will gradually dissolve the metal forming the positive electrode of the cell. In any case, it is desirable to use a corrosion-retardant chemical. For calcium chloride brine, sodium chromate or dichromate is used. For sodium chloride brine, disodium phosphate in the proper proportions will retard corrosion significantly. In either case, it is desirable to maintain a condition of low alkalinity, a pH value of 7.5 to 8 is acceptable for copper or brass, and 8.0 to 9.0 for iron or steel.[3]

Brittleness at Low Temperature.—One fundamental consideration required in refrigerant piping, which is not pertinent in most other piping, concerns the low temperatures encountered in refrigeration practice. For working temperatures below zero, which are common in refrigerating piping systems as is evident from Fig. 13, the material may become seriously brittle. This must be recognized in the selection and use of materials.

Accordingly, the following allowances are specified in the Code for Pressure Piping:

[1] See Chap. XVII on Corrosion, or refer to

(a) "Scale and Corrosion Control," by J. A. Holmes, *Refrig. Eng.*, March, 1942.

(b) "Corrosion in Refrigerating Plants," *Am. Soc. Refrig. Eng. Circ.* 10.

[2] See "Chemical Effects in Refrigerating Systems," E. W. McGovern, *Refrig. Eng.*, May, 1942.

[3] See "Piping for Indirect Refrigeration," *op. cit.*, Vol. 10, p. 251.

Cast iron, wrought iron, and carbon steel except as they may qualify under the rule for *other materials* shall have the design pressures increased 2 per cent for each degree below 0 F and shall not be used below −60 F.

Copper, brass, and bronze, no adjustment.

Other materials must have, at the lowest working temperature, a Charpy impact value of not less than 15 ft-lb on a specimen 10 mm wide, or 10 ft-lb on a specimen 5 mm wide, or 5 ft-lb on a specimen 2.5 mm wide, as determined in accordance with ASTM Specification E23, using a keyhole notch. Where piping systems using such materials are welded in fabrication, the weld metal also shall meet the above requirements.

Since there is a large and constantly growing list of alloy compositions that can qualify under these requirements, it has not been considered advisable in the Code to infer limitations in their use by designating specific alloys as being particularly suited for low-temperature service. Some carbon steels meet the requirements for a considerable range of temperature if properly heat-treated, 3 to 4 per cent nickel steels are generally acceptable for quite low temperatures, and the 18 per cent chrome 8 per cent nickel steels are satisfactory for very low temperatures. Various other classes of alloys are acceptable but attention is directed to the fact that minor variations in composition and heat treatment may result in major changes in toughness at low temperatures.[1]

[1] See "Impact Resistance and Tensile Properties of Metals at Subatmospheric Temperatures," by the Joint ASME-ASTM Research Committee on the Effect of Temperatures on the Properties of Metals, published by the American Society for Testing Materials, August, 1941.

CHAPTER XVII

CORROSION

This chapter deals with the causes and control of the types of corrosion encountered in piping systems. Generally speaking there are two important types: (1) the external corrosion of pipes buried in soil and (2) the internal corrosion of pipes caused by the fluids that they convey. Although the underlying mechanism is the same in both types, the methods of prevention usually differ. It often is possible to modify the fluids inside of pipes so that they will be less corrosive, but soils must be taken as they come and the pipes must be protected against them.

Following a brief exposition of the mechanism of corrosion, the problems associated with (1) and (2) will be taken up. Attention is given to the important question of protective coatings for buried pipes and to the use of materials other than iron and steel.

MECHANISM OF CORROSION

For a complete discussion of the mechanism of corrosion the reader should consult the extensive literature on this subject.[1] There are numerous reference works available concerning the many phases of corrosion. A bibliography has been compiled by the American Coordinating Committee on Corrosion, George H. Young, Secretary, Mellon Institute, Pittsburgh, Pa. The following paragraphs giving a brief summary of corrosion theory may be a useful introduction to the subject.

Electrochemical Theory.—Several basic theories have been advanced in the past as to the fundamental reactions in the

[1] (*a*) "Corrosion: Causes and Prevention," by Frank N. Speller, McGraw-Hill Book Company, Inc., New York, 1936. Contains an extensive bibliography.

(*b*) "The Mechanism of Corrosion," by J. Johnston, *Ind. Eng. Chem.*, 1923, pp. 904–905.

(*c*) "The Electrochemical Theory of Corrosion," by Oliver P. Watts, *Trans. Electrochem. Soc.*, Vol. 64, pp. 125–150.

(*d*) "Metallic Corrosion Passivity and Protection," by Ulick R. Evans, Edward Arnold & Co., London.

phenomenon of corrosion but in recent years there has been general agreement upon the so-called "electrochemical theory." Inasmuch as iron is the most common metal, the following discussion refers to iron, but the theory holds, in all essential respects, with other metals.

When iron is in contact with an aqueous solution, some iron goes into solution in the form of iron ions. Because the system must remain in electrical equilibrium, an equal number of hydrogen ions leave the solution and are deposited or "plated" upon the surface of the iron as a thin film. The action is quite analogous to the plating out of copper on the surface of iron in a solution of copper sulphate. Hydrogen behaves in this reaction exactly like a metal.

If the hydrogen film is not removed, the reaction is obstructed and, in fact, may stop completely. There are, however, two ways in which the hydrogen film may be removed. It may react with oxygen that is present in the solution, or it may accumulate in gaseous form and escape as bubbles. In acid solutions the predominant way in which the hydrogen is removed is by this gaseous evolution, while in neutral or alkaline solutions the reaction with oxygen predominates.

The iron ions that enter the solution react with whatever oxygen may be present and iron oxide is precipitated, clearing the way for more iron ions to enter the solution. If the amount of dissolved oxygen in the solution is small, the corrosion reactions will be correspondingly restricted because there will be less oxygen to react with the hydrogen film and less to react with the iron ions in solution. This explains the effectiveness of deaeration in controlling internal corrosion in such places as steam power plants.

Electrode Potentials.—The tendency of a metal to go into solution depends upon its electrode potential. If two dissimilar metals are placed in a solution and connected externally so as to form a galvanic cell, the metal having the higher potential (*i.e.*, the more "noble" metal) becomes the cathode and the metal having the lower potential becomes the anode. It is the anodic metal that goes into solution and thereby becomes corroded.

The electrode potentials of the common metals are shown in Table I. Hydrogen is arbitrarily assigned zero potential and the values are simply relative. The relationship is not rigidly fixed but depends also upon the ion concentration in the solution, a point that should not be overlooked. Under certain conditions

the polarity of metals that are near each other in the electrode potential series may be reversed.

TABLE I.—STANDARD ELECTRODE POTENTIALS OF THE ELEMENTS AT 25 C (77 F)[1]

Element	Reference ion[2]	Potential in volts[3]
Lithium	Li^+	-2.959_5
Rubidium	Rb^+	-2.925_9
Potassium	K^+	-2.924_1
Calcium	Ca^{++}	-2.7_{63}
Sodium	Na^+	-2.714_6
Zinc	Zn^{++}	-0.761_8
Chromium	Cr^{++}	-0.557
Chromium	Cr^{+++}	-0.50_5
Iron	Fe^{++}	-0.44_1
Cadmium	Cd^{++}	-0.401_3
Nickel	Ni^{++}	-0.23_1
Tin	Sn^{++}	-0.13_6
Lead	Pb^{++}	-0.12_2
Iron	Fe^{+++}	-0.045
Hydrogen	H^+	-0.000_0
Copper	Cu^{++}	$+0.344$
Copper	Cu^+	$+0.522$
Iodine	I^-	$+0.534_5$
Silver	Ag^{++}	$+0.797_8$
Mercury	Hg^{++}	$+0.798_6$
Bromine	Br^-	$+1.064_8$
Chlorine	Cl^-	$+1.358_3$
Gold	Au^{++}	$+1.3_6$

[1] From reference 1(*a*), p. 1552. Values taken principally from "International Critical Tables."
[2] Plus and minus signs indicate valence.
[3] The potential given is that between the element in its standard state and its ion at unit activity in the solution. Digits shown as subnumbers merely indicate that their accuracy has not been fully established.

Hydrogen-ion Concentration.—The chemical activity of an acid depends upon its degree of ionization. A "weak" acid such as acetic acid does not ionize to the degree of a "strong" acid such as sulphuric acid. The hydrogen-ion concentration is an important index of the corrosive tendency of a solution.

pH Value.—In a neutral solution such as pure water the concentrations of the hydrogen ions and the hydroxide ions are both 10^{-7} equivalent per liter. Acid solutions have a higher concentration of hydrogen ions and alkaline solutions have a lower concentration [see footnote 1(*a*), page 1552]. The hydrogen-ion concentration is expressed by the symbol pH which is defined as

$$\mathrm{pH} = \log \frac{1}{\text{hydrogen-ion concentration}}.$$

A pH value of 7 indicates a neutral solution, while acid solutions

have a pH value less than 7 and alkaline solutions greater than 7. Table II shows the concentration of hydrogen and hydroxide ions for various pH values. It should be noted that a difference of pH of 1 point, as from 5.0 to 4.0, means a tenfold increase in hydrogen-ion content.

TABLE II.—RELATION BETWEEN pH AND ION CONCENTRATION

$$\left(\text{Computed from equation pH} = \log \frac{1}{\text{hydrogen-ion concentration}}\right)$$

pH	Ion concentration, gram equivalent per liter	
0	1.0	
1	0.1	Acid
2	0.01	solution
3	0.001	
4	0.0001	Hydrogen
5	0.00001	ions
6	0.000001	
7	0.0000001	Neutral solution
8	0.000001	
9	0.00001	Alkaline
10	0.0001	solution
11	0.001	
12	0.01	Hydroxide
13	0.1	ions
14	1.0	

Factors that Influence Corrosion.—Of the various factors that influence the rate of corrosion some are inherent in the metal itself while others are the result of the environment or of the combination of a particular metal with a particular environment. Of the factors inherent in the metal perhaps the most prominent are *those which cause differences of potential* between areas of the exposed surface. Lack of homogeneity is important in that respect, and variations in chemical composition in a metal may definitely increase the corrosion rate, although such effects may be of short duration, lasting only until the more anodic areas are removed by corrosion.

The combination of dissimilar metals is akin in its effect to lack of homogeneity. The more anodic metal goes into solution and, if the solution contains a high concentration of hydrogen ions (*i.e.*, low pH value), corrosion may be rapid. The electrical conductivity of the solution is also a controlling factor in this

type of corrosion. The conductivity of an electrolyte depends upon its concentration and upon its nature.

Protective *plating*, if the plating is perforated or otherwise discontinuous, may have a similar effect, particularly if the base metal is anodic with respect to the plating.

Stray electric currents may be important in the corrosion of underground pipes. In a direct-current railway system and in the distribution of direct-current electricity one side of the system is grounded. Some current, therefore, may leave the intended path, such as the rails of a street railway system, and find its way through the earth back to the power station. In seeking the path of least resistance current may enter a buried pipe line at one point and leave it somewhere else. The point or area of the pipe where the current leaves is anodic to the earth and is therefore subject to corrosion, which may be severe. Remedies consist of providing more direct paths back to the power plant, and other expedients.[1] A thorough survey is usually essential before intelligent measures can be chosen.

The inherent ability of a metal to form a *protective film* is a most important factor in corrosion. Aluminum, chromium, and silicon, and many of their alloys such as the chromium "stainless" steels, form a thin adherent film which effectively stops corrosion under many conditions. These films are believed to be oxides and are very thin, rapidly formed, and self-repairing. Iron forms oxide and hydroxide films which are not strongly protective because they are unstable.

Protective scales precipitated from solution are sometimes important. Pipes and boilers coated with scales of calcium sulphate and calcium carbonate are fairly well protected against corrosion.

Dissolved oxygen is an important factor, particularly at high temperatures. It removes the hydrogen which is plated out on the metal surface and thus permits corrosion to proceed.

The *rate of flow* of the liquid over the metal surface is sometimes important. The effect of high rates of flow or of turbulent flow is to renew rapidly the liquid in contact with the metal and thus prevent a decrease in the rate of the corrosion reactions.

Temperature is an important factor. Nearly all chemical reactions are accelerated with increasing temperature, and the corrosion

[1] See "Report of the American Committee on Electrolysis, 1921" (publisher not stated).

reactions are not exceptions. This phenomenon has been strikingly noticeable in steam power plants where the higher temperatures concomitant with higher boiler pressures have greatly increased corrosion difficulties.

Stress, particularly alternating stress, causes a type of failure known as "corrosion fatigue." It is really a notch fatigue in which the notch is formed initially by corrosion and then progresses through successive sheddings of the film of corrosion products which would otherwise be protective.

Bacteria are sometimes associated with corrosion. There are several forms of bacteria, the crenothrix being typical, which are said to take dissolved iron from water and precipitate it as ferrous hydroxide which oxidizes to ferric hydroxide. The resulting coating and tubercles of red rust cause clogging of water pipes which is sometimes serious. Corrosion may be caused also through the action of sulphate-reducing bacteria. The bacteria are not believed themselves to attack iron, but they may have some influence upon the corrosion rate.[1]

WHERE CORROSION TAKES PLACE

Corrosion of piping may be external or internal or both, depending on the environment and the nature of the fluid carried. The external sort may be either atmospheric corrosion if the pipe is above ground, or soil corrosion in case it is buried. Internal corrosion may be caused by the nature of the fluid itself, or by dissolved gases or other impurities that attack the pipe material. Some common cases of corrosion are discussed in succeeding paragraphs.

Soil Corrosion.—The external corrosion of underground pipes usually depends on the chemical characteristics of the surrounding soil but, in some cases, not so frequently as often supposed, it is due to or accelerated by stray electric currents (see page 1256).

In a locality where other pipes have been laid for long periods there usually will be a background of corrosion experience which can be revealed by inquiry. In other localities, such as those encountered in laying long cross-country pipe lines, a careful study

[1] For an extensive bibliography see, "Corrosion of Water Pipes in a Steel Mill," by C. L. Clark and W. J. Nungester, *Trans. Am. Soc. Metals*, Vol. XXXI, No. 2, pp. 304–320, June, 1943.

of the various soils is highly important. Much work[1] has been done in the classification and study of soils and their corrosive characteristics. For several years the National Bureau of Standards has been conducting corrosion tests in many parts of the country and the reports of that work contain much important information.

The Bureau of Standards tests had as one of their original objectives the determination of the relative susceptibility to corrosion of various ferrous metals such as cast iron, wrought iron, and steel. One important conclusion drawn from those tests was that the effect of the different soils was much more pronounced than the differences in the corrodibility of the various ferrous metals.

The present status of knowledge about soil corrosion is ably summarized in a Bureau of Standards publication (see footnote (*a*), below) as follows:

> There are a large number of sources of potentials which cause the corrosion of buried metal. The seriousness of corrosion underground depends largely on the character of the films or thicker deposits resulting from corrosion processes. Poor material is not an important cause of underground corrosion. The chemical composition of the soluble material in soils is an important factor in corrosion but when the soil contains only small percentages of soluble salts other factors control the rate of corrosion. Well drained soils are usually non-corrosive. Wet soils, organic soils, and soils high in soluble salts usually are corrosive. Stray currents cause corrosion only when they flow from metal to an electrolyte, usually the earth. Only direct currents cause corrosion under normal conditions.
>
> Much underground corrosion is attributable not to the character of the metal used or the soil but to conditions incidental to pipe-line construction, such as the

[1] (*a*) "Corrosion in Soils," *Letter Circ.* 689, Apr. 22, 1942, National Bureau of Standards (contains bibliography on soil corrosion).

(*b*) "Soil Corrosion Studies, 1934," by Kirk H. Logan, *J. Research Nat. Bur. Standards*, *RP* 883.

(*c*) "Soil Corrosion Studies, 1939," by Kirk H. Logan, *J. Research Nat. Bur. Standards*, *RP* 1460.

(*d*) "Pipe Corrosion and Coatings," by Erick Larson, published by American Gas Journal, Inc., 53 Park Place, New York, N.Y.

(*e*) "Developments in Soil Corrosion and Pipe Protection," by F. N. Speller and V. V. Kendall, *Jour. AWWA*, Vol. 30, pp. 1635–1650. Lists 37 references.

(*f*) "Copper Pipe As It Relates to the Corrosion Problem in City (Gas) Plants," by E. A. Munyan, *AGA Paper*, 1935. Reprints may be obtained from the American Gas Association, 420 Lexington Ave., New York 17, N.Y. See also footnote, p. 1263.

(*g*) "Soil Corrosion and Pipe Line Protection," by Dr. Scott Ewing, published by the American Gas Association, 420 Lexington Ave., New York 17, N.Y.

(*h*) "Soil Corrosion Studies 1941; Ferrous and Non Ferrous Corrosion Resistant Materials and Non Bituminous Coatings," by Kirk H. Logan and Melvin Romanoff, *J. Research Nat. Bur. Standards*, *RP* 1602.

interconnection of old and new pipe, the crossing of different soils or soil horizons, voids in the backfill, and current picked up from other structures.

There are several tests which will indicate whether soils are potentially corrosive. When a sufficient number of tests are made they yield satisfactory indications as to the locations and extent of corrosive areas. The correlation of the results of tests of single samples of soil with the corrosion observable at the points where the samples were taken may not be good because other factors than soil characteristics may control the corrosion, and because the soil samples may not be representative of conditions at the point of corrosion.

There is no standard or generally accepted criterion for corrosivity or corrosion resistance. The relative merits of materials with respect to corrosion may change with the time of exposure, the area exposed and the conditions of exposure.

Usually most of the commonly used ferrous materials, including many low-alloy steels, corrode at nearly the same rates when exposed to the same soil conditions. Lead corrodes slowly in most soils because of the formation of protective layers of carbonate and sulfate. Copper and alloys high in copper corrode much more slowly than ferrous materials in most soils. Copper is much affected by soils containing hydrogen sulfide.

Bituminous coatings usually are imperfect or develop imperfections. Most of them after a few years permit some corrosion but the better coatings reduce losses of weight and pit depths for ten or more years. Zinc is the only metal extensively used for underground coatings. Its effectiveness is temporary.

Cathodic protection properly applied and maintained is an effective means of preventing corrosion. Under some conditions a combination of a protective coating and cathodic protection affords the most economical means of preventing corrosion.

Corrosion in Steam Heating Systems.—In steam heating systems in which all the condensate is returned to the boilers and therefore little make-up water is used, corrosion is negligible and the piping has a long life. Corrosion problems occur, for the most part, in heating systems that use some make-up water and in buildings that are supplied with steam from outside sources and do not return the condensate.

When corrosion is encountered in steam heating systems, it usually is found in the condensate return piping and in the radiators, unit heaters, or other apparatus that condenses steam. It seldom occurs in the steam supply piping. It attacks in some cases both nonferrous metals (copper and brass) and ferrous metals.

The two principal agents that cause corrosion are oxygen and carbon dioxide.[1] Oxygen is present in most boiler waters and carbon dioxide is produced by the breakdown of calcium bicarbonate, which often is present in water, and by the hydrolysis of sodium carbonate when that material is used as a scale preventive.

[1] "Corrosion in Steam Heating Systems," by Leo F. Collins with Everette L. Henderson, *Heating, Piping and Air Conditioning*, September, 1939, to May, 1940.

Oxygen by itself can be a source of corrosion of ferrous metals in heating systems. Its exact action is not fully understood and it is only under unusual conditions, such as high velocity of flow or above-normal temperature, that oxygen causes severe corrosion in such piping systems.

The severity of corrosive attack by condensate containing carbon dioxide and oxygen depends upon the concentration of those gases in the solution. That concentration depends, in turn, upon the partial pressure of the gases in the steam that is in contact with the condensate. The gases tend to accumulate in the radiators, water heaters, etc., and that accounts for the much greater severity of corrosion at those points as compared with the steam supply piping.

To ensure a negligible rate of corrosion in heating systems, particularly those portions supplied with high-pressure (50 to 150 psi) steam or medium-pressure (10 to 50 psi) steam, the carbon dioxide content of the steam should, in the author's judgment, be not more than 5 parts per million and the oxygen content not more than 0.03 parts per million. The restriction of carbon dioxide appears to be the more important.

At present the reduction of the gas content of the steam (see page 1271) appears to be the only effective way of reducing the corrosion of the ferrous metals in heating systems. Inhibitors, such as sodium silicate and oil, have been used in a few cases and have been tentatively reported[1] as somewhat effective, but further confirmation is needed and they are not in general use.

Corrosion in Steam-power-plant Piping.—In most cases corrosion in steam-power-plant piping is caused by carbon dioxide or oxygen, and the general conditions governing it are similar to those discussed in connection with steam-heating systems. Carbon dioxide is not a factor unless the plant uses a large proportion of make-up water. Most corrosion in power-plant piping is caused by oxygen and, because of the high temperatures and usually clean internal surfaces, the damage may be severe unless the oxygen content is reduced to practically zero. To do this the supply water is carefully deaerated, inleakage of air is eliminated, and often a reducing agent such as sodium sulphite is added to the water. This subject is further discussed on pages 1271 to 1272. As might be expected the effects of corrosion by oxygen are

[1] See various reports of the Corrosion Committee in the Proceedings of the National District Heating Association, 1930–1934.

observed mostly in the boilers and economizers where the temperatures are highest although the piping may be attacked to a lesser degree.

In a few cases ammonia has been the cause of corroding copper and copper alloys in piping, valves, heat exchanger tubes, etc. Unless continually removed from the plant circuit, ammonia, which is introduced with contaminated boiler-feed make-up, tends to accumulate and carry on a destructive cycle between the nonferrous and ferrous parts of the circuit. This may continue to the point of causing tube failures in the boiler or heat exchangers. The presence of ammonia may be detected in some cases, at least, by the bluish products of corrosion that are formed at the areas attacked.

Corrosion in Water-supply Systems.—When certain kinds of water flow through *ferrous* pipes, a reaction takes place which results in the formation of aggregations of iron compounds with other substances, the aggregations being known as "tubercles" and the process as "tuberculation." Tubercles form inside the pipe where the iron or steel has rusted, presumably where the pipe coating is defective. Although cast iron resists water corrosion better than wrought iron or steel, it is about equally susceptible to tuberculation. The tubercle builds up as a combination of iron oxide, silica, lime, and organic matter. It often is cone shaped and may increase in size up to a 2-in.-diameter base and 1 in. height.

Under adverse conditions the carrying capacity of water-supply mains may be reduced fully one-half by tuberculation so that their value as part of the waterworks plant is correspondingly decreased. According to tests reported to the New England Water Works Association,[1] the average actual loss in flow capacity of tar-coated cast-iron pipe after 30 years' service, based on a total of 473 tests in 19 different systems, was 52 per cent. The loss predicted in the Williams-Hazen Tables[2] for mains of similar age and diameter (average, 20.5 in.) and for "average soft unfiltered river water" is 32.3 per cent. The committee concluded the Williams-Hazen age coefficient relation to be applicable primarily to large-diameter mains carrying relatively inactive water. For small-diameter mains carrying active water, the actual loss after 30 years may be

[1] "Report of Committee on Pipe Line Friction Coefficients and Effect of Age Thereon," *J. New Eng. Water Works Assoc.*, Vol. XLIX, No. 3, September, 1935. Also available in reprint form.

[2] "Hydraulic Tables," by Gardner S. Williams and Allen Hazen, John Wiley & Sons, Inc., New York, 1920.

twice the Williams-Hazen predicted loss. Other factors being equal, the data show marked correlation between pH value of water carried and rate of capacity loss, average conditions reported indicating for supplies with a pH value of 6.5 twice the loss observed in supplies with a pH value of 8.

Since cast-iron pipe otherwise would remain useful for a great many years, perhaps even for centuries, it is usual practice to be generous in the selection of pipe sizes for underground water-supply systems so as to offset the adverse effect of tuberculation. As a direct means of combating the corrosion or tuberculation of ferrous pipe by soft or corrosive waters, various protective linings, such as cement and enamel, have found considerable favor; also the use of cement and cement-asbestos pipe.

Nonferrous materials are used to a considerable extent for water-supply piping in the smaller sizes obtaining in service connections and house plumbing. Although galvanizing iron or steel pipe retards corrosion for a time, this form of protective coating wastes away by electrolysis and the line eventually has to be replaced. In the case of domestic hot-water supply systems, the deterioration of galvanized pipe may be quite rapid so that the affected portions are apt to need replacement in the course of 5 to 10 years. Consequently nonferrous pipe of various sorts has been used extensively for conditions to which the respective kinds are suited. To avoid premature failure of galvanized-steel hot-water tanks, it is inadvisable to use copper and galvanized steel in the same water system where the water supply contains more than two parts per million of carbon dioxide. Copper has a tendency to dissolve in water of higher CO_2 content and to redeposit on the galvanized-steel tank, thus creating numerous electrochemical cells that stimulate corrosion. Direct contact between the dissimilar metals seems to play only a minor part in this action. The best way known at present to avoid the difficulty is to use all-copper or all-galvanized hot-water systems.[1]

Whereas lead pipe has been used for many underground water services in the past, it has been found susceptible to corrosion by soft water containing oxygen, carbon dioxide, or organic acids. Hence, rain water and soft well waters should not be conveyed by lead pipe. The compounds formed in the corrosion of lead are

[1] See "The Problem of Copper and Galvanized Iron in the Same Water System," by L. Kenworthy, formerly Investigator, British Non-Ferrous Metals Research Association, *J. Inst. Metals* (London), February, 1943.

apt to be soluble in water, which they render unsuitable for drinking purposes owing to the hazard of lead poisoning. Red brass pipe containing 80 to 90 per cent copper resists soft waters the best. Admiralty metal (70 to 75 per cent copper with about 1 per cent tin and the remainder zinc), red brass, and lead are recommended for use with sea water. Lead-lined steel pipe is extensively used for sea water aboveground, although unlined steel pipe is sometimes used.[1]

Because of its good corrosion-resisting properties for the conditions usually encountered, copper water tube, copper pipe, or red brass pipe are recommended for water service lines.[2] If rigid pipe is used, it should be laid to provide proper compensation for expansion and for the shifting of the earth in which it is placed. In underground work, pipe and tubing are subject to corrosive attack from both within and without, and red brass and copper are, in general, satisfactory for all soil conditions except cinder fill or salt marshy soils. For these corrosive soils the pipe or tubing should be protected by painting with asphalt paint and wrapping with heavy burlap or wrapping paper. It also is desirable to backfill around the pipe with clay or clean sand mixed with lime. Other materials that are suitable for this purpose are limestone or broken plaster, if they happen to be available.

Corrosion of Industrial Process Piping.—A complex situation is encountered with industrial process piping where corrosive chemicals and organic products have to be conveyed through pipes. In the piping and apparatus of chemical plants, corrosion conditions often are so severe that no material will stand up for any great length of time and it is expected that replacements will be frequent. What is regarded as good corrosion resistance in a chemical operation may be inadequate length of life in a steam power plant. Where conditions and available materials permit, however, the piping not only should be sufficiently resistant from a durability standpoint, but reactions between the piping and the fluids conveyed should be minimized to avoid unnecessary pollution or wastage of the product.

Owing to the variety of conditions experienced with industrial piping and chemical process work, it is difficult to generalize on this

[1] See "Recommendations for Using Steel Pipe in Salt Water Systems," by Paul Ffield, *Jour. Am. Soc. Naval Engrs.*, Vol. 57, No. 1, pp. 1–20, February, 1945.

[2] See "The Use of Copper and Brass Tubes," by Carter S. Cole, Engineer, Copper and Brass Research Association, New York, *Jour. AWWA*, Vol. 32, pp. 2071–2076, 1940.

subject. All sorts of materials and pipe linings are used including ferrous and nonferrous metals, stainless steels, glass, plastics, wood, etc., depending on circumstances. Specific suggestions on the probable suitability of various materials for conveying different chemicals and organic products will be found on pages 1273 to 1285.

CONTROLLING CORROSION

The natural tendency is for metals to corrode and revert to their oxides or other forms in which they originally existed in the earth. Since it is not often possible or economical to eliminate completely the process of corrosion in the case of structures such as pipe lines, many attempts to control corrosion are aimed at extending their useful life to a reasonable economic span. Several methods are available for controlling corrosion, depending on the nature of individual cases and the economics of the situation. As discussed in succeeding pages, the principal means employed, either separately or in combination, are:

1. Protective coatings applied to relatively inexpensive base materials, usually iron or steel.
2. Deaeration, deactivation, and/or chemical treatment of the fluid being conveyed.
3. Cathodic protection used as a shield against electrochemical action associated with corrosion.
4. Adopting corrosion-resistant materials which may, or may not, be more attractive economically than plain ferrous materials, depending on first cost, useful life, and general fitness for the purpose.

Protective Coatings.—Where pipe material is subjected to corrosive conditions to which it is susceptible, some form of protective coating may be more economical than resorting to a more resistant but more expensive material. Since corrosion may arise from the fluid conveyed by the pipe, from the atmosphere, cr from the soil environment in the case of a buried pipe, protection may be needed either internally or externally, or both, depending on circumstances. In the case of internal attack it may be possible to reduce the corrosive nature of the fluid conveyed through deaeration or chemical treatment as discussed elsewhere in this chapter.

Protective Coatings for Steel Pipe.—To meet the problem of *soil* corrosion, many types of external coatings have been tried on steel pipe particularly on gas, oil, and water transmission lines of

comparatively large size, and in field tests made under the joint sponsorship of the National Bureau of Standards, the American Petroleum Institute, the American Gas Association and others.[1] As explained later, an *internal* protective coating may be called for with some fluids, or the pipe may be dipped or otherwise treated so as to coat both the inside and the outside as frequently is done with underground water pipe.

External coatings for buried pipes may be classified in five general groups: (1) natural and petroleum by-product asphalt, (2) asphalt mastic, (3) coal-tar pitch, (4) wax and grease, and (5) miscellaneous metallic and synthetic products. Although there is now little choice in the physical properties of blended and treated asphalts and coal tars, the latter are generally considered more resistant to moisture absorption. Wax and grease have found favor in the gas-service field on the score that they can be applied at or near atmospheric temperature. Bituminous products are applied at temperatures approximating 400 F. Exterior zinc, lead, and tin coatings[1] have not proved satisfactory underground. Hard rubber and porcelain enamel show physical if not economic promise. Desirable characteristics of protective coating ordinarily considered in meeting corrosion problems are described in the next paragraph.

To be effective, an external pipe-protection coating must have four fundamental characteristics: (1) a tenacious bond to the pipe, (2) imperviousness to moisture and corrosive salts, (3) high electrical resistance, and (4) strength to withstand the load and abrasion of stones and clay soil.[2] As these conditions cannot be met so far by a single-coat product, manufacturers have employed a composite coating.

Condition 1, that of obtaining a continuous strong bond, requires that steel pipe have a clean, roughened surface. Methods used

[1] (*a*) "Soil Corrosion and Pipe Line Protection," by Dr. Scott Ewing, published by the American Gas Association, 420 Lexington Ave., New York 17, N.Y.

(*b*) "Soil-Corrosion Studies, 1939," by Kirk H. Logan, *Res. Paper* 1446, *Bur. Standards J. Research*, Vol. 28, No. 1, January, 1942.

[2] (*a*) "Specifications for a System of Bituminous Pipe Line Protection," by Ulric B. Bray and Frederick S. Scott, *Gas Age-Record*, July 14, 1934, p. 33.

(*b*) "Recent Developments in the Application of Asphalt-mastic Coating to Both New and Operating Pipe Lines," by W. W. Colley. Presented before the Natural Gas Section at the 1940 annual meeting. Copies obtainable from the American Gas Association, 420 Lexington Ave., New York 17, N.Y. See also, "Somastic Pipe Coating for Permanent Protection of Underground Pipe Lines," by W. W. Colley, *Mines Mag.*, Vol. 30, No. 8, August, 1940.

to obtain this end include the following: (*a*) shot blasting with steel grit, (*b*) sand blasting, (*c*) power-driven rotary wire brushing, and (*d*) wire brushing by hand. Depending upon what is to be used as a first protective coat, an intermediate pipe primer may or may not be necessary. Zinc chromate, red lead, and asphalt and coal-tar base paints are used as primers. The embodiment of corrosion inhibitors in primers does not afford adequate protection to pipe in corrosive soils.[1]

The first coat of materials should present an impregnable barrier against corrosive solutions (Condition 2). It should have also extreme adhesiveness to steel pipe to provide a firm foundation upon which to build subsequent coatings needed for mechanical protection of the moisture barrier. Coal-tar and wax products have found general favor for this purpose on account of their low moisture absorption properties although by-product asphalt products are in general use.

Electrical resistance (Condition 3) is required on two scores: (1) Localized corrosion manifested as pitting of steel pipe is viewed as part of an electric circuit in which the interaction of soluble salts and steel constitutes the generating cell. A high-resistance coating minimizes the available paths for circulation of current. (2) Corrosion is greatly stimulated by stray electric currents traveling in the earth from street-car systems, etc. Since steel pipe offers a low-resistance conductor, these stray currents pass along the pipe, in some cases jumping to and from the pipe according to soil-conductance conditions. Wherever current passes from the pipe, severe corrosion may take place. Electrical resistance may be a property of the first coat or it may be incorporated as a wrapper in the second coat.

The second and subsequent coats and wraps are used, where required, to give mechanical strength and resist distortion occasioned by stones and clay soil. This coat usually consists of (1) a strong wrapper of asphalt or coal-tar saturated paper, asbestos, or linen tape, with an outer coat of coal tar[2] (see also page 1267), (2) a heavy coat of asphaltic mastic, or (3) a dielectric wrapper of

[1] "API Pipe-coating Tests—Final Report," by Kirk H. Logan, *Proc. API, Production Bull.* 226, Vol. 21, Sec. IV, 1940.

[2] "Standard Specifications for Coal-tar Enamel Protective Coatings for Steel Water Pipe; Sizes 30 inches and Over, No. 7A.5; Sizes 4½ Inches Up to But Not Including 30 Inches, No. 7A.6." Copies may be obtained from the American Water Works Association, 22 East 40th St., New York 16, N.Y.

reinforced cellulose acetate with an outer coat of wax. Where soil conditions are particularly severe, the second coat sometimes is repeated. Over all is an optional wrapping of heavy kraft paper to facilitate handling, or a coat of whitewash for protection against sun heat. In general, thick coatings provide greater protection.

Protective coatings may be applied to pipe: (1) at a treatment plant specializing in this work, (2) at a rail-head plant, (3) by a portable machine, or (4) by a manually operated sling. Although in the API test Dr. Logan reported[1] that "it is impossible to determine positively whether the machine applied coatings were superior," Dr. Ewing concluded[2] that machine coating was to be preferred.

Although no standard specification for coating steel pipe exists in the gas industry, two specifications for coatings in water service have been formulated by the American Water Works Association.[3] These AWWA specifications cover the material and application requirements for coal-tar enamel protective coatings for the *interior* and *exterior* of steel water pipes as follows:

1. For the *inside* of all pipe, a coat of coal-tar primer followed by a hot coat of coal-tar enamel applied either by manual or mechanical means.

2(*a*). For the *outside* of all pipe 30 in. and over in diameter to be placed *underground*, a coat of coal-tar primer followed by a coat of coal-tar enamel and one coat of water-resistant whitewash.

2(*b*). For the *outside* of all pipe 4½ in. O.D. up to but not including 30 in. O.D. to be placed *underground*, a coat of coal-tar primer followed by a hot coat of coal-tar enamel into which shall be bonded an asbestos felt wrapper, and finished with a kraft paper or one coat of water-resistant whitewash.

3. For the *outside* of all pipe to be placed *aboveground exposed to the weather*, two coats of synthetic red-lead primer (or one coat of synthetic red-lead primer and one coat of synthetic white enamel) and one coat of aluminum paint.

Physical tests for coal-tar enamel are provided in the foregoing AWWA specifications. In addition, electrical and visual inspection is required for every length of pipe, with checks for adhesion of coating to pipe. The electrical flaw detector, or holiday tester,

[1] See footnote 1, p. 1266.

[2] "Field Tests of Pipe Coating," Dr. Scott Ewing, *Proc. AGA*, 1935, p. 627.

[3] See footnote 2, p. 1266.

should be of approved design and capable of delivering 8,000 to 10,000 volts at low amperage.

Corrosion protection for pipe joints and fittings is applied in the field by hand. In general, the same material is used as was applied to pipe in the shop. In some cases this procedure necessitates the use of heating equipment and hot applications with an attendant hazard which is minimized where a molding form is used. Greases and compounds for cold application also are available. Great care should be exercised in coating joints as any moisture that can enter the coating at this point may travel extensively along the pipe itself and result in corrosion occurring underneath an otherwise satisfactory form of pipe protection.

For resisting *internal* corrosion, particularly in water-supply piping, sewage lines, and similar services where mild corrosion is expected, several kinds of protective coatings are in use. The following account of present commercial practice is quoted from a paper by Deming Bronson.[1]

Perhaps the simplest and best known form of interior protection for steel water pipe is galvanizing. Thousands of miles of galvanized steel pipe have been installed annually for a great many years, both for water service lines and for the smaller distribution mains. Briefly, galvanizing is the application of zinc in molten form to steel pipe to accomplish a deposition of the zinc on the interior walls of the pipe. In this process, the pipe is pickled in acid, washed and neutralized in water, and dipped in a fluxing bath. It is then heated and rolled into a molten bath of zinc which is maintained at over 800 F. In this the pipe is allowed to remain until it acquires the temperature of the molten zinc. The weight of the galvanizing usually averages two ounces per square foot of surface coated. One prominent manufacturer of galvanized steel pipe supplements this process with a special chromate treatment to resist discoloration and formation of what is called white rust. It is claimed that such additional treatment preserves, for a considerable time, the smooth glistening surface or metallic lustre of the hot galvanizing.

Two steel pipe manufacturers offer cement lining.[2] The type of cement used is usually of low lime and high silica content, having low solubility properties. The mix is applied by a centrifugal method and is followed by a curing treatment that increases the strength of the cement and reduces the inherently low shrinkage factor. This type of lining has substantial acceptance for special installations, particularly for hot water lines in building construction.

[1] "Protective Coatings for Steel Water Lines," by Deming Bronson, *Jour. AWWA*, Vol. 32, No. 8, pp. 1385–1393, August, 1940.

[2] See AWWA Standard Specifications 7A.7 for Cement-mortar Protective Coating for Steel Water Pipe 30 Inches and Over. Copies may be obtained from the American Water Works Association, 22 East 40th St., New York 16, N.Y.

A third form of interior protection for steel water pipe is the application of two coats of a cold-applied liquid coal-tar solution. Only a two-coat job is recommended or offered by leading applicators because the double coat removes the possible existence of pin holes or voids in a single coat. This liquid coal-tar coating, which is maintained in liquid form by use of coal-tar solvents, is usually applied by a combination of spraying and brushing. After the first coat is dry, the second coat is applied similarly, and is then allowed to dry before shipment of the pipe is made from the pipe mill. The resultant thickness of these two coats of cold-applied coating is about $\frac{1}{64}$ inch. The surface is black and glossy and, with the interior surface of the pipe properly prepared by removing rust and mill scale, the surface is smooth, hence increasing the flow of water through the pipe during the life of the lining.

There is still commercially available, in the industry, pipe that has been dipped in a hot asphalt bath. This dipping is accomplished with the pipe either in horizontal or vertical position. The pipe is made clean of loose scale, rust, dirt, oil, and grease—the better practice includes the heating of the pipe until it is perfectly dry—then dipped into a tank of hot asphalt and allowed to remain until the pipe and the asphalt reach practically the same temperature. The pipe is then withdrawn, and, if horizontal dipping is used, is held at a sufficient angle to allow any surplus coating to drain off naturally without wiping or swabbing. Asphalt-dip lining is from 0.01 to 0.03 inch thick.

The most generally accepted and most highly regarded form of interior protection for steel water pipe is *hot spun coal-tar enamel*. In this process the interior surface of the pipe is made clean and free of oil, grease, rust, and loose mill scale and is then primed with a cold-applied liquid coal-tar primer. This primer is applied by spraying or brushing or by a combination of both. The primer is a bonding medium between the metal and the hot-applied enamel. After it is properly dry, the hot coal-tar enamel is applied from a trough, from a weir, or from a retractable feed line while the pipe is revolving on its own axis at a relatively high peripheral speed. The force of revolution of the pipe distributes the enamel evenly over the inner pipe wall in the desired thickness, as controlled, within close limits, by various means depending upon the mechanical devices used in application. The revolving of the pipe is usually continued (with or without the introduction of cooling water on the inner surfaces) until the enamel has cooled to a non-sagging temperature or has set to firmness. The resultant surface of the hot-applied enamel lining is black, smooth and glossy and provides the utmost in increased flow capacity. It is usually installed $\frac{3}{32}$ plus or minus $\frac{1}{32}$ inch thick.

As stated by Bronson, zinc is the most common protective coating, well known as galvanizing. Zinc is protective because it is anodic with respect to iron and is intended to be consumed by corrosion. Its protective effect, therefore, is in inverse ratio to its rate of consumption and too long life should not be expected. Other metallic coatings such as cadmium, nickel, and tin are of value under special conditions. Vitreous enamel linings are successful at moderate temperatures and in an alkaline or neutral environment.

Lead-lined pipe is suitable for many conditions such as for sea water and for weak acids (see pages 1273 to 1285). It is available commercially as are lead-lined valves and fittings. A lead lining can be inserted in iron or steel pipe quite easily by simply running in a lead pipe just small enough for insertion and then expanding it by air pressure. Flanged joints are used since lead-lined pipe cannot be welded conveniently. The lead lining is turned out over the face of the iron flange.

Protective Coatings for Cast-iron Pipe.—Except where otherwise specified, all cast-iron pipe for water service is completely coated inside and out by preheating and dipping in hot coal-tar pitch varnish, to which sufficient oil has been added to make a smooth coating, tough and tenacious when cold, not "tacky," and not brittle. Gas pipe can be furnished uncoated, or coated only on the outside with standard coating materials. American Standard specifications for Coal-tar-dip Coating for Cast Iron Pipe and Fittings are in course of preparation by ASA Sectional Committee A21 under the sponsorship of the AGA, ASTM, AWWA, and NEWWA. When required to meet service conditions, cast-iron pipe can be supplied with special coatings such as bituminous enamel applied externally and/or internally, or cement mortar lining.[1] Pipe and fittings should be thoroughly cleaned, have the rough spots removed, and be given a careful hammer test before coating.

Portland cement lining has been found effective for preventing tuberculation inside water-supply piping handling certain kinds of waters that are prone to cause aggregations of iron compounds. The type of cement used should be of low-lime and high-silica content, having low-solubility properties. A strong mix is applied by a centrifugal method which gives a dense uniform lining of $\frac{1}{8}$ to $\frac{1}{4}$ in. thickness, depending on pipe size. The lining is then given a curing treatment which produces a surface of enamellike smoothness and reduces the inherently low shrinkage factor. Cement lining thus applied remains intact even when the pipe is cut with a cold chisel or tapped for services in the usual way, and is not injured by railroad transportation. A cement lining is not wholly impervious to water, but the water that penetrates forms an alkaline solution in contact with the iron and tends to inhibit

[1] See American Standard Specifications for Cement Mortar Lining for Cast Iron Pipe and Fittings, ASA A21.4. Copies may be obtained from the American Standards Association, 29 West 39th St., New York 18, N.Y. See also Federal Specifications WW-P-421.

corrosion. Where the fluid has a low pH value, however, there is a continual reaction in progress between the coating and the fluid. Hence cement-lined pipe is suitable for only mildly acid conditions. When so specified, a bituminous seal coat may be applied to the cement mortar lining which is said to make it more impervious to corrosion and to reduce friction losses.

Deaeration and Deactivation.—Because corrosion in water pipes of power and industrial plants is so frequently attributed to dissolved oxygen, much attention has been given to methods of removing or neutralizing oxygen. There are several types of deaerators available which remove oxygen mechanically, by utilizing the principles that govern the solubility of gases in liquids.

The solubility of a gas in a liquid decreases with increasing temperature and with decreasing partial pressure of the gas in the atmosphere above the liquid. Where the water is to be heated, as in the case of boiler feed water, the deaerating process is combined with the heating process. The water is heated and, in one design, is sprayed into a space containing steam. Because of the low partial pressure of oxygen in the steam, the oxygen is set free from the water. In order to effect deaeration in the very short time allowable, the water must be broken up into a fine spray so as to expose as much surface as possible. Other designs use cascade trays instead of sprays.

If the water is not to be heated, deaeration can be accomplished by spraying the water into a chamber in which the pressure has been reduced well below atmosphere. Deaeration is not often necessary in cold-water systems but occasionally some severe corrosion has been encountered.[1]

Chemical deactivation is accomplished by introducing a substance that will unite with the oxygen and thus prevent it from attaching the pipe metal. One method is the Speller deactivator which is simply a tank containing iron in some such form as expanded steel sheets. This method has been successfully used in such applications as the hot-water supply systems of large apartment buildings.[2] A filter is desirable to secure perfectly clear water. The expanded metal sheets must of course be renewed periodically.

[1] "Vacuum Deaerator Combats Corrosion," by Sheppard T. Powell and Homer S. Burns, *Chem. Met. Eng.*, Vol. 43 No. 4, pp. 180–184, April, 1936.

[2] "The Preservation of Hot-water Supply Pipe," by F. N. Speller and R. G. Knowland, *Trans. ASHVE*, Vol. 24, pp. 217–234, 1918.

Reducing agents may be added to the water to react with the oxygen, provided the total amount of oxygen to be neutralized is not so great as to make the cost prohibitive. This method is suitable in steam power plants, for example, where the oxygen content of the boiler feed water has been reduced to a low value by mechanical deaeration but where as nearly complete neutralization as possible is desired. The chemicals commonly used are ferrous hydroxide, and potassium or sodium sulphite.

Cathodic Protection.—Corrosion of underground pipes occurs where the metal is anodic with respect to the adjacent soil as would be expected in view of the electrochemical nature of corrosion (see page 1252). Consequently, if the pipe is made cathodic by an applied electrical potential, corrosion will be markedly reduced in many cases. This is done by connecting the pipe to the negative pole of a source of direct current, the positive pole being grounded. The method is discussed in AGA and National Bureau of Standards papers.[1]

Cathodic protection can be applied to either coated or unprotected underground pipe lines and, what often is important, it can be applied to lines already installed. This may be of value in prolonging the life of lines where what originally was an adequate coating has deteriorated with time. Cathodic protection has been applied to the internal protection of water storage tanks aboveground, and it seems reasonable to expect that it might be beneficial in some cases as an internal pipe protection.

Choice of Corrosion-resisting Materials.—Although not ideally suited to all conditions, iron and steel have been regarded for many years as the conventional piping materials. Owing to internal and external corrosion problems, other products always have been used to a limited extent. Recently, nonmetallic materials have entered the field. Among the latter are vitrified clay, cement and cement asbestos, glass, and various synthetics, each having its own appropriate field from a standpoint not only of corrosion resistance but of mechanical properties. The applications of

[1] (*a*) "The Development and Application of a Practical Method of Electrical Protection for Pipe Lines against Soil Corrosion," by Starr Thayer, AGA 1933. Reprints may be obtained from the American Gas Association, 420 Lexington Ave., New York 17, N.Y.

(*b*) "The Status of Cathodic Protection in 1941," by Kirk H. Logan. Mimeographed copies are obtainable from the National Bureau of Standards, Washington, D.C.

these materials are discussed at various places throughout this handbook (see index).

The suitability of available materials for resisting attack by chemicals and organic products is of prime importance in specifying piping for industrial process work.[1] Problems of heat resistance often are of equal importance and may be allied to those of corrosion resistance. Alloys have been developed which have both qualities in varied degrees.[2]

Condensed recommendations are presented herewith as to the probable resistance to corrosion in different applications of some of the commonly used piping materials. No one of them will successfully resist corrosion by all the fluids encountered in the chemical process industries, however, and any evaluation is necessarily relative with respect to any given set of service conditions. As pointed out under Corrosion of Industrial Process Piping (see page 1263), corrosion conditions in the chemical industry often are so severe that no material will stand up for any great length of time.

CONDENSED RECOMMENDATIONS FOR SELECTING CORROSION-RESISTANT MATERIALS FOR PIPING[3]

Absorption Oil.—Choice of materials depends upon the character of the product absorbed and the nature of the process in which it is being used.

Acetate Solvents.—Iron and steel used, preferably galvanized. Trim iron valves with brass or bronze. All brass or acid resisting bronze may be required in some cases.

Acetic Acid.—Iron and steel sometimes used with hot, crude vapors and with waste liquors. Ni-Resist much better. Trim iron valves with 18-8 Cr-Ni steel, acid-resisting bronze, or Monel metal. Copper and acid-resisting bronze standard materials for handling this acid, both when pure and in the form of crude pyroligneous acid. Aluminum, 18-8 Cr-Ni steel, Monel metal, and Inconel used where contamination with copper, lead, zinc, etc., must be avoided. Silver, 18-8 Cr-Ni-Mo steel, Hastelloy C, and other special alloys required when exposure is to hot vapors or to acetylating mixtures. Monel metal and Inconel indicated for mixtures of the acid and salt.

Acetic Anhydride.—Iron and steel, copper, and acid-resisting bronzes used with crude product. Aluminum used with pure anhydride.

[1] "Materials Used in Chemical Industries," by B. E. Roetheli and H. O. Forrest, *Ind. Eng. Chem.*, September, 1932, pp. 1018–1022.

[2] "Tables of Chemical Compositions, Physical and Mechanical Properties, and Corrosion-resistant Properties of Corrosion-resistant and Heat-resistant Alloys," *Proc. ASTM*, Vol. 30, Part I, 1930.

[3] Reproduced by permission from Crane Company, "Combatting Corrosion in Industrial Process Piping," *Tech. Paper* 408. For composition and physical properties of alloys see Tables II and III on pp. 345 to 347.

Acetone.—Iron and steel used; also brass and copper. Trim iron valves with brass. Avoid rubber.

Acetylene.—Iron and steel used on crude gas. Brass commonly used with purified gas. Avoid higher copper alloys if gas is not pure and perfectly dry.

Acid Mine Waters.—See Mine Waters.

Air.—Iron, steel, and brass used. If these corrode look for presence of moisture and/or other gases.

Alcohols.—Iron and steel used, preferably galvanized; also brass and copper. Trim iron valves with brass.

Aldehydes.—Iron, steel, brass, and copper used. When these corrode look for impurities.

Aluminum Sulfates; Alums.—14% silicon cast iron, lead, rubber, and 18-8 Cr-Ni-Mo steel good. Copper, acid-resisting bronzes, and Monel metal also used.

Ammonia.—Cast iron and steel regularly used in standard ammonia valves. When these corrode look for impurities in the ammonia.

Ammonium Chloride.—Iron and steel used. Ni-Resist better than regular cast iron. Trim iron valves with Monel metal. Brass used, but copper base alloys not recommended if solutions are ammoniacal. Monel metal required in special cases.

Ammonium Hydroxide; Ammonia Liquors.—Iron and steel regularly used with crude liquors. Trim iron valves with 12% Cr steel or 18-8 Cr-Ni steel. Aluminum or 18-8 Cr-Ni steel good with pure reagent.

Ammonium Nitrate.—Iron and steel used. Trim valves with 18-8 Cr-Ni steel. Use all 18-8 Cr-Ni steel or aluminum in special cases.

Ammonium Phosphate, Mono- and Di-basic.—Iron and steel used. Trim iron valves with acid-resisting bronze or 18-8 Cr-Ni steel. Brass and acid-resisting bronze better than iron. 18-8 Cri-Ni steel required in special cases.

Ammonium Phosphate, Tri-basic.—Iron and steel good. Trim valves with 12% Cr steel, 18-8 Cr-Ni steel, or Monel metal.

Ammonium Sulfate.—Iron and steel used; also brass. Ni-Resist better than regular cast iron. Trim iron valves with brass, acid-resisting bronzes, or Monel metal. Acid-resisting bronzes or Monel metal required if much excess sulfuric acid is present.

Amyl Acetate.—See Acetate Solvents.

Amyl Alcohol.—See Alcohols.

Aniline; Aniline Oil.—Iron and steel sometimes permissible. Trim iron valves with brass; acid-resisting bronze, 18-8 Cr-Ni steel, or Monel metal.

Aniline Dyes.—Iron and steel used for drain lines, but corrode appreciably. Ni-Resist better than regular cast iron. Monel metal and 18-8 Cr-Ni steel used when better material is required.

Asphalt.—Iron and steel regularly used.

Barium Chloride.—Iron, steel, or brass used. Trim iron valves with brass or Monel metal. Use Monel metal or nickel when greater freedom from corrosion is required.

Barium Hydroxide.—Iron and steel used. Nickel, Monel metal, or 18-8 Cr-Ni steel may be required in special cases. If seats in all-iron valves corrode, trim with 12% Cr steel, 18-8 Cr-Ni steel, or Monel metal.

Barium Sulfate.—Iron and steel used in absence of free acid. Trim valves with brass or, if erosion is a problem, with 18-8 Cr-Ni-Mo steel or Monel metal.

Barium Sulfide.—Iron and steel used. Trim valves with 12% Cr steel, 18-8 Cr-Ni steel, or Monel metal.

Beer.—Copper, brass, aluminum, and 18-8 Cr-Ni steel used in the beverage industry. Brass, iron, and steel permissible in the alcohol industry. Trim iron valves with brass, 18-8 Cr-Ni steel, or Monel metal.

Beet Sugar Liquors.—Iron and steel used; also copper and brass. Ni-Resist better than regular cast iron. Trim iron valves with brass, 18-8 Cr-Ni steel, or Monel metal.

Benzene; Benzol; Coal Tar Naphtha.—See Coal Tar Solvents.

Benzine; Petroleum Naphtha.—See Petroleum Oils and Solvents.

Benzoic Acid.—Aluminum and nickel used with the anhydrous vapors. Monel metal and 18-8 Cr-Ni steel resist the aqueous solutions.

Bichloride of Mercury.—See Mercuric Chloride.

Black Liquor.—See Sulfate Liquors.

Blast Furnace Gas.—See Gases, Fuel.

Bleaching Solutions.—See Calcium Hypochlorite, Sodium Hypochlorite, Sodium Perborate, Hydrogen Peroxide, etc.

Blue Vitriol.—See Copper Sulfate.

Borax.—Iron and steel used; also brass. If seats in iron valves corrode, trim with 12% Cr steel, 18-8 Cr-Ni steel, or Monel metal.

Bordeaux Mixture.—Iron and steel used, preferably galvanized; also brass. 18-8 Cr-Ni steel indicated for valve trim when seating surfaces fail because of abrasion and other contributing factors.

Boric Acid.—14% silicon cast iron, lead, copper, and acid-resisting bronzes used. Ni-Resist good. Monel metal and 18-8 Cr-Ni-Mo steel used when contamination must be avoided.

Brines.—Iron, steel, and brass used. Ni-Resist indicated when regular cast iron fails by "graphitic corrosion." Red brass indicated when yellow brass fails by "dezincification." Trim iron valves with brass, acid-resisting bronzes, or Monel metal. When equipment is exposed to brine on one side and to food products or other fluid on other side, choice of material must be based on satisfactory resistance to *both*.

Bromine.—Usually handled in glass and/or chemical stoneware. Hastelloy C indicated when metallic equipment is required

Butane.—See Gases, Hydrocarbon.

Butanol; Butyl Alcohol.—See Alcohols.

Butyl Allosolve; Ethylene Glycol Mono Butyl Ether.—See Ethers.

Calcium Bisulfite.—Lead, copper, and bronze used with cold solutions. Corrosion-resisting chromium-nickel steels used for hot liquors and vapors.

Calcium Hydroxide.—Iron and steel used. Nickel, Monel metal, or 18-8 Cr-Ni steel may be required in special cases. If seats in all-iron valves corrode, trim with 12% Cr steel, 18-8 Cr-Ni steel, or Monel metal.

Calcium Hypochlorite.—Iron and steel used but corrode appreciably. Ni-Resist better. Trim iron valves with 18-8 Cr-Ni-Mo steel. Results with all metallic materials may be erratic. 14% silicon iron sometimes good; molybdenum bearing modification better. Do not confuse with Sodium Hypochlorite, which is more corrosive.

Calcium Sulfate.—Iron and steel used in absence of free acid. Trim valves with brass or, if erosion is a problem, with 18-8 Cr-Ni-Mo steel or Monel metal.

Caliche Liquors.—Iron and steel used. Trim valves with 18-8 Cr-Ni steel.

Cane Sugar Liquors.—Iron and steel used; also copper and brass. Ni-Resist better than regular cast iron. Trim iron valves with brass, 18-8 Cr-Ni steel, or Monel metal.

Carbitol; Diethylene Glycol Monoethyl Ether.—See Ethers.

Carbolic Acid; Phenol.—See Phenols.

Carbon Dioxide.—Iron, steel, and brass used with dry gas. Trim valves with 12% Cr steel or 18-8 Cr-Ni steel. For wet gas, see recommendations for Carbonic Acid.

Carbon Disulfide.—Iron and steel used; Ni-Resist required in some cases. Trim iron valves with 18-8 Cr-Ni steel.

Carbonic Acid.—Iron and steel used; but corrode. Ni-Resist better than regular cast iron. Red brass and acid-resisting bronzes good. Yellow brass not good. Trim iron valves with 18-8 Cr-Ni steel. Aluminum and 18-8 Cr-Ni steel required for piping in special cases.

Carbon Tetrachloride.—See Chlorinated Solvents.

Carburetted Water Gas.—See Gases, Fuel.

Casing Head Gasoline.—See Petroleum Oils and Solvents.

Castor Oil.—See Vegetable Oils.

Caustic Potash and Caustic Soda.—See Potassium Hydroxide and Sodium Hydroxide.

Cellosolve; Ethylene Glycol Monoethyl Ether.—See Ethers.

Cellulose Acetate; Cellulose Nitrate.—Choice of material depends upon the solvent used and the stage of process at which the chemicals are being handled. See also acetylating mixtures under Acetic Acid or Nitrating Acids.

Chile Saltpeter.—See Sodium Nitrate.

China Wood Oil.—See Drying Oils or Vegetable Oils.

Chloride of Lime.—See Calcium Hypochlorite.

Chlorinated Solvents.—Steel, copper, and brass regularly used; with no corrosion in the absence of moisture. In the presence of water and/or steam, corrosion may become serious; in which case the preferred materials are Monel metal, nickel, silicon bronze, or aluminum bronze.

Chlorine.—Iron, steel, and bronze used with dry gas. Trim valves with Monel metal. Wet vapors corrode most metallic materials. Hastelloy C and silver used.

Chlorex; Dichloroethyl Ether.—See Chlorinated Solvents.

Chloroacetic Acid.—Lead, nickel, and copper used. Silver and Hastelloy C probably best materials.

Chrome Alum; Chromium Potassium Sulfate.—See Aluminum Sulfate; Alums.

Chromic Acid.—Lead preferred for tank linings, coils, etc. Cast iron fairly good and used for valves on electroplating tanks. 18-8 Cr-Ni steel excellent and recommended for trim in iron valves.

Citric Acid.—Copper and acid-resisting bronze used. Aluminum, 18-8 Cr-Ni steel, Monel metal, and Inconel used where contamination with copper must be avoided.

Coal Tar Solvents.—Not corrosive when dry and free from chemicals used in refining operations. Iron, steel, and brass used. When these corrode look for water and/or other impurities. If ammonia or other alkalis are present avoid brass and use iron or steel. Valves may be trimmed with 12% Cr steel, 18-8 Cr-Ni steel, Monel metal, or other high nickel alloys. If acids are present,

trim valves with acid-resisting bronze, Monel metal, or 18-8 Cr-Ni-Mo steel. Ni-Resist better than regular cast iron in the latter case.

Coke Oven Gas.—See Gases, Fuel.

Copper Cyanide.—Solutions usually contain sodium cyanide, and behave as do simple solutions of that chemical. See Sodium Cyanide.

"Copper Solutions."—These solutions contain excess ammonium hydroxide (which see); but may be encountered in circumstances where slight corrosion is highly objectionable. Iron and steel used. 18-8 Cr-Ni steel indicated where perfect resistance is required. *Avoid brass.*

Copper Sulfate.—18-8 Cr-Ni steel indicated. Lead useful in presence of free sulfuric acid. Copper and acid-resisting bronze used.

Core Oils.—Iron, steel, and brass used.

Corn Oil.—See Edible Oils or Vegetable Oils.

Cottonseed Oil.—See Edible Oils or Vegetable Oils.

Creosote; Cresols; Cresylic Acid.—Iron and steel regularly used for crude products. When valve seats corrode, trim with 18-8 Cr-Ni steel, or Monel metal. For refined products, see recommendations for Phenols.

Dichloro-di Fluoro-methane; Freon.—See Gases, Refrigerant.

Dichloroethyl Ether.—See Chlorinated Solvents.

Diethylene Glycol.—See Alcohols.

Diethylene Glycol Monoethyl Ether.—See Ethers.

Distilled Water.—Use aluminum or 18-8 Cr-Ni steel. Sanitary type tubing and fittings preferred.

Distillery Wort.—Copper and brass regularly used. Tin coatings sometimes desired.

Doctor Solutions; Sodium Plumbite.—Iron and steel regularly used. Ni-Resist better than regular cast iron. If valve seats corrode, trim with 18-8 Cr-Ni steel or Monel metal.

Drying Oils.—Iron, steel, and brass used where discoloration is not a factor. Ni-Resist better than regular cast iron. If valve seats corrode, trim with bronze, 18-8 Cr-Ni steel, or Monel metal. Aluminum, 18-8 Cr-Ni steel, or Monel metal used when contamination and discoloration must be avoided.

Edible Oils.—Iron, steel, copper, brass, and bronze used. Aluminum, 18-8 Cr-Ni steel, and Monel metal employed when contamination with copper, lead, etc., must be avoided.

Epsom Salts.—See Magnesium Sulfate.

Ethane.—See Gases, Hydrocarbon.

Ethers.—Not ordinarily corrosive to iron and steel. Copper and bronze regularly used.

Ethyl Acetate.—See Acetate Solvents.

Ethyl Alcohol.—See Alcohols.

Ethyl Chloride.—See Gases, Refrigerant.

Ethyl Sulfate.—Iron and steel regularly used. If valve seats corrode, trim with brass, acid-resisting bronze, or Monel metal.

Ethylene Glycol.—See Alcohols.

Ethylene Glycol Monobutyl Ether.—See Ethers.

Ethylene Glycol Monoethyl Ether; Cellosolve.—See Ethers.

Fatty Acids.—Organic acids typified by acetic acid and oleic acid. See specific acid for detailed recommendations.

Ferric Chloride.—Very corrosive. Rubber, stoneware, or Hastelloy C indicated.

Ferric Sulfate.—Acid-resisting bronzes, Monel metal, and lead have fair to good resistance. Fully resisted by 18-8 Cr-Ni steel.

Ferrous Chloride.—Behaves like dilute hydrochloric acid, which see.

Ferrous Sulfate.—Iron or brass used. When these corrode, treat like dilute sulfuric acid, which see.

Foamite Solutions.—Mixtures of sodium bicarbonate and aluminum sulfate or dilute sulfuric acid. See recommendations for those chemicals.

Formaldehyde.—Iron, steel, and brass used. When valve seats corrode, trim with acid-resisting bronze or Monel metal.

Formic Acid.—Corrodes aluminum; otherwise recommendations are as for acetic acid.

Fruit Juices.—Copper and acid-resisting bronzes used. Aluminum, 18-8 Cr-Ni steel, Monel metal, nickel, or Inconel indicated when contamination with copper must be avoided.

Fuming Sulfuric Acid; Oleum.—See second paragraph under Sulfuric Acid.

Furfural.—Iron and steel regularly used when this chemical is used in refining lubricating oils.

Fusel Oils.—See Alcohols.

Gallic Acid.—Monel metal, nickel, and 18-8 Cr-Ni steel preferred materials. Copper, acid-resisting bronzes, and brass used.

Gases, Fuel.—Iron, steel, and brass regularly used. When valve seats corrode, trim with bronze, 18-8 Cr-Ni steel, or Monel metal.

Gases, Hydrocarbon.—Iron, steel, and brass regularly used. If corrosion occurs, look for impurities. Trim steel valves in high-pressure lines with high nickel alloys.

Gases, Inert.—No corrosion at ordinary temperatures. At high temperatures special materials may be required.

Gases, Refrigerant.—Iron, steel, brass, and copper used. No corrosion unless gases are contaminated with water and/or decomposition products.

Gasoline.—See Petroleum Oils and Solvents.

Gelatine.—Aluminum, nickel, and 18-8 Cr-Ni steels used.

Glauber's Salt.—See Sodium Sulfate.

Glucose.—Iron, steel, copper, and brass used. Trim iron valves with brass, 18-8 Cr-Ni steel, or Monel metal.

Glue.—Iron, steel, copper, and brass used. Trim iron valves with brass, 18-8 Cr-Ni steel, or Monel metal.

Glycerine; Glycerol.—See Alcohols.

Grain Alcohol.—See Alcohols.

Green Liquor.—See Sulfate Liquors.

Helium.—See Gases, Inert.

Hydrochloric Acid; Muriatic Acid.—Corrodes most metallic materials. Rubber, glass, and stoneware used extensively. Monel metal, nickel, silicon bronze, and aluminum bronze used. Hastelloys good. Temperature, acid concentration, and presence of oxidizing agents are more important with this and related acids than with most other corrosive fluids.

Hydrocyanic Acid.—Iron, steel, and brass used with dry gas. When valves and piping corrode, look for impurities.

Hydrofluoric Acid.—Corrodes glass, stoneware, and 14% silicon cast iron. Otherwise recommendations are as for hydrochloric acid, with the exception that lead is successfully used with the crude acid containing sulfuric acid. C grade is best of the Hastelloys.

Hydrofluosilicic Acid.—Lead used extensively. Copper and acid-resisting bronzes useful.

Hydrogen.—See Gases, Inert.

Hydrogen Peroxide.—Use aluminum or 18-8 Cr-Ni steel. Sanitary tubing and fittings preferred.

Hydrogen Sulfide.—Iron and steel used. 5% chromium steel indicated if temperature is above normal. Trim valves with 12% Cr steel or 18-8 Cr-Ni steel. Aluminum and 18-8 Cr-Ni steel used for piping in special cases.

"Hypo."—See Sodium Thiosulfate.

Iodine.—Usually handled in glass or acid-resisting stoneware. Hastelloy C indicated if metallic equipment is required.

Kerosene.—See Petroleum Oils and Solvents.

Lacquers.—Ordinarily not corrosive, in which case iron and steel (preferably galvanized), copper, and brass can be used. When these corrode, use aluminum, 18-8 Cr-Ni steel, or Monel metal.

Lime Sulfur.—Iron and steel used, preferably galvanized. Ni-Resist better than regular cast iron. When valve seats corrode, trim valves with 18-8 Cr-Ni steel.

Linseed Oil.—See Drying Oils or Vegetable Oils.

Lubricating Oils.—See Petroleum Oils and Solvents.

Lye.—See Sodium Hydroxide.

Magnesium Chloride.—See Brines for general recommendations. Use Monel metal and nickel when complete freedom from corrosion is desired. Acid-resisting bronzes and Monel metal good, and more apt to be required than in the case of calcium chloride or sodium chloride.

Magnesium Sulfate.—Iron, steel, and brass used. Acid-resisting bronzes or Monel metal required in presence of free sulfuric acid. Trim iron valves with brass, acid-resisting bronze, or Monel metal.

Mercuric Chloride.—Iron, steel, and Monel metal used. Hastelloy C or special chromium-nickel alloys indicated in some cases. *Avoid brass and copper.*

Mercury.—Iron and steel used. Steel valves can be trimmed with any of the corrosion-resisting steels ordinarily used for that purpose. *Avoid brass and copper.*

Methane.—See Gases, Hydrocarbon.

Methanol; Methyl Alcohol.—See Alcohols.

Methyl Chloride.—See Gases, Refrigerant.

Milk.—Choice of material depends upon stage at which product is being handled. Glass, 18-8 Cr-Ni steel, nickel, Inconel, and Monel metal best materials. Sanitary tubing and fittings required.

Milk of Lime.—See Calcium Hydroxide.

Mine Waters.—Usually contain free acid and dissolved salts of iron and/or copper which play an important part. Ni-Resist much better than ordinary iron and steel. Corrosion-resisting non-ferrous alloys required in some cases and corrosion-resisting steels in other cases. Latter indicated when terric and copper salts are present. Former better when these salts are absent. Furnish complete information regarding dissolved constituents.

Mixed Acids.—Too indefinite to warrant recommendations. See Nitrating Acids for mixtures of nitric and sulfuric acids. See Pickling Acids for other mixtures.

Molasses.—Iron and steel and brass used where contamination is not objectionable. Copper, acid-resisting bronzes, chromium-nickel corrosion resisting steels, and Monel metal used where something more resistant is required.

Monochloroacetic Acid.—See Chloroacetic Acid.

Monochlorobenzene.—See Chlorinated Solvents.

Muriatic Acid.—See Hydrochloric Acid.

Naphtha, Coal Tar; Benzene; Benzol.—See Coal Tar Solvents.

Naphtha, Petroleum; Benzine.—See Petroleum Oils and Solvents.

Natural Gas.—See Gases, Fuel.

Neon.—See Gases, Inert.

Nickel Chloride.—Frequently encountered admixed with nickel sulfate in electroplating solutions. See Nickel Sulfate. For handling the pure salt, acid-resisting bronzes and Monel metal are used.

Nickel Sulfate.—Most frequently encountered in electroplating solutions. 14% silicon cast iron used for pumps and valves. Lead used for coils and tank linings. Hard rubber used. 18-8 Cr-Ni-Mo steels used.

Nitrating Acids.—Recommendations depend upon the relative amounts of each acid and of water present. Cast iron, steel, 14% silicon cast irons, and corrosion-resisting Cr-Ni steels used. Furnish complete information regarding acid concentration.

Nitre Cake.—See Sodium Bisulfate.

Nitric Acid.—High chromium alloys (over 18%) resist all concentrations of the pure acid at all temperatures. Some grades of 18-8 Cr-Ni steel are satisfactory. Aluminum good with very dilute or with full strength acids. Neither high chromium alloys nor aluminum good if acid contains hydrochloric acid as an impurity.

Nitrobenzene; Oil of Mirbane.—Iron and steel used. If valve seats corrode, trim valves with 18-8 Cr-Ni steel.

Nitrogen.—See Gases, Inert.

Oil of Mirbane.—See Nitrobenzene.

Oleic Acid.—Iron, steel, brass, copper, and acid-resisting bronzes used. Ni-Resist better than regular cast iron. Trim iron valves with brass, bronze, or 18-8 Cr-Ni steel. Trim Ni-Resist valves with 18-8 Cr-Ni steel or Monel metal. Aluminum, Monel metal, and 18-8 Cr-Ni steel good with pure acid. Latter used for distillation equipment. 18-8 Cr-Ni-Mo steel required for hot vapors.

Oil of Vitriol; 66° Sulfuric Acid.—See second paragraph under Sulfuric Acid.

Oleum Acid; Fuming Sulfuric Acid.—See second paragraph under Sulfuric Acid.

Oleum Spirits.—See Petroleum Oils and Solvents.

Oxalic Acid.—14% silicon cast iron, copper, acid-resisting bronzes, and Monel metal good.

Oxygen.—Steel and brass used. No corrosion when dry except at elevated temperatures.

Palmitic Acid.—Iron, steel, brass, copper, and acid-resisting bronzes used. Ni-Resist better than regular cast iron. Trim iron valves with brass, bronze, or 18-8 Cr-Ni steel. Trim Ni-Resist valves with 18-8 Cr-Ni steel or Monel metal. Aluminum, Monel metal, and 18-8 Cr-Ni steel good with pure acid. Latter used for distillation equipment. 18-8 Cr-Ni-Mo steel required for hot vapors.

Pentane.—See Gases, Hydrocarbon.

Petroleum Oils and Solvents.—Iron, steel, and brass used with refined products, which should be non-corrosive. Occasional corrosion may result from

incomplete purification; but is more apt to be due to the accidental presence of water or—in the case of blended products—to the presence of compounds not of petroleum origin.

Corrosion by crude oils may be due to hydrogen sulfide and/or entrained salt water—choice of valve trim depends upon which is present. Iron and steel ordinarily satisfactory for the bulk of this equipment.

Corrosion by incompletely refined products may be due to hydrogen sulfide or organic sulfur compounds, sulfurous acid, hydrochloric acid, or treatment chemicals. Carbon steel used at moderate temperatures in absence of free acids or corrosive waters. Brass and acid-resisting bronzes good at moderate temperatures in absence of much hydrogen sulfide. 5% Cr-Mo used at high temperatures. 12% Cr steel used for valve trim in same range. 18-8 Cr-Ni steels good in both high and low temperature range provided temperatures are not such as to produce embrittlement. 18-8 Cr-Ni-Mo steel better than regular grade on low temperature side if hydrochloric acid is present.

Phenol.—Iron, steel, and brass used where some corrosion is permissible. Aluminum, 18-8 Cr-Ni steel, nickel, and silver used where contamination must be avoided.

Phosphoric Acid.—Use 18-8 Cr-Ni steel for diluted pure acid. 18-8 Cr-Ni-Mo steel may be preferred in some cases. Special chromium nickel alloys required for concentrations over 45% at elevated temperatures. Lead used with pure dilute acid and with crude acid containing sulfuric acid. Acid-resisting bronzes sometimes permissible.

Phthalic Acid.—Usually handled in alcoholic solutions. Use aluminum or 18-8 Cr-Ni steel.

Pickling Acids.—These include sulfuric, nitric, hydrochloric, and hydrofluoric, either individually or in the form of mixtures of two or more. Furnish complete information regarding acids present.

Picric Acid.—Use aluminum or steel for molten acid. Use lead or 18-8 Cr-Ni steel for aqueous solutions.

Potassium Carbonate.—See recommendations for Sodium Carbonate.

Potassium Chloride.—Not ordinarily used for commercial brines and may necessitate the use of Monel metal or nickel. Otherwise see recommendations for Sodium Chloride.

Potassium Cyanide.—See Alkaline Cyanide. See recommendations for Sodium Cyanide.

Potassium Hydroxide.—See Fixed Alkalis. See recommendations for Sodium Hydroxide.

Potassium Nitrate.—See recommendations for Sodium Nitrate.

Potassium Sulfate.—See recommendations for Sodium Sulfate.

Potassium Sulfide.—See recommendations for Sodium Sulfide.

Printing Inks.—Composition indefinite and requirements vary. Acid-resisting bronzes or Monel metal good in almost all cases. Iron, steel, and brass frequently good enough.

Producer Gas.—See Gases, Fuel.

Propane.—See Gases, Hydrocarbon.

Propionic Acid.—See recommendations for Acetic Acid.

Propyl Alcohol.—See Alcohols.

Prussic Acid.—See Hydrocyanic Acid.

Pyridine.—Iron and steel used. When these corrode look for other corroding

Pyrogallic Acid; Pyrogallol.—See recommendations for Phenol.

Pyroligneous Acid; Pyroligneous Liquor.—See recommendations under Acetic Acid.

"Red Oil."—See Oleic Acid.

Return Condensate.—Iron, steel, and brass ordinarily satisfactory. Red brass indicated when these fail, unless corrosion can be prevented by elimination of dissolved carbon dioxide and/or dissolved oxygen. Low alloyed irons and steels sometimes superior to carbon steel.

Rosin.—Iron and steel used with crude product. Aluminum, 18-8 Cr-Ni steel, and Monel metal used when contamination and discoloration must be avoided.

Salammoniac.—See Ammonium Chloride.

Salicylic Acid.—Copper, lead, and acid-resisting bronzes used. Aluminum, nickel, and Monel metal indicated when contamination must be avoided.

Salt.—See Sodium Chloride.

Salt Cake.—See Sodium Acid Sulfate.

Salt Water; Sea Water.—Iron and steel rust profusely but have satisfactory life in many cases. Galvanizing increases life and is often worth while. Ni-Resist better than regular cast iron, especially where latter fails by graphitic corrosion. Red brass better than yellow brass, especially when latter fails by dezincification. Cupro-nickel alloys indicated when both aeration and turbulence are involved. Trim iron valves—when used—with red brass, bronze, Monel metal, or other copper-nickel alloys.

Sewage.—Cast iron, vitrified clay, brick, and concrete are commonly used underground. Cast iron used for sluice gates, with brass or bronze trim. Ni-Resist often good in locations where regular cast iron corrodes badly.

Shellac.—Iron and steel (preferably galvanized) used when solutions are not corrosive. Red brass and acid-resisting bronze may be required in some cases. Aluminum, 18-8 Cr-Ni steel, or Monel metal indicated when solutions are corrosive and contamination must be avoided.

Sludge Acids.—Composition indefinite; corrode most regular materials. Ni-Resist better than ordinary cast iron. Copper-nickel alloys good in many cases. More special materials may be required during concentration and other phases of recovery process.

Soap.—Iron and steel used with strong solutions encountered in manufacture. Trim iron valves with 18-8 Cr-Ni steel or Monel metal. Latter preferred when salt is present and alkalinity is low. Brass used with dilute solutions encountered in laundries and for trim in iron valves in the same service. 18-8 Cr-Ni steel good.

Soda Ash.—See Sodium Carbonate.

Sodium Acid Sulfate; Sodium Bisulfate.—Quite corrosive to iron and steel, but uses are such that these materials are often employed. Red brass and acid-resisting bronzes better. Ni-Resist better than regular cast iron.

Sodium Bicarbonate.—Iron and steel used where contamination is not a factor; *e.g.*, when it is used in Foamite Systems. When valve seats corrode, trim with bronze, 12% Cr steel, 18-8 Cr-Ni steel, or Monel metal. Latter two indicated when contamination must be avoided.

Sodium Carbonate.—Iron and steel regularly used. When valve seats corrode, trim with 12% Cr Steel, 18-8 Cr-Ni steel, or Monel metal.

Sodium Chloride.—Iron, steel, brass, and bronzes regularly used. Ni-Resist better than regular cast iron. Red brass better than yellow brass. Trim

iron valves with brass, bronze, or Monel metal. Use Monel metal or nickel when contamination must be avoided.

Sodium Cyanide.—Iron and steel used. Vapors above liquid level may be more corrosive than solutions. When valve seats corrode, trim with 12% Cr steel, 18-8 Cr-Ni steel, or Monel metal. Avoid brass.

Sodium Hydroxide.—Iron and steel used when complete resistance to corrosion is not required. Low nickel cast irons and Ni-Resist better than unalloyed cast iron. When valve seats corrode, trim with 12% Cr steel, 18-8 Cr-Ni steel, or Monel metal. Use Monel metal or nickel when contamination must be avoided or when service conditions are such that steel may fail by "caustic embrittlement."

Sodium Hypochlorite.—Results with metallic materials very erratic. Bronze and copper used, but corrode. Special Cr-Ni steels good in some cases. Hastelloy C good. Concrete, tile, vitreous enamel, and rubber indicated when and where their use is feasible.

Sodium Hyposulfite.—See Sodium Thiosulfate.

Sodium Metaphosphate.—Iron and steel used but may corrode. Red brass, acid-resisting bronzes, and Monel metal better. 18-8 Cr-Ni steel indicated in some cases and recommended for trim in iron valves.

Sodium Nitrate.—Iron and steel used. Trim iron valves with 12% Cr steel or 18-8 Cr-Ni steel.

Sodium Perborate and Sodium Peroxide.—Behavior similar to sodium hydroxide but may be more corrosive because of decomposition and liberation of oxygen. Iron and steel used. Trim iron valves with 12% Cr steel, 18-8 Cr-Ni steel, or Monel metal. Use 18-8 Cr-Ni steel or Monel metal when contamination must be avoided.

Sodium Phosphate, Mono- and Di-basic.—Iron and steel used, but may corrode. Trim iron valves with 18-8 Cr-Ni steel. Use 18-8 Cr-Ni steel when complete resistance is required.

Sodium Phosphate, Tribasic.—Iron and steel good. Trim valves with 12% Cr steel, 18-8 Cr-Ni steel or Monel metal. Aluminum should not be used.

Sodium Plumbite.—See Doctor Solutions.

Sodium Silicate.—Iron and steel used. Avoid alternate wetting and drying if possible. Keep gate valves full of liquid silicate to prevent sticking.

Sodium Sulfate.—Iron, steel, and brass used. If these corrode use acid-resisting bronzes or Monel metal.

Sodium Sulfide.—Iron and steel used. Vapors above liquid level may be more corrosive than solutions. When valve seats corrode, trim with 12% Cr steel or 18-8 Cr-Ni steel. Avoid brass.

Sodium Thiosulfate; Sodium Hyposulfite; "Hypo."—Iron and steel used when contamination is not a factor. Lead, 18-8 Cr-Ni steel, or Monel metal used when contamination must be avoided.

Soja (Soy) Bean Oil.—See Drying Oils or Vegetable Oils.

Stearic Acid.—Iron, steel, brass, copper, and acid-resisting bronzes used. Ni-Resist better than regular cast iron. Trim iron valves with brass, bronze, or 18-8 Cr-Ni steel. Trim Ni-Resist valves with 18-8 Cr-Ni steel or Monel metal. Aluminum, Monel metal, and 18-8 Cr-Ni steel good with pure acid. Latter used for distillation equipment. 18-8 Cr-Ni-Mo steel required for hot vapors.

Sulfate Liquors.—General name applied to "Black Liquors," "Green Liquors," and "White Liquors" employed in manufacture of paper pulp. Iron and steel

standard materials. Low nickel cast irons better than nickel-free iron. Ni-Resist still better. Trim iron or steel valves with 12% Cr steel, 18-8 Cr-Ni steel, or Monel metal.

Sulfur.—Iron and steel used.

Sulfur Chloride.—Iron and steel used without much corrosion in absence of moisture. Ni-Resist better than regular cast iron. Monel metal good. Avoid brass.

Sulfur Dioxide.—Hot gas encountered in manufacture of sulfuric acid and sulfite liquors can be handled in iron and steel. Trim valves with acid-resisting bronze or 18-8 Cr-Ni steel. Pure gas used for refrigerating purposes handled in copper and brass. See Gases, Refrigerant.

Sulfur Trioxide.—Hot gas encountered in manufacture of sulfuric acid can be handled in iron and steel. Latter preferred because of tendency for cast iron to absorb gas and become brittle. If valve seats corrode because of presence of moisture, trim valves with bronze or Monel metal.

Sulfuric Acid.—Lead preferred material for lower concentrations whenever feasible. 14% silicon cast iron good. Acid-resisting bronzes and Monel metal and other copper-nickel alloys with low concentrations. Ni-Resist sometimes useful. 18-8 Cr-Ni-Mo steel good with very dilute solutions (under 10%) at ordinary temperatures. Iron not good for dilute acids.

Iron and steel used with concentrations above 60° Bé. 18-8 Cr-Ni steel used for trim in valves handling full strength acid (98% and fuming), especially when hot. Brass and bronze not good with strong acids. Lead not good if both concentration and temperature are high.

Sulfurous Acid.—Lead very good if acid is cold. Bronzes also good with cold acid. 18-8 Cr-Ni steel good with hot pure acid and hot vapors. Do not confuse pure acid with sulfite liquors encountered in pulp and paper industry. These are apt to be more corrosive and usually call for 18-8 Cr-Ni-Mo steel or other corrosion-resisting Cr-Ni steels.

Sweet Water.—Iron, steel, or brass used. Iron valves may be trimmed with brass, acid-resisting bronzes, Monel metal, or 18-8 Cr-Ni steel.

Tannery Liquors.—Iron and steel used but may corrode. Trim iron valves with bronze or Monel metal. Red brass and bronze also used. 18-8 Cr-Ni steel or Monel metal may be required in special cases.

Tannic Acid.—Monel metal, nickel, and 18-8 Cr-Ni steel preferred materials. Copper, acid-resisting bronzes, and brass used.

Tar.—Iron and steel used. When valve seats corrode, trim with bronze or 18-8 Cr-Ni steel.

Tartaric Acid.—Copper and acid-resisting bronze used. Aluminum, 18-8 Cr-Ni steel, Monel metal, and Inconel used where contamination with copper must be avoided.

Titanium Chloride.—Iron and steel good in absence of moisture. Acts like hydrochloric acid when wet, and in that case calls for silicon bronze, aluminum bronze, Monel metal, or Hastelloy.

Toluene; Toluol.—See Coal Tar Solvents.

Trichloroacetic Acid.—See recommendations for Chloroacetic Acid.

Trichloroethylene.—See Chlorinated Solvents.

Triethanolamine.—Iron and steel used. If valve seats corrode, trim with 18-8 Cr-Ni steel or Monel metal.

Tri-sodium Phosphate (T.S.P.).—See Sodium Phosphate, Tribasic.

Turpentine.—Iron, steel, and brass used. If valve seats corrode, trim with bronze, 18-8 Cr-Ni steel, or Monel metal. Aluminum, 18-8 Cr-Ni steel, or Monel metal used when discoloration must be avoided.

Varnish.—Iron, steel, copper, and brass used. When discoloration must be avoided, use aluminum, 18-8 Cr-Ni steel, or Monel metal.

Vegetable Oils.—Iron, steel, copper, and brass used with crude oils and with refined oils in the absence of much free acid. When these corrode, use aluminum, 18-8 Cr-Ni steel, or Monel metal.

Vinegar.—Copper and acid-resisting bronzes used. Aluminum and 18-8 Cr-Ni steel used when contamination with copper must be avoided and when salt is absent. Monel metal or Inconel indicated for vinegar-salt mixtures.

Water.—Iron and steel give good results when protective coatings form or when dissolved oxygen is absent. Galvanizing desirable. Brass better than iron, but yellow brass may not be good enough if iron corrodes badly. When yellow brass fails by dezincification, use red brass or bronze. When brass or bronze valve trim corrodes, replace with 12% Cr steel and/or copper-nickel alloys. See also Distilled Water, Acid Mine Waters, Return Condensate, and Salt Water.

Water Glass.—See Sodium Silicate.

Whiskey.—Copper and brass used. 18-8 Cr-Ni steel used to a limited extent.

White Liquor.—See Sulfate Liquors.

Wine.—Copper and brass used. 18-8 Cr-Ni steel, Monel metal, and nickel preferred in some cases.

Zinc Chloride.—When admixed with creosote in wood preservation, iron valves trimmed with Monel metal are indicated. For pure salt, materials resistant to hydrochloric acid are sometimes required.

Zinc Sulfate.—For acid solutions use lead, copper, bronze, or Monel metal. When solutions are neutral, iron and steel are often satisfactory. Ni-Resist better than regular cast iron.

Zylene; Yylol.—See Coal Tar Solvents.

CHAPTER XVIII

HYDRAULIC-POWER TRANSMISSION PIPING

The number of applications in industry and transportation where hydraulic-power transmission systems can be used to good advantage is steadily increasing. To mention a few: manufacturing industries have hydraulically operated machine tools and hydraulically operated equipment for pressing, forming, extruding, and other fabricating or processing operations; aircraft have extensive hydraulic control systems for retractable landing gears, wing flaps, or wherever considerable power or a high degree of controllability is required; motor vehicles have hydraulic braking systems and hydraulically operated devices for lifting dump bodies, road scrapers, bulldozers, and the like; modern ships have hydraulic steering gear in place of steam steering engines on large ships or hand operation on smaller craft. As used in such applications, hydraulic power provides an effective means for producing a moving or holding force of unlimited magnitude for operating various devices and for pressing, forming, extruding, drawing, or forging of steel, brass, copper, aluminum, magnesium, plastics, rubber, and other materials. In other applications, high-pressure water is discharged in high-velocity jets through specially designed nozzles for descaling hot steel during the process of rolling or forging, for removing bark from logs in the pulp and timber industry, and for removing coke from coking chambers in the oil industry. Water from hydraulic systems also is used in mills and shops for the hydrostatic testing of pipe and tubing, valves, pressure vessels, and other pressure-containing equipment.

ELEMENTS AND PRINCIPLES OF HYDRAULIC SYSTEMS

The fundamental principle on which hydraulic-power transmission systems operate is that of Pascal's law (see page 27) which is shown diagrammatically in Fig. 1. Through application of power to the small piston over a long travel, or in a large number

of strokes, fluid is delivered at the same unit pressure to perform one stroke of the large piston. Neglecting friction, the mechanical advantage obtained is equivalent to the ratio of the area of the large piston to the area of the small piston. Some of the applications made of this principle in hydraulic systems are described in succeeding paragraphs.

Basic Systems.—A simple fluid system consists primarily of three main parts; a fluid pump, a cylinder containing a piston driven by the fluid, and piping to convey the fluid power from the pump to the operating cylinder. One of the simpler forms of hydraulic systems is the automobile brake mechanism in which the

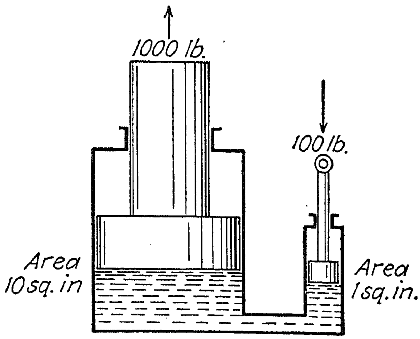


Fig. 1.—The fundamental principle known as Pascal's law on which all hydraulic power transmission systems operate.

power is applied by depressing a pedal, thus operating the master cylinder, or pump, which forces oil through the lines to the wheel cylinders to apply the brake shoes.

In the basic hydraulic system, shown schematically in Fig. 2, two additional units are present which are included in most hydraulic systems, *viz.*, the sump, or reservoir, and the control valve which permits reversing the direction of operation of the power cylinder. An unloading valve (see Fig. 30) is required with positive-displacement pumps to prevent building up excessive pressure at times of no demand from the work cylinder. The solid direction arrows indicate the path of flow when fluid is forced by the pump from the reservoir into the left-hand end of the cylinder which, in turn, forces fluid out the right-hand end and back through the 4-way valve to the reservoir. The dotted directional lines indicate the flow when it is desired to reverse the

piston direction. Unless a reversible pump is used, a 4-way valve or equivalent device for directional control is required in any system that must operate in two directions. The part of the system between the pump and the valve where the fluid always flows in the same direction is known as the power system, while the part beyond the valve is known by the name of the device it operates; for instance, as the landing-gear system or the flap system in the case of aircraft.[1] From this basic system any

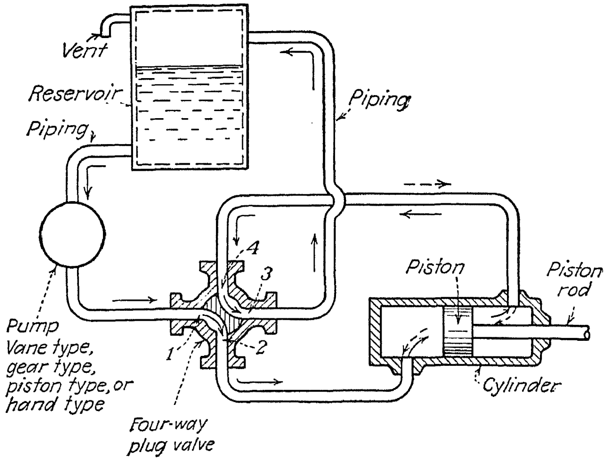


FIG. 2.—Schematic diagram of basic hydraulic system. With the 4-way valve in the position shown, fluid flow is in through port 1 and out through port 2; also in through port 4 and out through port 3. Turning the plug 90 deg connects ports 1 and 4 and ports 2 and 3, thereby causing the fluid to flow in the direction of the dotted arrows so as to reverse the direction of travel of the piston. (*Courtesy of Product Engineering.*)

hydraulic system can be derived. Additions can be made for the purpose of providing more than one source of power, for operating more than one cylinder, and for making operations more automatic or for increasing their reliability.

An elaboration of this system involving the use of several operating cylinders is shown in Fig. 3, which illustrates how cylinders can be operated independently or in parallel from the

[1] (*a*) See, "Aircraft Hydraulics," by Harold W. Adams, McGraw-Hill Book Company, Inc., New York, 1943.

(*b*) "Aircraft Hydraulic Control and Power Systems," *Product Eng.*, November, 1942, pp. 656–658, and December, 1942, pp. 731–733.

same control valve, or in series if desired. Any number of cylinders can be added on one control valve circuit in parallel since the hydraulic pressure, neglecting friction losses, is equal throughout all connected branches of a system in which fluid is free to flow. When cylinders are added in parallel on a system of limited supply

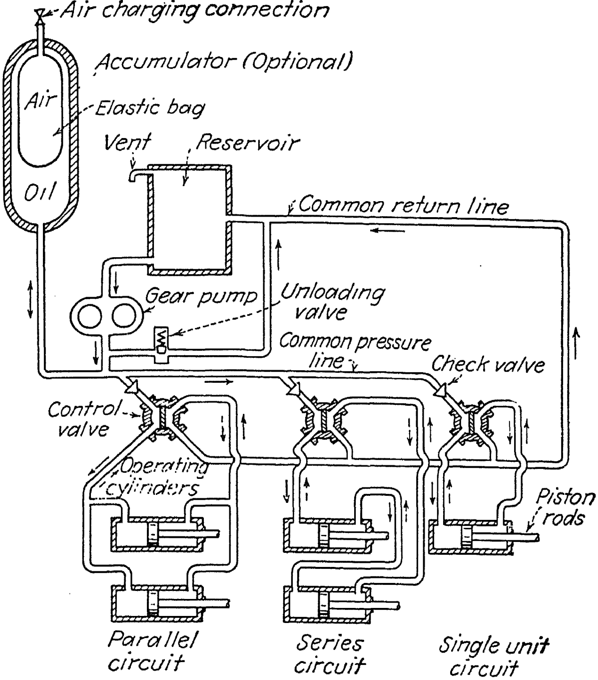


FIG. 3.—Multiple operation of hydraulic cylinders from a common power system.

capacity, however, the piston that requires the lowest unit pressure to move its load will operate first, and it will continue to move until it reaches the end of its travel or until the unit pressure builds up enough to permit another piston to move its load. This process of increasing pressure continues until the cylinder that requires the greatest unit pressure for operation has moved. This delayed-action trouble can be corrected by providing a control valve of more ample size or, if necessary, by installing an accumulator or by using a pump of greater capacity.

When cylinders are connected in series, as shown also in Fig. 3, an entirely different condition results. Fluid is then trapped between the cylinders, and only that on the exhaust side of the last cylinder in series can return to the reservoir. When the control valve on this circuit is operated, trapped oil is forced from the first cylinder into the second, and from the second into the third, if there is a third, and so on. Any leakage of fluid past one or more of the pistons, or in the piping between cylinders, will result in the pistons getting out of synchronism so that they do not reach the end of their travel together. A similar problem occurs when the trapped fluid expands or contracts as a result of a change in temperature. These troubles can be overcome by the use of synchronizing valves arranged to open automatically when the pistons reach the end of their travel, in order to permit a flow of fluid into the cylinder to replace leakage.

In multiple-cylinder systems such as those illustrated in Fig. 3 it may be necessary to install check valves in the individual control valve lines in order to prevent backfeed from a highly loaded cylinder into a lightly loaded one before its piston has reached the end of its travel.

Direct and Accumulator Systems.[1]—Where the fluid demand and rate of flow from a pump are well matched, particularly with a single machine (see page 1295), a direct system may be used that does not include an accumulator. In this case, the pump should have sufficient capacity to supply energy at the maximum rate required to operate the controlled equipment. In general, however, the accumulator is included for large single machines or for a number of machines supplied through a central system because it is possible, for short periods, to supply large hydraulic demands considerably in excess of the pump capacity. The pump then may be sized to meet the average demand rather than to provide the maximum short-time flow required. The accumulator serves to store energy delivered by the pump when the supply is greater than the demand and likewise to deliver energy when the demand rate exceeds the rate at which energy is supplied by the pump.

A schematic diagram of an accumulator system designed for 3,000 psi and suitable for operating large forging presses is shown in Fig. 4.[2] In this case, the hydraulic generating units consist

[1] See reference in footnote 2 and "Hydraulic Tables and Other Data," Baldwin-Southwark *Bull.* 150.

[2] From "Modern Practice in the Generation and Application of Hydraulic

of two reciprocating pumps. The accumulator is the hydro-pneumatic air-bottle type employing two air bottles and a combination air-and-fluid bottle. The bottles are charged with compressed air at the necessary pressure. The expansion of the compressed air in the hydro-pneumatic bottle displaces an equivalent volume of fluid which should be adequate for the maximum demand of fluid required to operate the equipment in use. It has been found in air-bottle accumulator practice that expansion

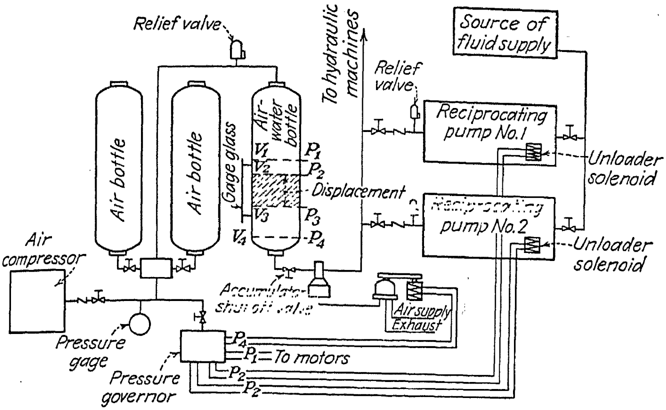


FIG. 4.—Schematic diagram of multiple-unit hydraulic system with air-bottle accumulators and reciprocating pumps. (*From ASME paper by J. E. Holveck, see footnote 2, p. 1290.*)

of the air between the higher and the lower pressures approaches the adiabatic and can be represented by the expression $p_2V_2^{1.35} = p_3V_3^{1.35}$.

In most hydraulic systems the total volume of air bottles required is determined by an allowable pressure variation in the order of 10 per cent between p_2 and p_3 which corresponds to a volume variation of about 8 per cent between V_2 and V_3 indicated by the shaded section in Fig. 4. The normal fluid displacement of the accumulator should be equivalent to the maximum demand

Power," by J. E. Holveck, ASME paper presented at Pittsburgh, Pa., Apr. 23, 1940, published by *Iron Steel Eng.*, Vol. 18, No. 5, May, 1941, pp. 58–66. Also available in reprint form from Worthington Pump and Machinery Corp., Harrison, N.J.

for an operating cycle as determined from flow charts.[1] It is considered good practice to provide a total hydraulic-fluid content in the air-and-fluid bottle of two to three times the normal fluid displacement. From these considerations the total volume of all the bottles will be from twelve to fifteen times the volume of fluid drawn off in a maximum operating cycle.

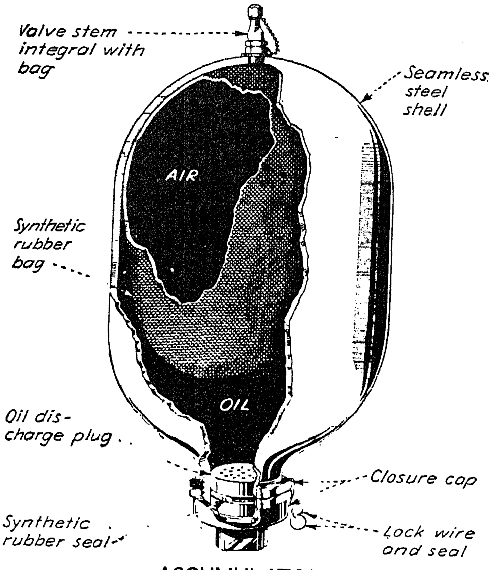


FIG. 5.—Accumulator chamber with rubber diaphragm to separate air from oil. (*Courtesy of Product Engineering.*)

The accumulator shutoff valve shown in Fig. 4 is needed as a safety measure to prevent loss of air through the piping system in case the fluid level falls to the point V_4 and its corresponding pressure p_4. This valve is automatically operated to respond to pressure variations. The automatic-unloader solenoids, also shown in Fig. 4, are intended to prevent excess pumpage by blocking open the pump suction valves in case the pressure reaches p_1

[1] See Holveck paper referenced in footnote on p. 1290; also "Hydraulic Tables and Other Data," Baldwin-Southwark *Bull.* 150.

corresponding to accumulator level V_1. Such automatic unloaders usually are set to operate in sequence so as to cut off pumpage in one pump at a time (see also Fig. 10 and accompanying text).

The basic principles of accumulator systems are much the same whether of large size for forging plants or of small size for aircraft. The details of construction are, of necessity, radically different, as described later. In all air-bottle types of accumulator systems provision should be made for charging the bottles with the required volume of air at the operating pressure and for maintaining this volume. Since the air is trapped, the only occasion for loss with all joints tight is through absorption of air by the fluid. This loss is small and requires only infrequent operation of the compressor or other source of supply.

Accumulators for oil-fluid systems on aircraft, self-contained systems for machine tools, and the like cannot have oil in direct contact with air on account of the explosion hazard. Hence such accumulators utilize a cylinder chamber and piston, a chamber and rubber diaphragm, a chamber and bellows, or gas-filled chambers. Such an accumulator chamber with a rubber diaphragm is illustrated in Fig. 5.

The *gravity-type accumulator*, shown diagrammatically in Fig. 6, is used for some large forge-plant installations. It is not suitable for aircraft, automotive, or marine use owing to its weight. In order to avoid excessive inertia forces, particularly on the downward stroke, the diameter of the ram should be large enough to avoid too high speed of travel. The principal advantages of the air-bottle type as compared with the weighted accumulator are:

1. The air-bottle type is lighter and requires less floor space and less costly foundations. Much greater accumulator effect can be obtained from a relatively small set of air bottles than is possible from a reasonable size of gravity accumulator.

2. The air-bottle type has no moving parts to be serviced while the gravity type requires maintaining the packing and ram.

3. The air-bottle type eliminates the shocks caused in the gravity type by the inertia of starting and stopping the weights. For instance, on a 2,000-psi hydraulic system, a ton of weights has to move through a vertical distance of 1 in. for each cubic inch of fluid passed in or out of the accumulator.

4. The air-bottle type tends to absorb fluid shocks from water hammer in the piping system whereas the gravity type must sustain these shocks.

5. The operating pressure in the air-bottle type may be readily changed by adjusting the pressure governor, while pressure changes with the gravity type are made by the more cumbersome method of adding or removing weights.

On the other hand, the gravity accumulator has an advantage where oil is used as the hydraulic medium in that no air is present

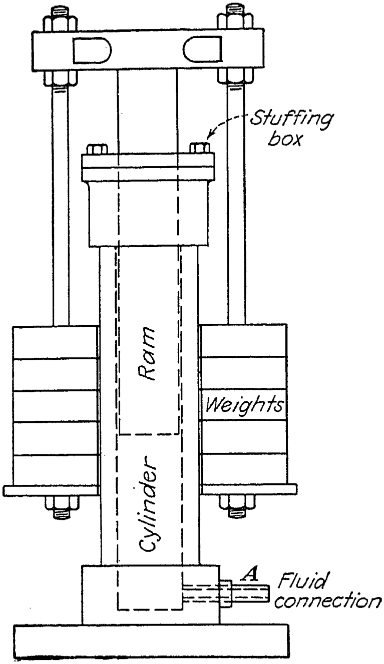


FIG. 6.—Sectional view of gravity-type accumulator.

to form an explosive mixture with oil vapor. Where oil is used in a large air-bottle accumulator system, a special sealing fluid must be floated on top of the oil in the accumulator so as to prevent oil vapor from mingling with the air, or an inert gas such as nitrogen must be substituted for air, or a diaphragm must be used to separate the oil from the air.

A limited amount of accumulator effect can be obtained with a plunger acting against springs using an arrangement similar to

the shock valve shown in Fig. 19, on page 1320. This device removes the above-mentioned explosion hazard with oil in contact with air, and at the same time dispenses with the excess weight required for gravity accumulators.

Hydraulic Intensifiers.—Sometimes it is desirable to have available high-pressure liquid at pressures higher than those of the system installed, or higher than those under which pumps operate most satisfactorily. In such cases an intensifier is indicated. An intensifier is essentially a device consisting of two cylinders of unequal diameters, having a common piston or ram, also of unequal diameters. The pressures in the two cylinders then vary inversely as the areas of the common ram. In calculating volumes of liquid required in connection with the design of the supply system, it should be considered that the ratio of volume of low-pressure liquid to that of high-pressure liquid is directly as the areas of the respective ends of the ram; *i.e.*, a larger volume of low-pressure liquid is required to produce the relatively smaller volume of high-pressure liquid.

Individual-unit or Self-contained Systems.—The action of many modern machine tools and presses is accomplished through individual-unit or self-contained hydraulic systems which constitute an integral part of each machine.[1] Such hydraulic systems consist essentially of a pumping unit driven by a constant-speed electric motor which delivers fluid through piping and control valves to a hydraulic motor or cylinder which actuates the tool mechanism. Valves, available in a great variety of types (see pages 1334 to 1340), are selected to obtain the desired sequences or cycles of operation with respect to time, force, and velocity. The application of hydraulics in the industrial field covers the widest ranges of power use. This is evidenced by the contrast between the tremendous rams of hydraulic presses working in tons, and the sensitive hydraulic controls in gear-grinding and milling machines which produce precision work within a fraction of a milinch.

There are two basic types of individual-unit or self-contained hydraulic systems used for operating machine tools. In the *constant-volume* system shown in Fig. 7, a constant-speed pump,

[1] See footnote 1, p. 1297, and footnote 1, p. 1300. For an account of large, self-contained hydraulic presses, see, "Improved Hydraulic Presses for Wartime Requirements," by J. M. Maude, *Trans. ASME*, Vol. 65, No. 4, May, 1943, pp. 287–296.

usually of the gear or vane type, discharges a constant volume of oil at a constant pressure. Flow from the pump is throttled to regulate the rate of delivery to the operating cylinder. Excess pumpage is by-passed through a relief valve which is set at the highest pressure needed in the system. At the end of each stroke of the operating piston, its direction of travel is reversed automatically by a trip (not shown) which actuates the 4-way valve.

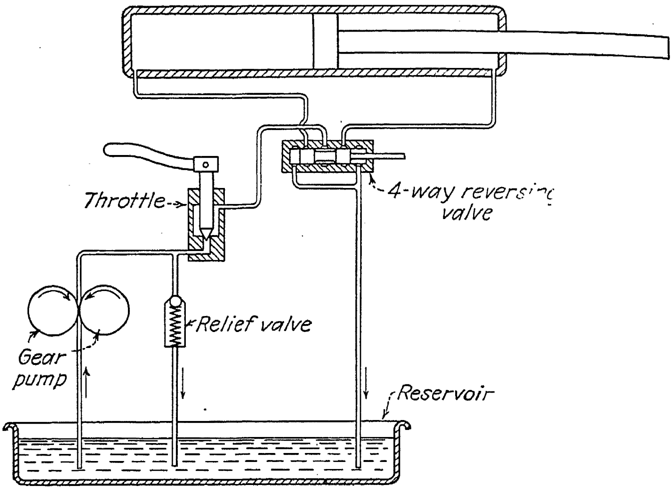


FIG. 7.—A *constant-volume* hydraulic system for an individual unit.

In the *variable-volume* system shown in Fig. 8, a constant-speed pump, usually of the multipiston variable-displacement type, delivers, as required, a variable volume of hydraulic oil ranging from maximum flow in one direction, through zero in neutral position, to maximum flow in the opposite direction. Oil under pressure is thus delivered to one end of an operating cylinder (or to a hydraulic motor in some cases) forcing the piston to move through its working stroke. Oil drainage from the opposite side of the piston flows back to the pump suction or to the reservoir. At the end of each stroke, the slide strikes an adjustable stop and actuates a linkage which automatically reverses the direction of pump discharge, thus causing the piston to move in the opposite direction. With the particular tool illustrated in Fig. 8, a large-

diameter piston rod is used so as to give different piston areas on the up and down sides. This difference in area gives full thrust with relatively slow piston travel on the down, or power, stroke and a reduced thrust with faster travel on the up, or return, stroke. Differences in oil volume between the two ends of the cylinder are compensated for in this hookup through the action of the relief and check valves.

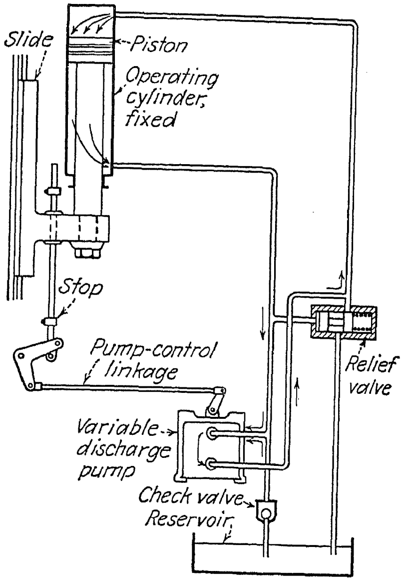


FIG. 8.—A *variable-volume* hydraulic system for an individual unit.

Complex Hydraulic Systems.—Numerous variations and combinations of the simple circuits outlined above are in common use for machine tools[1] as well as for aircraft.[2] Fluid motors are used to secure rotary motion such as is imparted by an electric motor. Variable displacement pumps are used whereby fluid motor speeds may be chosen from full pump speed, to standstill, to full speed in the reverse direction. This type of drive can be

[1] See "Hydraulic Control Affords Design Flexibility," by C. E. Grosser, *Machine Design*, December, 1940, p. 37. See also oil company bulletins referenced in footnote on p. 1300.

[2] For references, see footnote, p. 1288. See also the following papers published

applied to provide speed regulation for machine tools and for conveyers, steel and paper mills, hoisting machinery, etc. Characteristics of the individual circuits as regards flow of the fluid to secure the required motion will influence the selection of pipe size, but other factors influencing piping design such as the pressure will be independent of the type of circuit involved.

High- and Low-pressure Piping.—Depending on the type of equipment served, any number of main, auxiliary, or pullback cylinders or fluid motors may be used. In some downward-stroke hydraulic presses, such as that illustrated in Fig. 9, the stroke of the main ram is divided into the clearance stroke and the working stroke. The long clearance stroke of the main cylinder may be actuated through a low-pressure hydraulic system, whereas pressure is applied through the working stroke by the high-pressure system. With such installations the smaller pullback rams usually can be served to good advantage by the high-pressure system. The high-pressure system should be designed to sustain the maximum working pressures developed therein, which may go up to 3,000 psi or higher. High velocities and considerable pressure drop can be permitted in the high-pressure system because the high pressures used give a large margin for such losses. The pressure drop in the low-pressure system, however, should be more carefully considered so that the loss of head in the piping will not result in slow or erratic operation of the controlled equipment.

The drain piping through which spent fluid returns eventually to the pump suction is not shown in Fig. 9, but this usually is subject to disposal through the 4-way (or equivalent) valve used

---

in *Trans. ASME*, Vol. 66, No. 7 (October, 1944):

(*a*) "An Introduction to Aircraft Hydraulic Systems," by Howard Field, Jr., p. 569.

(*b*) "Controversy over Choice of Medium for Aircraft Power Transmission," by R. L. Haymen, p. 577.

(*c*) "The Evolution of the Hydraulic Pump as Applied to Aircraft," by Dale Herman, p. 583.

(*d*) "The Modern Hydraulic Reservoir: How It Provides Micron-range Filtration and Pump Supercharging," by W. W. Thayer, p. 589.

(*e*) "Aircraft-engine Temperature Control," by W. A. Ray, p. 595.

(*f*) "High- and Low-pressure Airplane Hydraulics in Europe," by Jean Mercier, p. 599.

(*g*) "Maintenance of Aircraft Hydraulic Systems in the Field," by R. E. Middleton, p. 605.

(*h*) "Some Characteristics of Rotary Pumps in Aviation Service," by R. J. S. Pigott, p. 615.

to control the working and return strokes of the press. In most cases the spent hydraulic fluid returns to a reservoir or sump from which it is drawn again as needed by the pump suction. With large multiple-unit water systems, water may be drawn from the reservoir by a centrifugal booster pump which discharges in turn

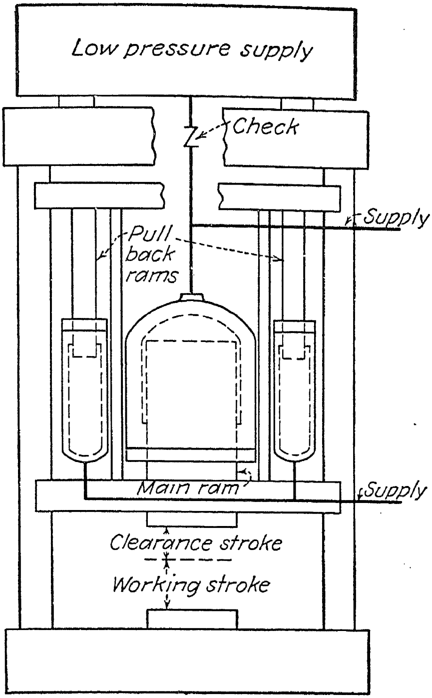


FIG. 9.—Simple hydraulic press having one main cylinder and two pullback cylinders.

to the suction of the high-pressure pump or pumps which may be either centrifugal or reciprocating, depending on the working pressure of the power system.

With an upward-stroke press where the action of the main ram is inverted from that shown in Fig. 9, the force of gravity often is employed to return the ram and other moving parts to their original position, thus eliminating pullback cylinders. This is

the simplest type of press which is used mainly for hot-plate work in plastics and rubber. Such presses have a long clearance stroke and a short working stroke where it is particularly advantageous to use low-pressure hydraulic power for taking up the clearance stroke and save applying high-pressure power until the final operation of the working stroke.

Types of Pumps.—In discussing pumps it is convenient to consider them first with reference to the two basic types of hydraulic systems which they serve: *accumulator* systems and *direct* systems. Modern practice calls for constant-speed electric motors for both types. Owing to space limitations and to the very considerable number of pump designs involved, it is impracticable here to go beyond the bare mention of some of the principal types of pumps. Information about centrifugal pumps for hydraulic work can be obtained from well-known manufacturers of such pumps for all purposes. Reciprocating pumps for hydraulic work are of special designs built by a limited number of manufacturers. Many varieties of rotary pumps using oil as the hydraulic fluid are made by different manufacturers; for a description of the different types, reference may be made to bulletins issued by the oil companies.[1]

Accumulator-system Pumps.—The accumulator system applies, generally, to all installations where two or more hydraulic machines are supplied from a central hydraulic generating station. This system also is applicable to single hydraulic machines, usually of large size, where the average demand for high-pressure fluid over each operating cycle is considerably less than the momentary demand. This is apt to be true where the clearance stroke is actuated with low-pressure fluid, as shown in Fig. 9. The three types of pumps commonly used with accumulators are

1. Reciprocating, constant speed and constant displacement. Output is controlled by starting and stopping the motor in the case of small pumps, or through a by-pass or unloading valve of one sort or another. With large pumps this may be done by automatically blocking open the suction valves when enough fluid has been pumped for the time being, or by providing a separate unloading valve in the piping circuit (see Fig. 30). Popular types

[1] See (*a*) "Hydraulic Oils and Their Applications," Sun Oil Company, *Tech. Bull.* B-4.

(*b*) "Hydraulic Systems, Circulating Oils for Machine Tools," Socony-Vacuum Oil Co., Inc.

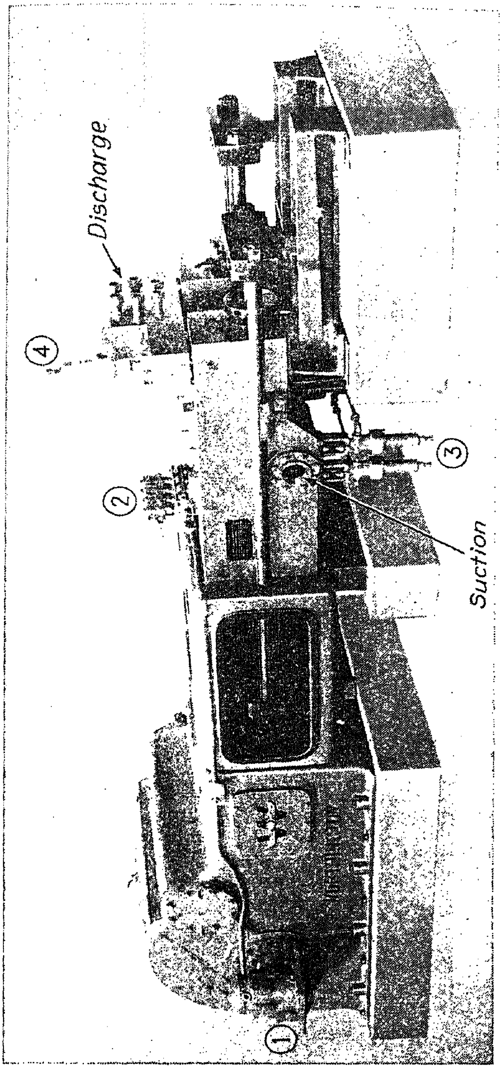


FIG. 10.—Horizontal double-acting duplex reciprocating pump suitable for large high-pressure systems using either oil or water as the hydraulic fluid. (*Courtesy of Worthington Pump and Machinery Corporation.*)

of reciprocating pumps are either duplex or triplex double-acting in the horizontal position, and triplex single-acting in the vertical position. The horizontal double-acting duplex pump shown in Fig. 10 is a 300-hp size suitable for working pressures up to 3,000 psi. This pump is unloaded with an automatic suction valve lifting device, an enlarged view of which is shown also. The numbered parts in Fig. 10 perform the following functions in unloading the pump: (1) is a synchronizing electric distributor on the end of the crankshaft which controls the unloading device so as to obtain a gradual acceleration or deceleration of fluid flow when delivery is cut on or off; (2) is a gang of solenoid-operated air valves, one for each cylinder, which control compressed air lines for actuating the lifting devices shown at (3) and in the enlarged sectional view; (3) are the lifting devices, one for each cylinder, which act to block open the suction valves as required to prevent overpressure on the system. With the suction valves blocked open, the plungers operate on suction pressure only since the discharge valves remain seated. Under these conditions the power required by the pump is only enough to overcome mechanical friction. At point (4) is a spring-loaded relief valve which opens on overpressure if the unloading valve fails to function.

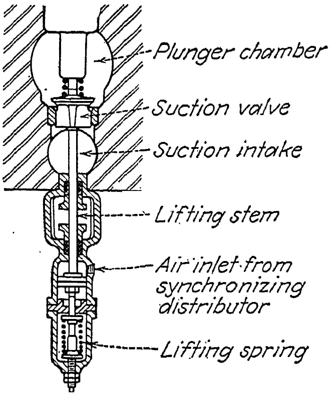


FIG. 10*a*.—Enlarged section of valve lifting device shown at (3) in Fig. 10.

2. Centrifugal, constant speed and variable capacity. With a centrifugal pump, a control for variable and intermittent fluid demand is not necessary since the capacity and head characteristics of this type of pump will meet these flow demands. As the maximum design pressure is approached, the pump delivers less and less fluid until a point of no pumpage is reached. Under these conditions an automatic by-pass valve is needed to recirculate enough fluid back to storage in order to prevent overheating the fluid and pump at zero demand. Owing to design limitations centrifugal pumps are restricted to lesser pressures than are usual with reciprocating pumps. The general characteristics of cen-

trifugal pumps are too well known to warrant much description here. Multistage centrifugal pumps are used in hydraulic installations requiring moderately high pressures combined with large gallons-per-minute capacities. They may be used with or without accumulators, with either water or oil as the hydraulic fluid. In operations requiring pressures not over 1,500 to 2,000 psi during comparatively long cycles of operation, no accumulator is needed. Where the flow fluctuates and high-pressure peaks occur, an accumulator becomes desirable in that it permits a smaller pump to handle the job than would be necessary for equal performance without the accumulator.

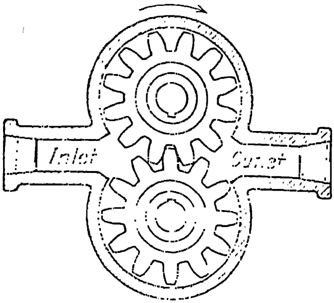


FIG. 11.—Gear variety of rotary pump suitable for medium-pressure systems using oil as the hydraulic fluid.

3. Rotary, constant speed and constant capacity. Like the reciprocating pump, its delivery should usually be controlled by starting and stopping the pump, or through an unloading valve which by-passes fluid back to storage. In the gear variety of rotary pump shown in Fig. 11, oil is trapped in the pockets between the case and the teeth of the revolving gears and carried around the periphery of both gears from the suction side of the pump to the discharge side. Return of oil to the suction is prevented by the meshing of the teeth. In the vane variety of rotary pump illustrated in Fig. 12, free-sliding vanes held out by centrifugal force are used to carry trapped oil from suction to discharge. In another variety of vane pump, variable discharge is obtained by adjusting the eccentricity of the outer ring. Rotary pumps are limited to oil as the fluid medium, and to lower pressures than are common with reciprocating pumps owing to their susceptibility

to wear and leakage. For an understanding of the action of rotary pumps as well as the radial piston and variable-stroke reciprocating types mentioned in the next paragraph, reference may be made to pump manufacturers' catalogues,[1] to technical magazine articles,[2] and to oil company bulletins.[3]

Direct-system Pumps.—To be suitable for a direct system other than with a small self-contained unit, pumps should be selected which are capable of having their rate of delivery varied with the

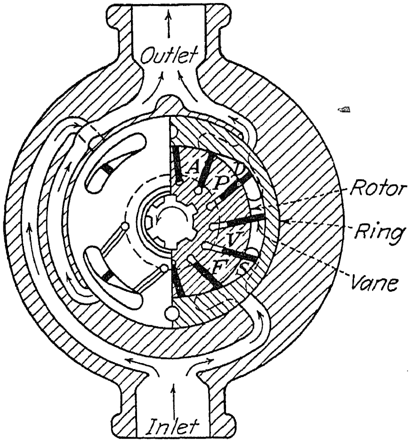


FIG. 12.—Vane variety of rotary pump suitable for medium-pressure systems using oil as the hydraulic fluid. As rotor revolves counterclockwise, vanes V and F follow contour of oval-shaped ring and space S increases in size forming a vacuum which draws in fluid. Rotor revolves farther, trapping liquid between V and F until they come to position of vanes A and P and liquid comes to space D. As rotor continues to turn, vanes A and P are pressed inward by contour of ring, space D keeps decreasing, and trapped fluid is forced into outlet.

load. The three principal types are (1) centrifugal, (2) multi-piston, and (3) variable-stroke reciprocating. The characteristics of centrifugal pumps for serving hydraulic power systems already have been mentioned.

The radial-piston variable-displacement pump shown in Fig. 13 is a constant-speed variable-capacity type in which packless

[1] For identification of pump manufacturers with different types of pumps, see footnote on p. 1300.

[2] See for instance, "Hydraulic Controls for Present and Post-war Products," *Product Eng.*, February, 1944.

[3] See footnote, p. 1300.

pistons oscillate radially within a rotor. The displacement of the pistons may be varied manually or automatically from zero to full stroke or full capacity, by means of a movable rotor ring.

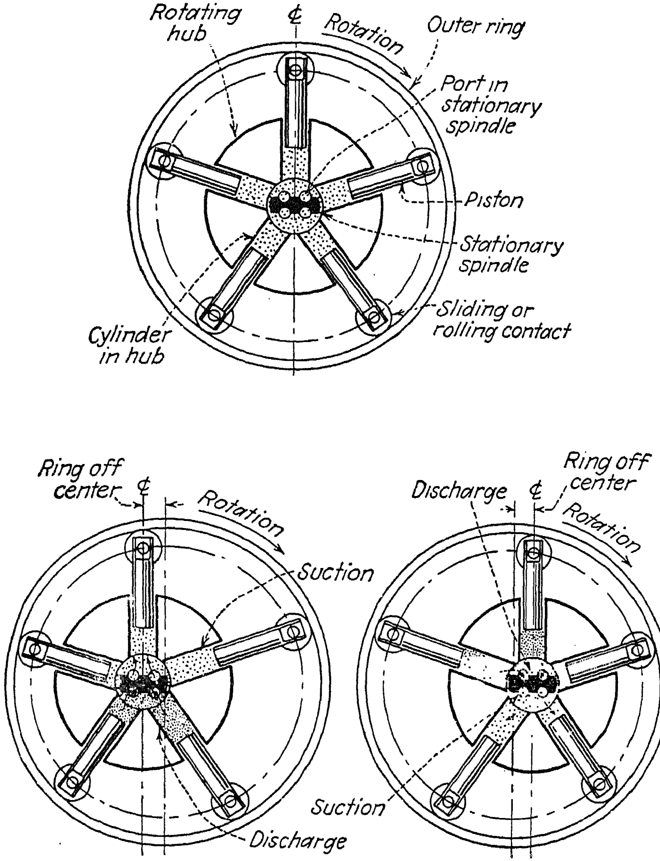
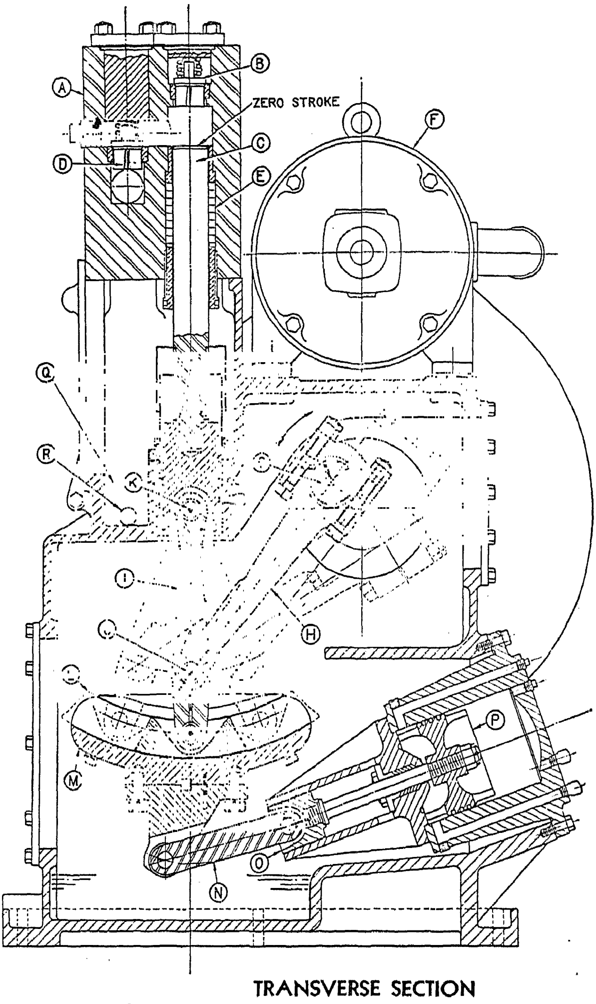


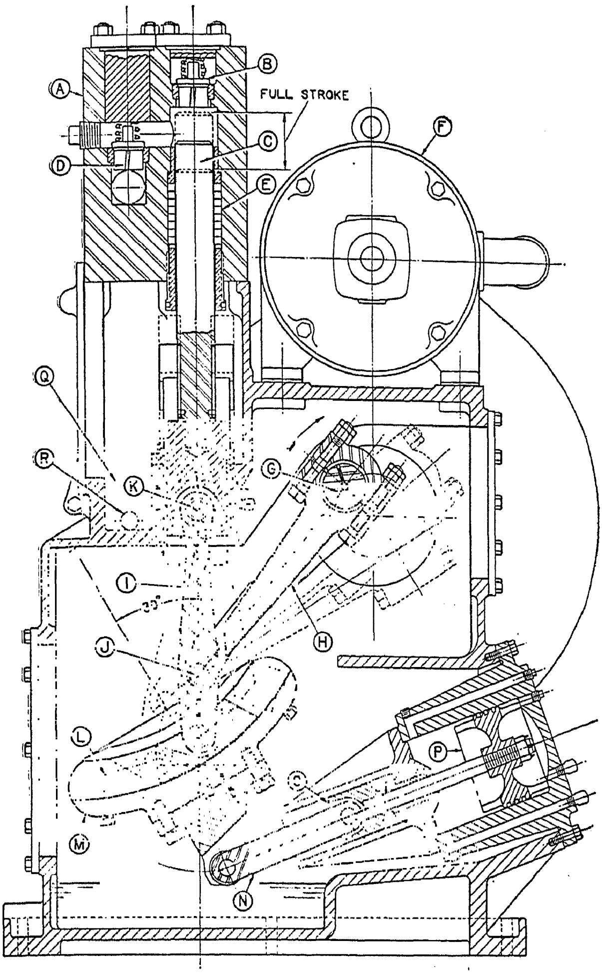
Fig. 13.—Radial piston variable-displacement pump suitable for high-pressure direct systems using oil as the hydraulic fluid (see text).

When the ring and the hub are concentric, the latter can rotate without any movement of the plungers. By moving the ring off center in either direction it is possible to reverse the direction of the



TRANSVERSE SECTION
Showing Pump adjusted for ZERO STROKE

FIG. 14.—Reciprocating variable-stroke pump (Aldrich-Groff) suitable for
p. 1308). (*Courtesy of*



TRANSVERSE SECTION
Showing Pump adjusted for FULL STROKE

high-pressure direct systems using oil or water as the hydraulic fluid (see text, *Aldrich Pump Company.*)

flow of oil without stopping the pump or changing its direction of rotation. The displacement of each piston is transmitted from the supply pressure to the discharge pressure without the use of pump valves or plunger packing. In another variety of the multipiston type, the pistons act in an axial instead of a radial direction and are actuated by two rotating elements having an adjustable angularity.

The multipiston variable-displacement pump is applicable, in general, only to a direct system, and its use is confined either to self-contained machines where the fluid demand favors this type of generating unit, or to direct applications calling for a variable pressure demand with certain conditions. Lubricating oil has to be used as the fluid medium with this type of pump which is limited to working pressures not in excess of 2,000 to 3,000 psi, depending upon the continuity of service.

The reciprocating variable-stroke pump (see Fig. 14) refers to that type of constant-speed pump having outside-packed multiple plungers and pump valves. The displacement of the plungers may be manually or automatically varied from zero to full stroke or capacity by various mechanical means. Variable-stroke pumps are applicable only to the direct system, and usually to self-contained machines where the fluid demand favors this type of generating unit. Under certain conditions they can supply a variable pressure demand. The variable-stroke reciprocating pump is suitable for use with oil or water as the fluid medium and is unlimited in working pressure. The capacity can be controlled gradually, in stepless straight-line fashion, to give any desired continuous delivery between zero and maximum, as compared to variation by steps or with intermittent flow as done with reciprocating pumps employing suction-valve control.

The pump shown in Fig. 14 is driven by a constant-speed electric motor *F* through herringbone reduction gearing from motor to crankshaft *G* which operates the connecting rods *H*. The other end of each of rods *H* is pivotally connected to links *I* by the connecting-rod pins *J* and serves to oscillate the same about the axis of crosshead pins *K* as centers. To the bottom end of each of links *I* is pivotally connected a guide block *L* which slides back and forth within a smooth curved track on the "stroke transformer" *M*. The radius of curvature of this track is equal to the vertical distance therefrom to the center of crosshead pin *K*, measured along the center line of link *I* with the stroke transformer positioned as

shown for *zero* stroke in the left-hand view. In this position, rotation of cranks G merely oscillates links I and blocks L back and forth, and no stroking motion is imparted to the pump plungers C. For zero stroke the pump *delivery* also is reduced to zero.

When the plungers C are being reciprocated with their *full* stroke, the stroke transformer M has been tilted through an angle of 30 deg about the longitudinal axis of its pivotal trunnion-journals X, as shown in the right-hand view.

For pumping strokes and deliveries anywhere between zero and the maximum, the stroke transformer M is positioned correspondingly between the two positions shown. This is done by means of

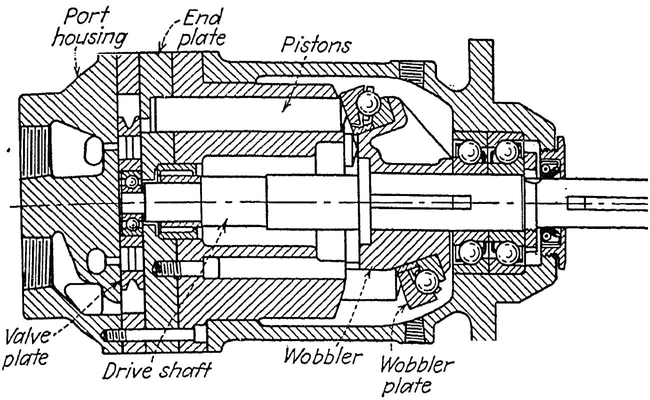


FIG. 15.—Axial-piston type of hydraulic motor employing a wobbler plate.

link N and crosshead O under control of a manually operated screw mechanism with a handwheel, or else automatically by a hydraulic servo-piston P working in the double-acting cylinder as shown.

Hydraulic Motors.—In addition to the piston and cylinder applications of hydraulic power described elsewhere in this chapter, there are hydraulic motors of the gear type, wobbler type, radial- and axial-piston type, etc., which can be used for obtaining rotary power. The axial-piston type of motor employing a wobbler plate, shown in Fig. 16, is one of these devices suitable for use with oil as the hydraulic fluid. Motors are available in fixed-displacement and variable-displacement types similar to the corresponding types of pump construction previously described. In the fixed-displacement type, speed changes are made by varying the volume

of oil flowing through it from the pump. This arrangement works out well where only one motor is driven per pump. In the variable-displacement type, wide speed ranges are obtainable by varying the displacement of the motor in addition to the control, if any, of oil supply from the pump. By means of a lever, a thrust bearing, and a sliding key in the shaft it is possible with the variable-displacement version of the type shown in Fig. 15 to adjust the angularity of the wobbler plate while rotating so as to vary the speed of the motor or to reverse its direction of rotation.

With Sundstrand, Waterbury, and similar hydraulic transmissions a variable-displacement, multipiston pump delivers oil to a constant-displacement fluid motor. Speed of the driven member is controlled by varying the rate of discharge and direction of delivery from the pump. Pump and motor may be assembled in one unit or located separately.

PROPERTIES OF HYDRAULIC FLUIDS

The two basic types of fluid used in hydraulic systems are water and oil. Water has been, and still is, used extensively for operating large presses in forge shops and for similar applications, particularly on multiple-unit systems where several presses are run from a single piping system supplied by a common set of pumps. Although oil is used to some extent for such applications, its principal field is in self-contained units for performing certain operations on individual machine tools and in small multiple-unit systems for control equipment on aircraft and mobile equipment of all sorts. Hydraulic actuation lends itself well to automatic control, to adjustment of speed, and to reversal of motion in a smooth, vibrationless way which gives a cushioning effect not readily obtained with a mechanical drive.

Incompressibility of Liquids.—Water and oil in common with other liquids are practically incompressible, having a modulus of elasticity in compression only about 1 per cent that of steel. Oil, like water (see page 266), has a coefficient of compressibility of about 0.00005 for each atmosphere of added pressure, and a modulus of elasticity in compression of about 300,000 psi at room temperature. This is the same property as what is referred to as K under Water Hammer on page 295, where it is said, "The bulk modulus of elasticity K of water and other liquids is approximately 300,000 psi, within a limit of about 10 per cent plus or

minus, varying somewhat with the pressure and temperature (see page 266), which is sufficiently accurate in view of the uncertainty of other factors entering the problem." One aspect of the incompressibility of liquids is manifest in the need for air chambers or other means for cushioning water hammer where there are sudden stoppages of flow. This subject is discussed in a general way on pages 298 and 1031, and with particular reference to hydraulic-power transmission piping on pages 1317 to 1322.

Rational Flow Formula.—In considering the flow of water as well as oil through hydraulic-power transmission piping, it is now the usual practice to use the "rational formula" method described on pages 107 to 137. This is advisable owing to the fact that the usual empirical formulas for the flow of water (see pages 269 to 288) were not derived for the high velocities employed in hydraulic-power transmission piping. With oil, of course, use of the rational formula is essential because of the considerable variation in density and viscosity between oils, at different flow temperatures. Hence the flow tables and charts for both water and oil which are given in this chapter are based on the rational formula and have been extended to higher velocities than considered elsewhere in this handbook. The equivalent resistance of bends, fittings, and valves can be found from Table XIV on page 100.

Need to Dissipate Frictional Heat.—The use of high pressures for actuating hydraulic-power systems permits employing high fluid velocities with relatively large frictional losses in the piping and control equipment. As a result, considerable mechanical energy is lost in friction and goes to heat the hydraulic fluid. The amount of heat developed in this way can be computed from consideration of the mechanical equivalent of heat given under Energy on pages 7 and 8. For instance, 1,000 lb of water per minute flowing through a pipe line with a pressure drop p_λ of 100 psi represents a mechanical energy loss of $1{,}000 \times 100 \times 2.31 = 231{,}000$ ft-lb per min which, in turn, represents $231{,}000 \div 33{,}000 = 7$ fluid horsepower (see pages 289, 290), or $231{,}000 \div 778 = 297$ Btu per min. This relation can be reduced to general terms through consideration of the fact that the amount of water to be heated is the same quantity as that being pumped through the frictional resistance. Hence, neglecting losses to the pipe and surrounding air, the temperature rise Δt in degrees Fahrenheit for each trip around the circuit is $\Delta t = 297 p_\lambda 10^{-5}$ in the case of water which has a specific gravity and a specific heat of unity.

With oil the specific heat C and the specific gravity S both vary with temperature as shown in Fig. 32 on page 209. For the conditions obtaining in hydraulic systems the specific heat of oil usually has a value of about 0.50 Btu per lb per deg F. The specific gravity ordinarily is less than unity, depending both on the characteristics of the oil and on its operating temperature. The general expression for the temperature rise of oil or other liquids for each trip around the circuit, neglecting losses to the pipe and surrounding air, is

$$\Delta t = \frac{0.00297 p_\lambda}{CS}.$$

Where a reservoir and a hydraulic accumulator are present in the circuit, or where there is much branch piping to be considered, due allowance should be made for the amount of active storage capacity in the system. For instance, if only one-half the total quantity of fluid in the system is in circulation at any one time, the average temperature rise will be only half as fast as would be the case if the system contained no reserve supply of fluid. This condition, coupled with the need for considering the amount of heat going to warm the piping and equipment or lost as heat emission from the same, may have to be taken into account by making a heat balance. For this purpose the heat H developed in Btu by the frictional losses of a flow weight of w lb can be computed from the relation

$$H = \frac{2.31 p_\lambda w}{778S} = \frac{p_\lambda w}{336.8S}.$$

It is thus evident that, if the same fluid traverses the circuit often enough, there will be an appreciable rise in fluid temperature unless surplus heat is removed in some way. Frequently, temperature conditions can be kept in balance by heat losses from the surface of the piping and equipment that is not insulated. Under some circumstances, however, it is necessary to provide a heat exchanger or a cooling coil in the fluid reservoir. Likewise where a pump is merely churning the hydraulic fluid at times of no-power demand, it is usually necessary to provide some means of bleeding a small amount of the fluid back to the reservoir to prevent overheating the pump and its fluid contents.

Relative Merits of Water and Oil.—Both water and oil possess advantages of their own as hydraulic [illegible] different applica-

tions. For instance, water is inexpensive, is noninflammable, and has a low viscosity which is conducive to low pumping cost. On the other hand, oil is noncorrosive, is a better lubricant than water, and acts as a viscous seal which makes possible the use of some varieties of rotary pumps that would not perform well with water at high pressure. Certain properties of water and some of the numerous oils used for hydraulic fluid are discussed in following sections.

WATER AS A HYDRAULIC FLUID

The *physical* and *thermal properties* and the *chemical composition* of water are given on pages 264 to 269. For use in the rational formula, the *viscosity* of water at different temperatures can be obtained from Fig. 16 on page 119. When water is used as the fluid in hydraulic-power transmission systems, an emulsifying oil is added in a concentration of 2 to 3 per cent by weight as a corrosion inhibitor for the piping and equipment. Centrifuges are employed to separate the water from the oil periodically and start over with a fresh mix. Such systems should be checked for rubber gaskets or other materials that might be adversely affected by the emulsion, causing swelling or slight disintegration.

Water as a fluid for hydraulic-power transmission is used at much higher pressures and velocities than are found in other applications. This creates a number of special problems which are peculiar to hydraulic piping and which involve unusual construction and severe shock conditions under some circumstances. These problems are discussed in succeeding sections.

Flow Charts.—A *water velocity* of 35 ft per sec is deemed conservative in hydraulic-power transmission piping, and velocities of 50 ft per sec or higher are sometimes employed. The large pressure drops accompanying these velocities usually do not represent a big percentage of the initial pressure since lines are short and pressures high. An initial pressure of 1,000 psi is considered low for water and would be employed, perhaps, in connection with a centrifugal pump. Pressures of 2,000 to 3,000 psi developed by reciprocating pumps are usual in large forge-plant installations.

Owing to the heavy-wall pipe and high water velocities used in hydraulic-power transmission systems it has been necessary to provide special flow charts to suit these conditions. The charts furnished in Figs. 16 to 18 for standard-weight, extra-strong, and

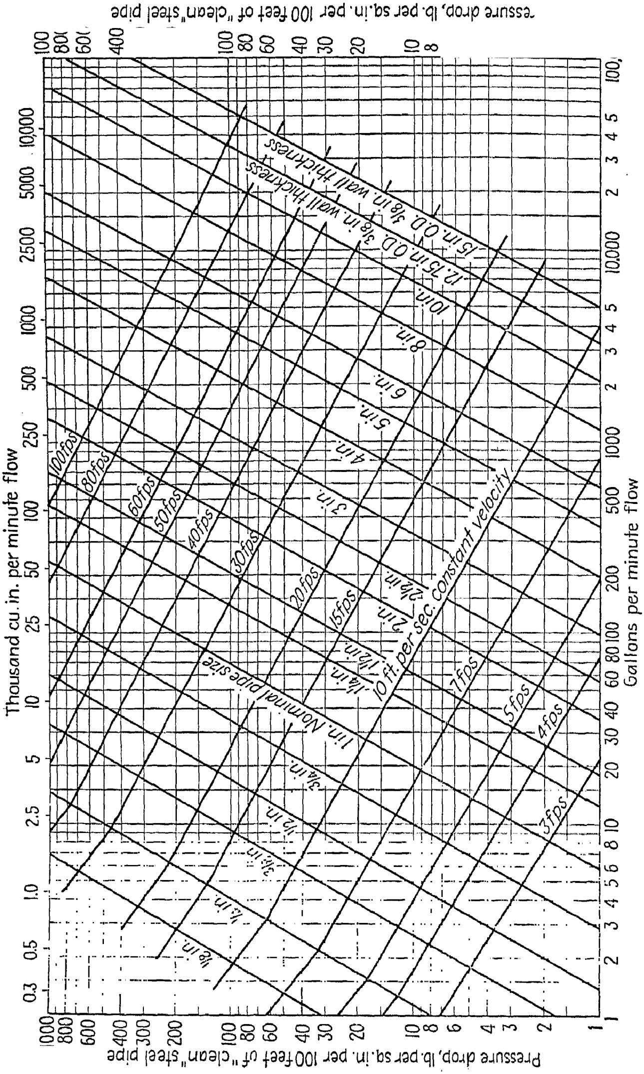


FIG. 16.—Flow chart for standard-weight (Schedule 40) pipe carrying water.

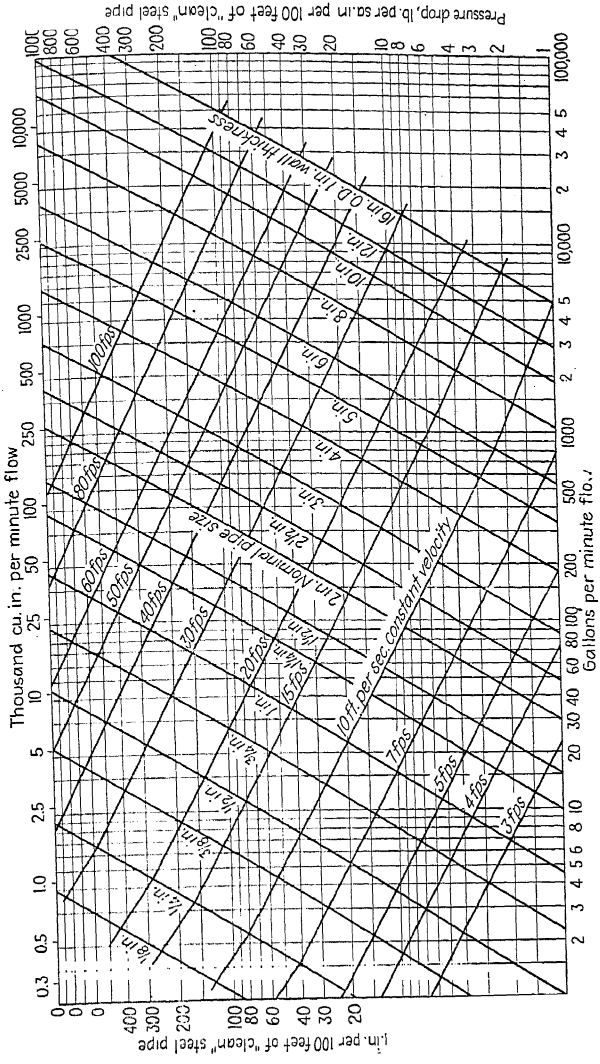


Fig. 17.—Flow chart for extra-strong (Schedule 80) pipe carrying water.

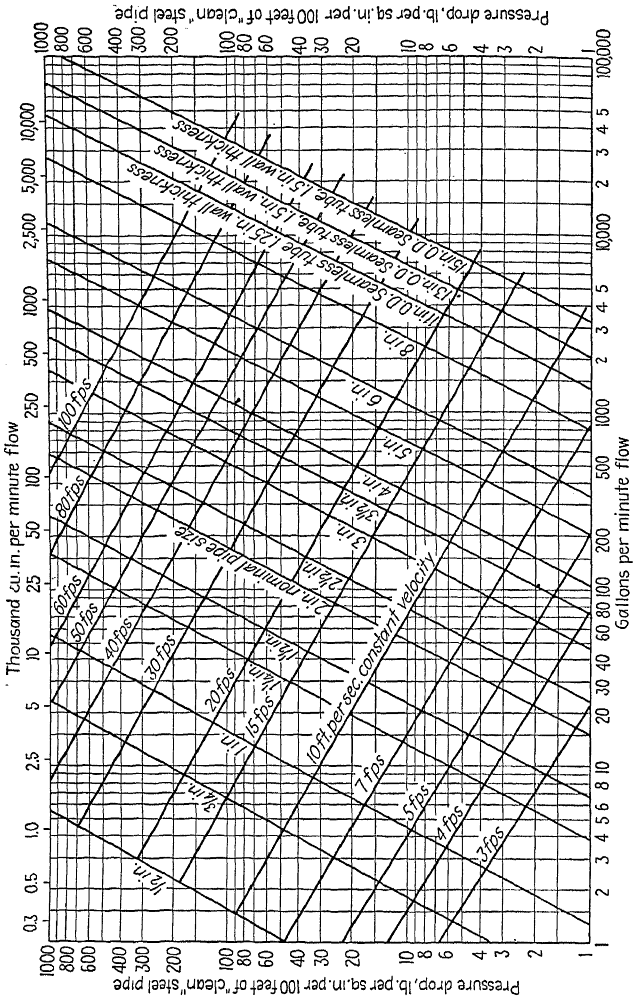


FIG. 18.—Flow chart for double-extra-strong pipe carrying water.

double-extra-strong pipe cover the usual range of conditions for water flow through "clean"[1] steel pipe in hydraulic systems. If problems are encountered beyond the scope of these charts, it is recommended that they be solved by the rational formula method given on pages 107 to 137. For lower velocities, see the rational formula charts given in Figs. 33*a*, *b*, *c*, *d*, *e*, *f*, and *g* on pages 212 to 218.

Water velocities in *pump-suction* and *drain* lines are in keeping with usual practice, being of the order of 2 to 5 ft per sec. On large systems using reciprocating pumps for high-pressure delivery, a centrifugal booster pump sometimes is used to deliver water from the storage reservoir into the suction of the high-pressure pumps at 40 to 60 psi. In other installations a separate low-pressure pump is used to deliver water for actuating the clearance stroke of the press rams before the high-pressure water is turned on for the working stroke.

Velocity Head.—Where water velocities are as high as is the case here, the amount of pressure loss needed to create this velocity may become a factor in design. The pressure losses required to produce water velocities of from 1 ft per sec to 100 ft per sec given in Table I were computed from the fundamental relation $h = v^2/2g$ (see page 81 for definition of symbols) and converted into pounds per square foot and specific gravity instead of feet of head. These tabular values are given for a specific gravity of 1 but can be converted to suit other conditions simply through multiplying by the specific gravity of any liquid under flow conditions.

Where the pressure loss going to produce velocity head is appreciable, it should be added to the frictional loss through the piping and control valves in arriving at the effective pressure left for operating presses and other equipment.

Water Hammer.—The subject of water hammer is covered in a general way on pages 291 to 300. Owing to the high velocities obtaining in hydraulic systems and to the rigidity of the heavy-walled pipe used to withstand the severe pressure, water hammer may be very high in case of sudden interruption of flow. Referring to Table L on page 296, it appears that for each foot per second of reduction in velocity within the critical period a shock pressure of about 60 psi may be expected. Hence with flow velocities of 25 to 50 ft per sec it is possible under worst conditions to set up shock pressures of 1,500 to 3,000 psi, with correspondingly

[1] See Formula B, p. 130.

large energy blows which may tend to tear the piping loose from its anchors. As stated on page 293, the energy developed in inch-pounds per cubic inch of water affected by the change in velocity is $p_0^2 \div 2K$, where p_0 is the shock pressure in pounds per square inch, computed as explained above, and K is the bulk modulus of elasticity for water which is 300,000 psi numerically.

TABLE I.—LOSS IN PRESSURE REQUIRED TO PRODUCE WATER VELOCITIES SHOWN[1]

Velocity, ft per sec V	Pressure loss, psi p_v	Velocity, ft per sec V	Pressure loss, psi p_v	Velocity, ft per sec V	Pressure loss, psi p_v	Velocity, ft per sec V	Pressure loss, psi p_v
1	0.01	26	4.56	51	17.6	76	39.0
2	0.03	27	4.92	52	18.3	77	40.0
3	0.06	28	5.29	53	19.0	78	41.1
4	0.11	29	5.68	54	19.7	79	42.1
5	0.17	30	6.08	55	20.4	80	43.2
6	0.24	31	6.49	56	21.2	81	44.3
7	0.33	32	6.91	57	21.9	82	45.4
8	0.43	33	7.35	58	22.7	83	46.5
9	0.55	34	7.80	59	23.5	84	47.6
10	0.68	35	8.27	60	24.3	85	48.8
11	0.82	36	8.75	61	25.1	86	49.9
12	0.97	37	9.24	62	25.9	87	51.1
13	1.14	38	9.75	63	26.8	88	52.3
14	1.32	39	10.3	64	27.6	89	53.5
15	1.52	40	10.8	65	28.5	90	54.7
16	1.73	41	11.3	66	29.4	91	55.9
17	1.95	42	11.9	67	30.3	92	57.1
18	2.19	43	12.5	68	31.2	93	58.4
19	2.44	44	13.1	69	32.1	94	59.6
20	2.70	45	13.7	70	33.1	95	60.9
21	2.98	46	14.3	71	34.0	96	62.2
22	3.27	47	14.9	72	35.0	97	63.5
23	3.57	48	15.6	73	36.0	98	64.8
24	3.89	49	16.2	74	37.0	99	66.2
25	4.22	50	16.9	75	38.0	100	67.5

[1] Computed by formula $p_v = 0.00675\,Sv^2$ for water weighing 62.4 lb per cu ft which corresponds to $S = 1$. To get the value of p_v for water or other liquid at a different specific gravity, multiply the tabular values by the specific gravity of the liquid under flow conditions.

The kind of *anchors* and *supports* used with large hydraulic systems may be of considerable importance where severe water hammer is experienced. With several forging presses or other hydraulically operated equipment taking water through a header from a common set of pumps, certain combinations of demand sometimes occur which produce enough shock to make the whole piping system whip about. Under these circumstances serious

damage may be done to the equipment connections and piping unless the line is properly restrained.

Since thermal elongation is of small proportions in most hydraulic systems, it seldom is necessary to make much provision for expansion due to change in temperature, and some designers prefer to anchor such systems rigidly to the building steel at frequent intervals. On the other hand, excessive rigidity has its drawbacks, and some other designers prefer to cushion the whipping of the pipe with spring shock absorbers (see Figs. 8 and 9, pages 736 and 737) rather than to attempt to fasten it down too rigidly. Where shock absorbers are used on hydraulic systems, they should be a size or two larger than the recommendations given for power-house piping. With shock absorbers, hydraulic piping can be supported in box guides so as to permit free longitudinal movement except as restrained by the shock absorbers. This arrangement is said to be very effective in cushioning shocks which are due to water hammer and which might be destructive to fixed anchors.

With well-arranged air-bottle accumulator systems and smoothly operating control valves, shock pressures may be held to a level where they pass unnoticed or give little or no trouble. With gravity-type accumulators, however, it often is necessary to use shock-absorber valves such as that illustrated in Fig. 19. It is common practice to put one near the accumulator, one near each of the presses, and one at the end of each long run of straight pipe. Shock valves are sometimes used with air-bottle accumulator systems too if bad shock conditions are experienced without them. In such a system no relief valve should be needed near the accumulator which provides its own air cushion. Shock-relief valves should be placed as close as possible to the point at which flow is interrupted. This usually means on the branch lines close to the presses and other equipment being operated. The idea is to cushion the initial shock somewhat before the wave has a chance to rebound in full force throughout the entire system.

In the shock-absorber valve shown in Fig. 19 the plunger operates through a stuffing box which does not permit fluid to escape from the system. Movement of the plunger is controlled by heavy springs. Small sizes usually are arranged with double springs one inside the other, while large sizes have multiple springs. The springs and plunger assembly can be mounted on the branch outlet of a tee or other fitting convenient for the purpose. Shock valves accomplish their purpose by absorbing excess energy through

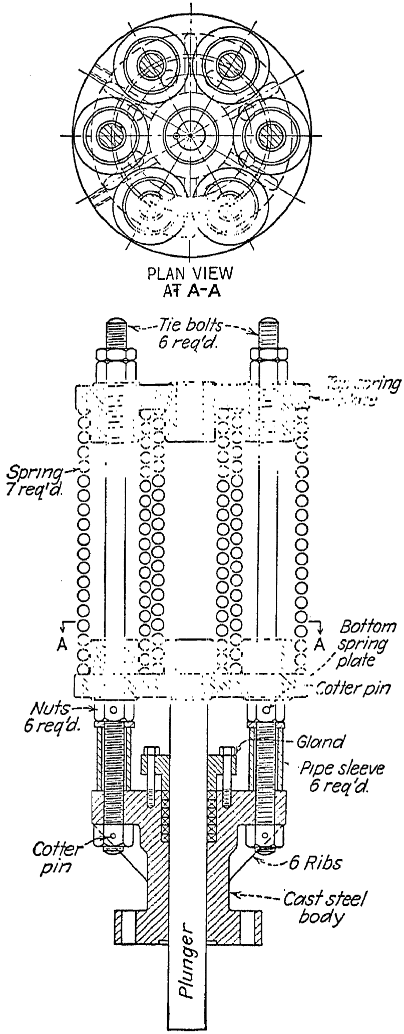


FIG. 19.—Shock-absorber valve for weighted-accumulator hydraulic system. (*Courtesy of Baldwin-Southwark.*)

having the plunger move out against the resistance of the springs, following which the plunger returns to restore this energy to the fluid in the proper part of the cycle. No fluid is wasted aside from incidental leakage.

As previously stated, the total shock energy set up by water hammer is $p_0^2 \div 2K$ in.-lb per cu in. of fluid affected by the instantaneous change of velocity, but not all of this energy necessarily has to be absorbed in order to reduce shock to reasonable proportions. Practically, shock valves are usually designed to have an operating range from $\frac{1}{2}$ to $1\frac{1}{2}$ normal pressure without striking the stops. This, in the case of 2,400-psi normal pressure, corresponds to a range from 1,200 to 3,600 psi and provides on the upper side for an extinguished velocity of 20 ft per sec. On the downswing the bottom spring plate may approach or strike the inboard stop, in which position it is shown in Fig. 19. The beneficial action of the shock valve is in retarding or cushioning the rate of pressure surge and in dampening out the oscillations of the fluid column.

An approach to selecting the right diameter of plunger and spring resistance for a shock-absorber valve can be made through computing the inch-pounds of shock energy that have to be absorbed by the springs and proportioning them for developing this amount of work in being pushed from the half compressed to the fully compressed position. Since the load is applied through pressure against the plunger, the diameter of the plunger should be sufficient to compress the springs about one-half their travel under normal pressure and to compress them fully under maximum design shock conditions. According to Baldwin-Southwark[1] a rough approximation of the size of plunger can be found from the formula $A_1/A = 5.14\ p\ 10^{-4}$ where A is the internal area of the pipe, A_1 the area of the plunger, and p is the normal pressure on the system in pounds per square inch. In a more exact solution, consideration should be given to the length of line, ratio of pipe diameter to wall thickness, physical characteristics of springs, limits of travel, amount of dampening required, etc.

Water hammer sometimes occurs in a long *exhaust* line leading from an *operating valve* back to the *reservoir*. In such cases it is associated with the inertia set up by high exit velocity from the valve and through the line. When the valve cuts off, the inertia of the fluid column may create a partial vacuum between the

[1] See "Hydraulic Tables and Other Data," Baldwin-Southwark *Bull.* 150.

column and the valve so that when the inertia is expended the resulting pressure unbalance returns the fluid column to the valve and thus produces hammer shocks. Relief may be effected by the use of a suppressor of one sort or another, or the tendency to form a vacuum may be avoided through providing a vacuum breaker near the valve. This usually takes the form of a check valve on a stub off the exhaust line, with the check arranged to admit air when the line is under vacuum and to prevent the escape of fluid when the line is under pressure.

OIL AS A HYDRAULIC FLUID

Manufacturers of hydraulic equipment usually specify the grade of oil best suited to their product and their recommendations should be followed. Consideration has to be given to working temperatures, pressures, ambient temperatures, and the type of service. Hydraulic oils generally used have viscosities at 100 F ranging from about 150 sec Saybolt Universal for light oils to about 1,000 sec Saybolt Universal for heavy oils. The lighter oils are used on pumps and machines requiring low to medium pressures; the heavy grades are used where high pressures are encountered. An ordinary operating temperature for hydraulic oils is 150 to 165 F, which usually can be maintained without the use of a heat exchanger or a cooling coil in the reservoir. Hydraulic oils used with power-pumping equipment must have adequate lubricating properties. The use of nonlubricating oils such as Lockheed fluid is restricted to manually operated braking devices and the like.

The properties of oils in general are described on pages 200 to 210, in addition to which viscosity, specific gravity, density, and temperature corrections for oils and other liquids are discussed on pages 118 to 129. More specific information about hydraulic oils is given in succeeding sections.[1]

Viscosity.—High pressures are maintained in two ways between moving parts in hydraulic systems: (1) by closely ground and lapped clearances assisted by a *viscous* seal consisting solely of the oil film, and (2) the use of cup leathers and similar packing around piston rods, plungers, and the like. The first means is employed

[1] See also: (*a*) "Hydraulic Fluids," by Norman E. Miller, *Product Eng.*, September, 1940, pp. 407–410, and October, 1940, pp. 450–454.

(*b*) "Hydraulic Systems," Socony-Vacuum Oil Co. bulletin.

(*c*) "Hydraulic Oils and Their Applications," *Tech. Bull.* B-4, Sun Oil Co.

(*d*) "Lubrication of Industrial Machinery," *Tech. Bull.* B-5, Sun Oil Co.

for sealing internal clearances in rotary and piston oil pumps; the second is used for external packing with either oil or water.

The effectiveness of the viscous seal afforded by the oil film between the pump parts falls off rapidly if the oil loses viscosity too fast with increase in operating temperature, whereas frictional losses through the piping and control equipment mount rapidly if the oil is too viscous. Consequently an economic balance must be struck between losses due to slippage in the pump and frictional losses through the system so as to come out with the best over-all result. For any given hydraulic system there will be an optimum viscosity for obtaining the maximum over-all operating efficiency, and any viscosity lower than the ideal will permit pump slippage which will more than offset any saving in line losses. Hence, for hydraulic purposes where the fluid has to operate over a considerable temperature range, an oil should be selected with a flat characteristic for change in viscosity with temperature. This property is referred to as the *viscosity index* or "VI,"[1] and a high VI rating means that the oil changes less in viscosity with temperature than would an oil of lower VI.

According to the recommendations of Vickers, Inc., the VI must be above 78, and the normal operating temperature should not exceed 155 F for satisfactory results. High and low temperatures and abnormal pressure conditions are said to require special engineering consideration. Vickers offer the recommendations in the accompanying table on the viscosity of oil for hydraulic systems where their equipment is used:

OIL VISCOSITY RECOMMENDATIONS IN SSU AT 100 F FOR HYDRAULIC SYSTEMS USING VICKERS PUMPS AND HYDROMOTIVE CONTROLS
(For machine and press installations where the ambient temperature is 50 to 90 F)

General operating conditions	Vane pump installations			Piston pumps or piston-type hydraulic transmissions
	With flow-control valves	Without flow-control valves	With Vicker's fluid motors	
Average conditions	150	225	225	315
Low-pressure conditions, below 500 psi and special conditions[1] . .	100	150	225	225

[1] Should be given special engineering consideration.

[1] See ASTM Standard, D567.

According to Norman Miller[1] oils having a viscosity of 150 to 300 SSU at 100 F are in the desired range for normal service. Where extremes of pressure or temperature are encountered, oils having viscosities ranging from 200 to 1,000 SSU at 100 F sometimes are employed. Where low temperatures are encountered, such as installations subjected to winter weather, oils having viscosities as low as 50 to 100 SSU at 100 F may be used. All these oils if properly applied, however, should have approximately the same viscosity at the normal operating temperature of the system.

In Miller's opinion an oil having a viscosity index of less than 60 will, generally speaking, contribute to loss of power and reduce over-all efficiencies. Many good fluids can be obtained that possess VI's ranging from 60 to 125. Broadly speaking, it is not safe to say that the one with the high VI will necessarily be better than the low. The high one will be superior, however, speaking strictly from a viscosity temperature and hydraulic slip standpoint. If one attempts to make an over-all comparison on a basis of VI only, he is likely to go astray on such specifications as demulsibility, wear resistance, pour point, neutralization number, and similar characteristics. (*Note.*—Reference may be made to Miller's excellent articles for a discussion of these points.)

The viscosity-temperature relationships for several typical oils commonly used in hydraulic systems are shown in Fig. 20. Viscosity is given both as seconds Saybolt Universal or as kinematic viscosity in centistokes. A more accurate conversion from one to the other may be made by means of Fig. 18 on page 123, or by using the table of values given in ASTM Specification D446. Although viscosity curves generally plot as straight lines on the sort of cross-sectional paper used in Fig. 20, the lines may bend somewhat as a result of additions of compounds to the oil to provide some particular characteristic such as increased wear resistance, improved demulsibility properties, or better viscosity index. This tendency is particularly evident for Lockheed 21 fluid shown in Fig. 20.

The specific gravity of the several oils at 60 F with reference to water at 60 F also is shown in Fig. 20. Specific gravity of an oil at any other temperature for use in flow formulas for determining pressure drop may be obtained from Fig. 32 on page 209.

[1] See footnote 2*a*, p. 1322.

It should be noted that brands of oil are shown in Fig. 20 only to demonstrate the viscosity-temperature characteristics of representative hydraulic oils. This does not constitute an endorsement or recommendation of them in preference to numerous other satisfactory hydraulic oils marketed by many of the oil

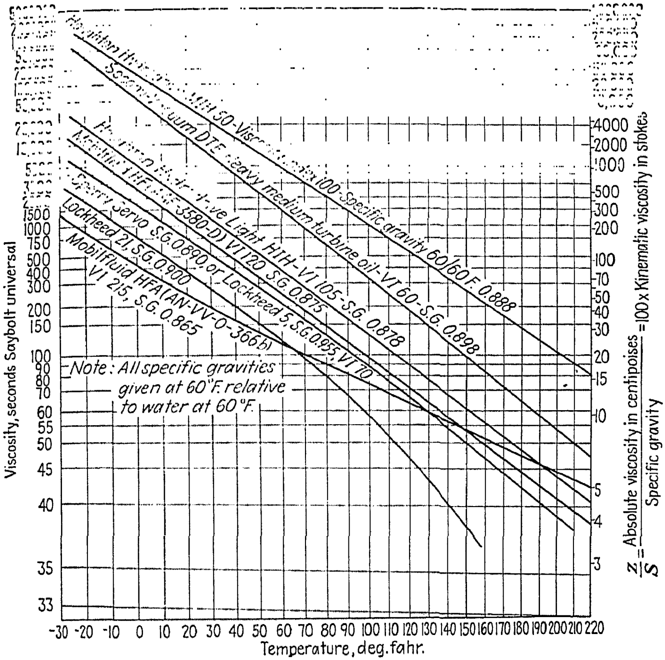


FIG. 20.—Temperature-viscosity characteristics and specific gravities of typical oils used in hydraulic systems.

companies. An oil should be selected with characteristics to suit the individual application as defined by the manufacturer of the equipment.

Oil Flow Data.—Oil velocities in hydraulic systems of all sorts are considerably lower than the water velocities customary with large multiple-unit presses and the like, but they still are on the high side with respect to oil velocities used in other applications. A good velocity range for discharge piping is 7 to 15 ft per sec,

but 20 ft per sec often can be permitted when this velocity is attained for only a short time or a small part of the cycle.[1] The design velocity actually chosen depends on the viscosity of the particular oil used and on the length of line required. Pump suction and drain lines carrying hydraulic oil are designed ordinarily for velocities of 2 to 4 ft per sec. Piping for low-pressure auxiliaries and controls should be amply sized so that pressure drop will not cause sluggish actuation of the controls and thus delay action of the main equipment. Since this piping usually is of small capacity, making it of generous size will not add much to the overall cost. Flow velocities through pipes and tubes of various sizes can be determined from Fig. 21 which is reproduced through the courtesy of Vickers, Inc., of Detroit.

The principal reason for using a lower velocity with oil than with water is the viscosity of the kind of oil which has to be used with rotary pumps in order to prevent excessive slippage. With an oil having a viscosity suitable for use with a rotary pump, the frictional loss through the piping would be excessive if the velocity were as high as could be used to advantage with water.

As stated previously, oil-flow problems should be solved by the rational formula method described on pages 107 to 137, or from charts or tables computed by that method. Such a chart for the flow of hydraulic oil through clean, smooth tubes of aluminum, brass, copper, lead, glass, etc., is given in Fig. 22.[2] The chart can be applied also to "very smooth" seamless-drawn *steel* tubes carrying oil. The example shown in dash lines is for oil having a viscosity of 340 SSU and a specific gravity of 0.9 at 50 F flowing at the rate of 10 gpm through 1 by 0.049-in. tubing. The pressure drop obtained from the chart is 0.29 psi per ft of tube, or 29 psi per 100 ft of tube.

In Table II are given the oil-carrying capacities and friction losses of "clean" steel pipe for the range of velocities and capacities encountered in hydraulic-power transmission piping. For the

[1] See "Hydraulic Piping, Determining (Oil) Pressure Drop and Power Losses," by Ransom Tyler, *Product Eng.*, November, 1941. Available also in reprint form as *Bull.* 90010 of The Oilgear Company, 1301 West Bruce St., Milwaukee, Wis.

[2] From "Aircraft Hydraulics," by Harold W. Adams (see footnote, p. 1288). The curves were drawn for a friction factor of $f = 0.0014 + \frac{0.125}{N^{0.32}}$ given in "Heat Transmission," by W. H. McAdams, McGraw-Hill Book Company, Inc., 2d ed., 1942, p. 119.

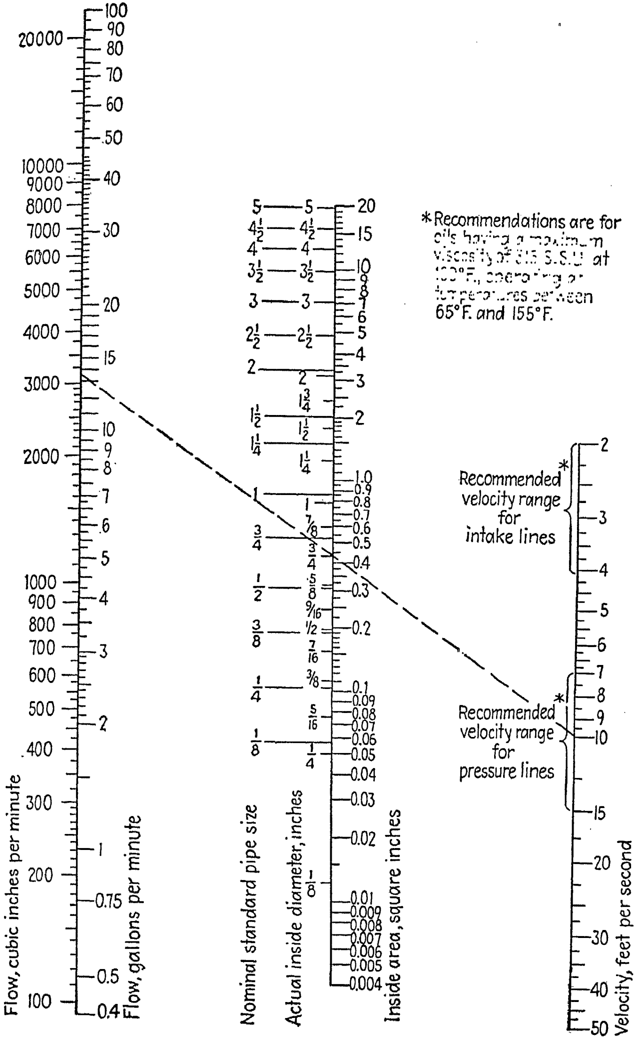


FIG. 21.—Nomographic chart for finding flow capacities of pipes and tubes at recommended flow velocities. (*Courtesy of Vickers, Inc.*)

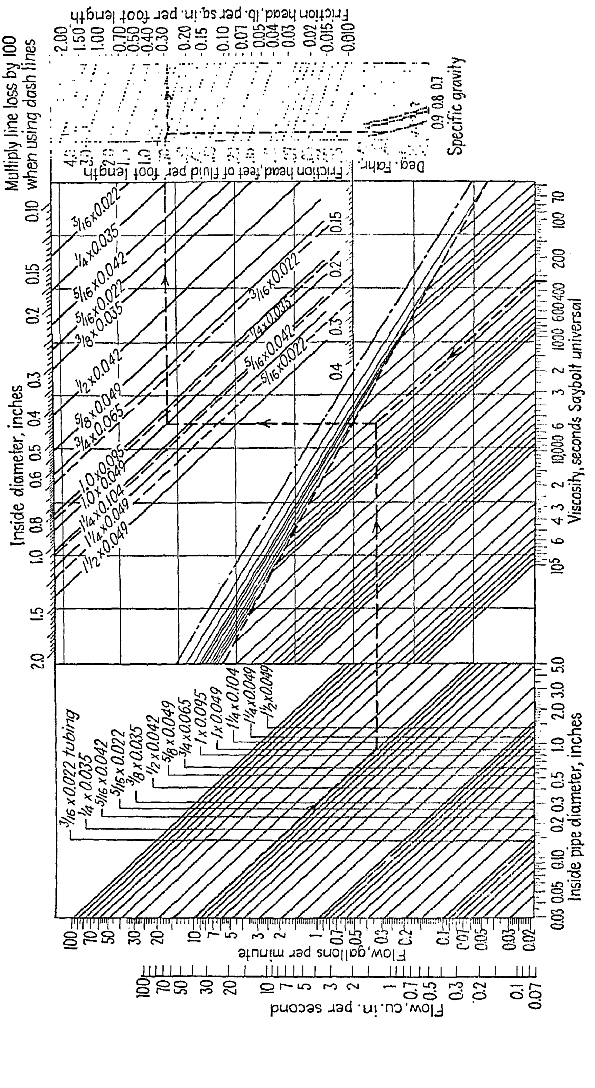


FIG. 22.—Flow chart for flow of hydraulic oil through tubing. (*Reproduced by permission from "Aircraft Hydraulics," by Harold W. Adams.*)

convenience of the user, carrying capacities are given in both gallons per minute and cubic inches per minute, velocities are given in feet per second and corresponding velocity heads in pounds per square inch. Pressure drops in the viscous-flow region are set in boldface type to distinguish them from the viscous region where pressure drops are set in ordinary type. As a further demarcation between viscous and turbulent flow, the SSU corresponding to the critical point for each rate of flow is shown in the tabulation. Pressure drops for higher SSU values are on the viscous side of the critical point, and for lower SSU values on the turbulent side. In computing the pressure drops tabulated, allowance was made for the fact that unstable flow may occur throughout the whole critical region (see Fig. 15*a*, page 108), and the larger pressure drop has been shown when any doubt exists as to which formula would obtain.

Since Table II is intended for use in selecting pipe sizes for systems using oil at medium to high pressure, the wall thicknesses shown are on the heavy side, *viz.*, Schedule 80 for nominal sizes ⅛ to ⅜ in. inclusive, and Schedule 160 for nominal sizes ½ in. and larger. Pressure drops through these weights of pipe when carrying *water* can be approximated from the 50 SSU column by taking 60 to 80 per cent of the tabular values.

The solution of oil-flow problems can be extended from one condition of viscosity and specific gravity to another through consideration of the relation of these terms in the rational formula, at least between the limits of 100 and 4,000 SSU.[1] Within these limits resistance to flow in the *viscous* region is directly proportional to the product of SSU and specific gravity. For instance, if p_λ is computed at 50 psi at 200 SSU, it would be double this or 100 psi at 400 SSU if the specific gravity remained the same. With the same oil, however, the drop in temperature which increased the viscosity from 200 to 400 SSU also would increase the specific gravity to about 1.01 times what it was with the 200 SSU viscosity. Hence the corrected answer would be $100 \times 1.01 = 101$ psi.

Also within the aforesaid limits, the extension from one condition to another with *turbulent* flow follows the approximate empirical relationship $F = CS'(SSU)^{1/4}$ where F is the factor by which the pressure drop computed for a viscosity of 200 SSU and a specific gravity of 0.875 should be multiplied to get the pressure drop

[1] See footnote 1, p. 1326.

TABLE II.—OIL-CARRYING CAPACITIES AND FRICTION LOSSES FOR "CLEAN" STEEL PIPE IN HYDRAULIC-POWER TRANSMISSION SYSTEMS

(Based on rational formula, see pages 107 to 137. For water use 60 to 80 per cent of pressure drops shown for oil in 50 SSU column)

Nominal pipe size, in., schedule number, and average inside diam, in.	Carrying capacity		Velocity, ft per sec	Velocity head, psi for[2] $S = 0.9$	SSU for critical point, SSU = $6.8G/d$	Pressure drop in psi per 100 ft of pipe[1]					
						Viscosity, SSU					
	Gal per min	Cu in. per min				50	100	200	500	1,000	4,000
1/8	0.5	116	4.43	0.12	16	54	111	225	570	1,150	4,655
Sched.	1.0	231	8.85	0.48	32	108	222	450	1,140	2,300	9,310
80	1.5	347	13.3	1.07	48	283	333	675	1,710	3,450	14,000
$d =$	2.0	462	17.7	1.90	64	472	537	900	2,280	4,600	18,620
0.215	2.5	578	22.1	2.97	80	645	835	1,125	2,850	5,750	23,310
1/4	1.0	231	4.48	0.12	23	29	59	120	304	615	2,485
Sched.	2.0	462	8.96	0.49	45	92	118	240	608	1,230	4,970
80	4.0	924	17.9	1.94	90	287	394	480	1,215	2,460	9,940
$d =$	6.0	1,396	26.9	4.40	135	590	650	720	1,822	3,690	14,910
0.302	8.0	1,848	35.8	7.76	180	987	1,475	1,800	2,430	4,920	19,880
3/8	2.5	578	5.71	0.20	40	31	40	82	208	420	1,700
Sched.	5.0	1,155	11.4	0.78	80	87	125	164	415	840	3,400
80	7.5	1,733	17.1	1.78	120	176	210	246	623	1,260	5,100
$d =$	10	2,310	22.8	3.15	160	293	305	520	832	1,680	6,800
0.423	15	3,465	34.3	7.14	240	662	860	1,055	1,340	2,500	10,200
1/2	2.5	578	4.70	0.13	36	13	27	54	150	303	1,227
Sched.	5.0	1,155	9.40	0.54	72	59	77	107	300	606	2,454
160	10	2,310	18.8	2.15	144	185	241	294	600	1,212	4,908
$d =$	15	3,465	28.2	4.82	216	376	490	580	900	1,818	7,362
0.466	20	4,620	37.6	8.57	288	632	780	950	1,200	2,424	9,816
3/4	5.0	1,155	5.42	0.18	55	16	24	48	122	248	1,000
Sched.	10	2,310	10.8	0.71	110	50	65	97	246	497	2,010
160	20	4,620	21.6	2.83	220	168	225	280	490	994	4,020
$d =$	30	6,930	32.5	6.40	330	350	465	580	736	1,491	6,030
0.614	40	9,240	43.3	11.4	440	580	760	930	1,400	1,988	8,040
1	10	2,310	6.15	0.23	90	13	17	24	60	122	493
Sched.	20	4,620	12.3	0.92	180	44	57	78	120	244	986
160	30	6,930	18.5	2.07	300	90	120	150	181	366	1,480
$d =$	40	9,240	24.6	2.67	450	149	206	252	350	483	1,972
0.815	60	13,860	36.9	8.26	750	312	410	500	635	732	2,960

[1] Computed for oil having a specific gravity S of 0.90 at 60/60 F. To adjust the tabulated pressure drops to suit an oil having a different gravity, multiply by the ratio of its specific gravity at 60/60 F to 0.90. Pressure drops in bold-face type are in the viscous flow region.

[2] Computed for $S = 0.90$ from the relation $p = 0.00675SV^2$. To adjust the tabulated velocity heads to suit a different gravity, multiply by the ratio of that gravity under flow conditions to 0.90.

TABLE II.—(*Concluded*)

Nominal pipe size, in., schedule number, and average inside diam, in.	Carrying capacity		Velocity, ft per sec	Velocity head, psi for[2] $S = 0.9$	SSU for critical point, SSU = $6.8G/d$	Pressure drop in psi per 100 ft of pipe[1]					
						Viscosity, SSU					
	Gal per min	Cu in. per min				50	100	200	500	1,000	4,000
1¼	15	3,465	4.56	0.13	90	5.0	6.5	**9.3**	**22**	**45**	**181**
Sched.	30	6,930	9.11	0.50	180	17	24	30	**44**	**90**	**362**
160	50	11,550	15.2	1.40	300	41	56	69	**100**	**149**	**603**
d =	75	17,325	22.8	3.15	450	84	111	138	175	**224**	**906**
1.160	125	28,875	38.0	8.74	750	210	270	324	411	522	**1,510**
1½	20	4,620	4.57	0.13	100	3.7	4.0	**8.5**	**20**	**34**	**138**
Sched.	50	11,550	11.4	0.65	250	21	30	37	**50**	**84**	**340**
160	100	23,100	22.8	3.15	500	68	92	112	152	**168**	**680**
d =	150	34,650	34.2	7.08	750	141	185	226	305	**252**	**1,020**
1.338	200	46,200	45.6	12.6	1,000	230	314	383	508	672	**1,360**
2	25	5,775	3.58	0.08	100	1.4	**1.8**	**3.6**	**9.0**	**17**	**67**
Sched.	50	11,550	7.17	0.31	200	8.8	11	14	**17**	**33**	**134**
160	100	23,100	14.3	1.24	400	25	33	40	55	**66**	**268**
d =	200	46,200	28.7	5.00	800	104	135	165	210	266	**536**
1.689	300	69,300	43.0	11.2	1,200	205	266	325	412	523	**805**
2½	50	11,550	4.52	0.12	160	2.3	3.6	4.4	**6.2**	**13**	**54**
Sched.	100	23,100	9.05	0.53	320	9.5	12	15	20	**26**	**108**
160	200	46,200	18.1	1.98	640	30	39	47	60	76	**215**
d =	300	69,300	27.2	4.48	960	60	78	95	121	153	**323**
2.125	400	92,400	36.2	7.92	1,280	110	131	160	203	248	**430**
3	100	23,100	5.93	0.21	260	3.2	4.1	5.0	**6.0**	**12**	**46**
Sched.	200	46,200	11.9	0.86	520	11	15	18	23	**23**	**92**
160	300	69,300	17.8	1.92	780	24	31	38	48	55	**138**
d =	400	92,400	23.7	3.41	1,040	38	49	60	76	97	**183**
2.626	500	115,500	29.6	5.30	1,300	57	74	90	114	145	**230**
4	200	46,200	6.92	0.29	400	2.7	3.6	4.4	**3.9**	**7.7**	**31**
Sched.	400	92,400	13.8	1.16	800	9.2	11	14	18	23	**63**
160	600	138,600	20.8	2.62	1,200	20	25	30	38	48	**94**
d =	800	184,800	27.6	4.62	1,600	33	40	49	62	79	**126**
3.438	1,000	231,000	34.6	7.27	2,000	50	60	73	93	118	**157**

under line conditions, C is a constant which varies from about 0.25 to 0.28 over the range considered, S' is the ratio of the specific gravity under line conditions to that obtaining when SSU = 220, and SSU is seconds Saybolt Universal. This adjustment for the effect of the change in viscosity with temperature can be conveniently made by reference to Table III which was computed for an oil having a specific gravity of 0.875 when the SSU is 200. If the pressure drop has been computed for the oil at any one of the SSU values shown, the pressure drop at any one of the other SSU

values can be approximated through the ratio of the respective F extension factors.

TABLE III.—FACTORS FOR DETERMINING THE APPROXIMATE PRESSURE DROP FOR AN OIL AT ONE SSU VALUE WHEN THE PRESSURE DROP AT SOME OTHER SSU VALUE IS KNOWN
(For turbulent flow, see text)

Viscosity, SSU	Specific gravity ratio S'	Empirical constant C	Extension factor F
50	0.958	0.248	0.63
100	0.987	0.263	0.82
150	0.995	0.263	0.92
200	1.000	0.266	1.00
300	1.006	0.269	1.13
400	1.008	0.269	1.21
600	1.014	0.273	1.37
800	1.020	0.279	1.51
1,000	1.022	0.281	1.61
1,500	1.025	0.283	1.81
2,000	1.027	0.284	1.95
3,000	1.032	0.284	2.17
4,000	1.037	0.280	2.30

Examples follow showing how to compute pressure drops through oil piping by the rational method and by Table II, and how to use Table III for extending the results computed for 200 SSU to other conditions.

Example 1.—What diameter of Schedule 160 steel pipe should be used for a hydraulic-power transmission line of 100-ft equivalent length for delivering to a press-operating cylinder 100 gpm of oil at a flow temperature of 120 F and an initial pressure of 2,000 psi? The oil has a specific gravity of 0.90 at 60/60 F and a viscosity of 200 SSU at 120 F. The flow velocity is not to exceed 15 ft per sec and the pressure drop is not to exceed 2 per cent of the initial pressure.

Solution by Table II.—Referring to Table II, it will be found that a 2-in. Schedule 160 pipe will just satisfy the conditions since it gives a velocity of 14.3 ft per sec and a pressure drop of 40 psi per 100 ft which is 2 per cent of the initial pressure. *Note.*—This example was chosen for an oil having the same specific gravity as that for which the table was computed. Owing to the approximate nature of all pressure-drop calculations, Table II will serve well enough without adjustment for any oil having a specific gravity at 60/60 F of the order of 0.85 to 0.95.

Solution by Rational Formula.—Since it is evident that no size smaller than 2-in. Schedule 160 pipe will satisfy the velocity requirement, a pressure-drop calculation will be tried for this size. Flow conditions are in the turbulent region since the SSU is considerably below that at the critical point which is determined as follows:

$SSU_{critical}$ u 1.059

Hence the pressure drop will be determined by the following form of the rational formula (see pages 107 to 137 and equation (55*a*) on page 87):

$$p_\lambda = \frac{fSLG^2}{18.6d^5}$$

in which f is read from Fig. 15, page 108, after determining the relation GS/dz for flow conditions in which $G = 100$, $S = 0.875$, $d = 1.689$, and z is computed as follows from Fig. 18, page 123, and Fig. 32, page 209. From Fig. 18 the value of z/S corresponding to 200 SSU is 44. From Fig. 32 an oil having a specific gravity of 0.90 at 60/60 F has a specific gravity of 0.875 at 120/60 F. Hence $z/S = 44$ and $z = 0.875 \times 44 = 38.5$. $GS/dz = 100 \times 0.875/1.689 \times 38.5 = 1.35$ and from Fig. 15, page 108, $f = 0.0115$. Then

$$p_\lambda = \frac{0.115 \times 0.875 \times 100 \times 10{,}000}{18.6 \times 13.74} = 39.4 \text{ psi},$$

which is just less than the permissible pressure drop of 2 per cent of the initial pressure.

Example 2.—Same conditions as Example 1 except that the oil is cold in starting, causing its viscosity to increase to 1,000 SSU. How will this affect the operation of the press, pump, and transmission piping?

Solution by Table II.—Referring to Table II, it is evident that the pressure drop through the piping will rise from 40 to 66 psi if the flow of 100 gpm is attained at 1,000 SSU. If this is accomplished by a rise in the initial pressure, the horsepower required by the pump will increase in proportion to the change in initial pressure. On the other hand, if it is accomplished merely by less throttling in the control value at the press, the effect on operation may be negligible. If sufficient pressure is not available to look after the added loss, the quantity of oil flowing will diminish until a balance is obtained. By interpolation in the 1,000 SSU column of Table II it appears that a pressure drop of 40 psi through the piping will deliver only about 60 gpm of the cold oil. Under this condition the action of the press would be slowed down a corresponding amount until the oil could be warmed up.

Approximate Solution by Extension Method.—Referring to Table III, it is found that the extension factor for 1,000 SSU is 1.61. Since the pressure drop at 200 SSU was computed to be 39.4 psi, the pressure drop at 1,000 SSU will be $39.4 \times 1.61 = 63.5$ psi, which agrees well enough with the result obtained by the other method. The rest of the comment remains the same.

Example 3.—Same conditions as Example 1. Does the pressure loss required to produce *velocity head* appreciably affect the results obtained?

Solution.—Referring again to Table II, it is seen that the velocity head corresponding to a flow of 100 gpm through a 2-in. Schedule 150 pipe is only 1.24 psi when the oil has a specific gravity of 0.90 at 60 F. At 120 F the specific gravity is 0.875 from Fig. 32 on page 209. Hence at 120 F the velocity head is $1.24 \times 0.875/0.90 = 1.21$ psi. A velocity head of this order is negligible with respect to the operating pressure and the over-all loss of 40 psi through the 100 ft equivalent length of pipe. If the velocity were several times higher or the equivalent length only a fraction of the 100 ft, however, the velocity-head loss

might become a significant item, in which case it should be considered with the pressure drop in determining a suitable pipe size.

HYDRAULIC CONTROL VALVES

The action of the various types of hydraulic control valves can be explained to best advantage with reference to schematic diagrams which serve to illustrate a number of basic principles common to both large and small systems. The ordinary valves, such as stops and checks, need no sketches to explain their action. Angle and globe valves are used for stops rather than gates owing to their simpler design and to avoid the dragging action experienced in pulling gates across their seats under high pressure. Check valves usually are of the ball or poppet rather than the swing type to minimize slam on closing.

The following schematic diagrams and explanation of control valves were taken in the main from an editorial résumé which appeared in *Product Engineering* for February, 1944. These descriptions may be understood better through reference to the circuit diagrams given in Figs. 2 to 4, and it is suggested that the basic explanations given herein be supplemented through reference to manufacturers' catalogues for information on actual hydraulic hookups.

Directional Control Valves.—*Four-way* is the common type, but directional valves can be obtained for 2-, 3-, 4-, 5-, and 6-way

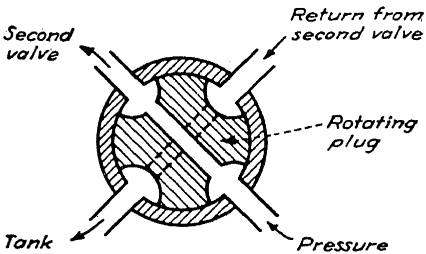


FIG. 23.—Plug-type 4-way pilot valve. (*Courtesy of Product Engineering.*)

ports. Four-way valves are used to reverse the travel of a piston by directing fluid in and out of either end of an operating cylinder, to reverse the direction of rotation of a hydraulic motor, and for similar purposes.

The three basic designs of directional valves are the *plug* type, the *poppet* type, and the *piston* type. Plug-type 4-way valves

are indicated in Figs. 2 and 3, and a pilot type of 4-way plug valve is shown in Fig. 23. Directional valves start, stop, or direct the flow of fluid by means of a movable plug, piston, or poppets which open, shut, or obstruct one or more ports. A piston type is shown diagrammatically in Fig. 24 and a poppet type in Fig. 25. The

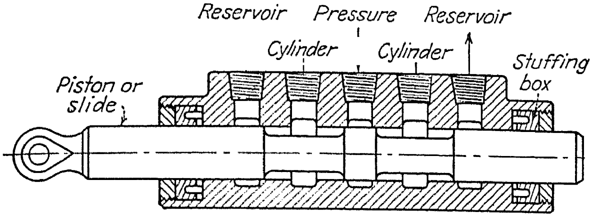


FIG. 24.—Piston-type 4-way valve. (*Courtesy of Aircraft Hydraulics.*)

spool type of 4-way valve shown in Fig. 26 is operated through a hydraulic relay circuit by a pilot valve. The valve spool floats between springs, and pilot valve action causes the spool to move in either direction. Other types are operated manually, electrically, or by mechanical or pneumatic means.

Time-delay valves (see Fig. 27) are used when change of liquid flow from one circuit to another must occur after a specified time

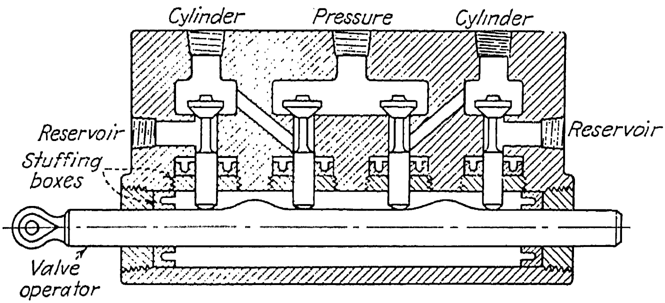


FIG. 25.—Poppet-type 4-way valve. (*Courtesy of Aircraft Hydraulics.*)

interval. The design generally incorporates a spool, and a needle valve for flow adjustment. Liquid from the 4-way valve flows past the ball check to hold the spool in the position shown. At the same time liquid flows past the upper check valve to perform work in the cylinder. When the 4-way valve is reversed, fluid

bleeds out past the needle valve from the chamber below the spool, since the ball check closes, and the spool drops, thus permitting fluid return from the cylinder to flow through chamber R back to the 4-way valve.

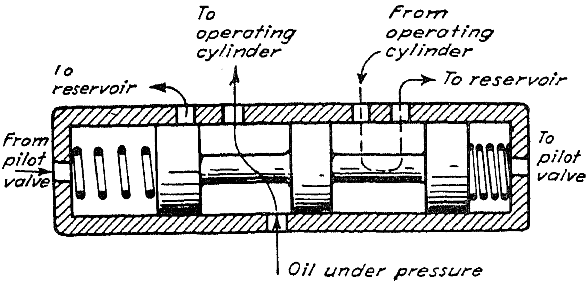


FIG. 26.—Spool-type 4-way valve operated through a hydraulic relay circuit controlled by a pilot valve. (*Courtesy of Product Engineering.*)

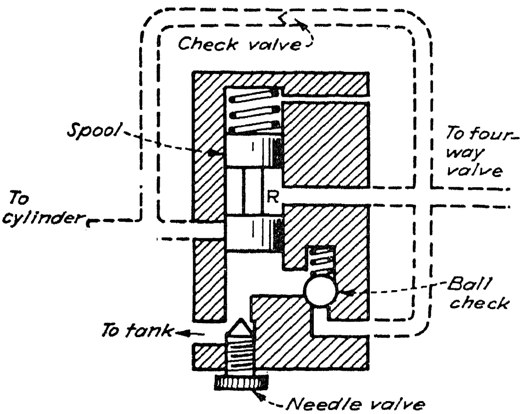


FIG. 27.—Time-delay valve. (*Courtesy of Product Engineering.*)

Servo valves (Fig. 28) are used to move large masses with a minimum of operator effort. They are basically similar to the directional control valves previously described, except that both the spool and the bushing move. The bushing is connected to the moving slide so that when movement of the slide occurs, the bushing goes with it to close off the valve. A small axial movement of the screw results in a small movement of the slide. The

only load on the screw is that necessary to move the valve in its bushing.

Pressure Control Valves.—These valves operate by pressure differentials for such purposes as limiting maximum pressure; controlling sequence of operation by governing pressure to branch circuits; returning fluid to the reservoir, with or without resistance,

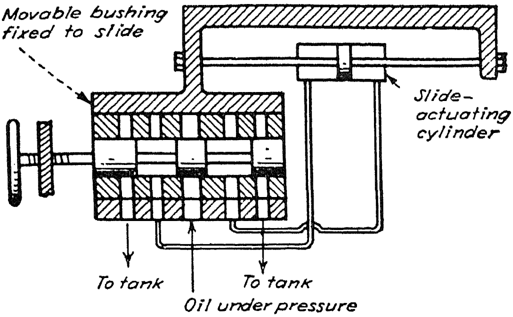


FIG. 28.—Servo valve and linkage. (*Courtesy of Product Engineering.*)

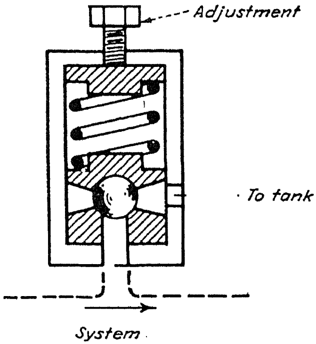


FIG. 29.—Relief valve. (*Courtesy of Product Engineering.*)

when the system pressure builds up; reducing pressure in system lines; and performing many other operations where pressure differentials can be used to open or close valves. Descriptions of some of the principal types follow.

Relief valves (see Fig. 29) are used to limit the pressure in a circuit to a desired maximum. An adjustment is provided which can be remotely controlled if desired. High-pressure oil is dis-

charged from these valves back to the reservoir. Since high-pressure energy is converted to heat when being relieved, such valves should be used only intermittently or as safety valves, and not as unloading valves if a large or continuous discharge volume is expected. Unloading valves are described below.

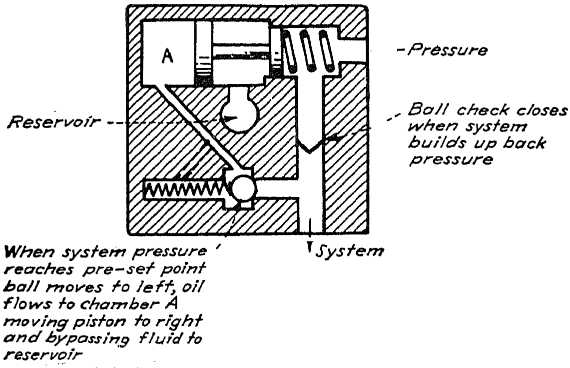


FIG. 30.—Unloading valve. (*Courtesy of Product Engineering.*)

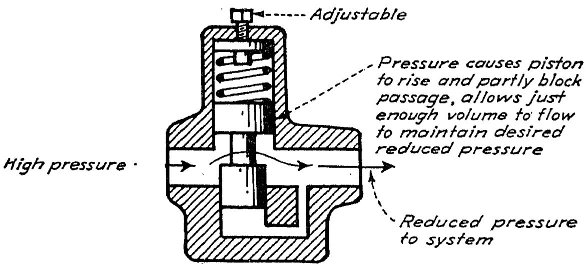


FIG. 31.—Pressure-reducing valve. (*Courtesy of Product Engineering.*)

Unloading valves (see Fig. 30) are used to return fluid to the reservoir without flow resistance when the system pressure builds up to a pre-set maximum. For instance, where a constant-delivery pump is supplying oil to the circuit and the cylinders have completed their stroke, pressure will build up in the system. In such cases, an unloading valve will open and by-pass fluid back to the reservoir with little or no back pressure so that a minimum amount of heat is dissipated. This also reduces the pump power requirements a corresponding amount during these intervals.

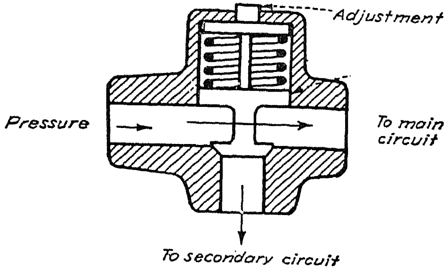
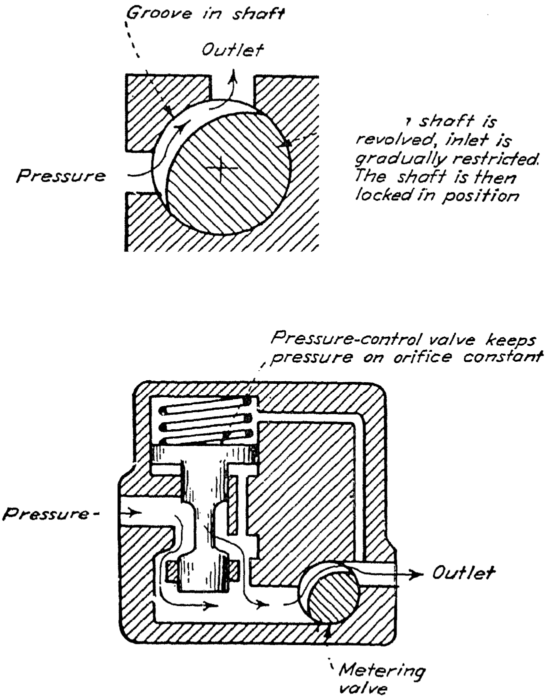


FIG. 32.—Pressure-sequence valve. (*Courtesy of Product Engineering.*)



b

FIG. 33.—(*a*) Metering valve. (*b*) Flow controller with metering valve. (*Courtesy of Product Engineering.*)

Pressure-reducing valves (see Fig. 31) are used where certain parts of hydraulic control circuits require reduced pressures for operation. Pressure regulation is obtained through the use of a spool which balances the reduced pressure against an adjustable spring so as to give a valve opening just sufficient to hold a predetermined reduced pressure.

Sequence valves (see Fig. 32) are pressure-operated adjustable devices for controlling the sequence of flow to various secondary circuits. Sequence valves are installed in the line so as to allow free flow to the main circuit until pressure builds up to the desired point. The valve then opens and supplies oil under pressure to the secondary circuit while still maintaining flow into the main circuit, the valve simply acting as a tee under these conditions.

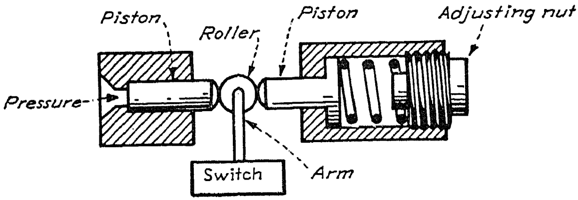


FIG. 34.—Pressure switch. (*Courtesy of Product Engineering.*)

Check valves sometimes are incorporated in the design to permit free reverse flow.

Flow controllers and metering valves in combination (see Fig. 33) are used in hydraulic circuits where the rate of fluid flow in a certain direction must be controlled. Piston movement in the work cylinder can be governed so that the rate of travel will be constant regardless of fluid pressure or viscosity, or of magnitude of the variable load. Among other applications, such controllers are used to prevent tools from jumping ahead when breaking through the work. They serve to meter in, meter out, or bleed off fluid from a circuit. The metering valve shown in combination with the controller can be used independently to limit the flow of fluid to be delivered to or discharged from an operating mechanism, to cushion pistons at the end of their stroke, or to delay the action of hydraulically operated valves.

Pressure switches (see Fig. 34) are used where flow in a hydraulic circuit is to be controlled electrically. Change in pressure causes the switch to open or close a relay circuit which actuates a solenoid

or other device used to operate a valve, or to start or stop the pump. Pressure switches can be fitted with needle valves for adjustment of reset speed.

SELECTING PIPE AND FITTINGS

In the absence of national standards and codes for hydraulic-power transmission systems, such piping has to be selected on the strength of manufacturers' recommendations and the designers' judgment. Certain empirical rules which have proved satisfactory for hydraulic work and become established through long usage are not necessarily acceptable for other applications. For instance, although fluid velocities are high and shock pressures correspondingly greater than would be encountered in other fields, hydraulic piping is expected to sustain this overpressure without much provision for it in the factor of safety. Under-thickness tolerances encountered in pipe manufacture as well as the encroachment on wall thickness where pipe is threaded often are neglected also. These liberalities may be justified, in some measure at least, by the fact that operation usually is within the range of indoor atmospheric temperatures, or only slightly above. Some common practices followed in selecting hydraulic pipe and fittings are described in succeeding sections.

Materials.—The required service conditions can be met in most hydraulic piping installations with carbon or chrome-molybdenum steel pipe or tubing, or by copper tubing and brass fittings. The larger hydraulic systems usually are constructed of carbon-steel pipe materials such as those specified in ASTM Specifications A53 or A106 (see pages 373 to 385). Where steel tubing is used with flared-tube fittings, a low-carbon variety of carbon steel such as SAE 1015 or 1020 is required in order to secure satisfactory flaring and fabricating properties. With high pressures, a harder material such as chrome-molybdenum steel (SAE X4130) often is used. Because of its light weight, aluminum-alloy tubes of 52S-0 alloy (Aluminum Company of America), and aluminum fittings in accordance with Federal Specifications QQ-A-367 Grade 1 are used in aircraft systems. For hydraulic systems operating at pressures of 1,000 psi or less and where the installation is free from excessive vibration and hydraulic shocks, copper tubing and brass fittings are commonly used. Copper tubing may be purchased in accordance with ASTM Specification B75 (see page

1344). Forged or extruded brass fittings of nonporous close-grained structure should be used with copper tubing.

Materials commonly used in flared-tubing installations are indicated in Table IV.

TABLE IV.—TUBE AND FITTING MATERIALS

Tube	Fitting	Nut	Sleeve
Copper, ASTM B75	Brass, QQ-B-611[1]	Brass, QQ-B-611[1]	Aluminum bronze, AN-B-16[2]
Carbon steel, SAE 1020	Carbon steel, SAE 1020 Alloy steel, SAE X4130	Carbon steel, SAE 1020 Alloy steel, SAE X4130	Carbon steel, SAE 1020 Alloy steel, SAE X4130
Aluminum alloy, 52S-0[3]	Aluminum alloy, QQ-A-367[1]	Aluminum alloy, QQ-A-367[1]	Aluminum bronze, AN-B-16[2] or aluminum alloy, QQ-A-367[1]

[1] Federal Specifications.
[2] Army-Navy Aeronautical Board Specification.
[3] Aluminum Company of America designation.

For high-pressure applications above 1,000 psi, carbon- or alloy-steel piping or tubing and steel fittings are recommended rather than heavy-wall copper tubing and heavy brass fittings. Recommendations to provide for mechanical strain or service abuse must depend on the nature of the service encountered. In general, it is considered desirable to use heavier weights of tubing and fittings rather than harder materials. Steel nuts sometimes are used with brass fittings where frequent disassembly of fittings is necessary. Mating of steel with aluminum-alloy fittings or parts is not recommended. All tubing that is to be flared or bent should be dead-soft annealed to secure best bending and flaring properties and should, of course, be free from scratches, draw marks, burrs, or scale.

Wall Thickness of Pipe and Tubing.—The thickness of pipe or tubing required for hydraulic-power work often is computed by Barlow's formula (see page 41) or read from tables based on that formula. For convenience in computing pipe wall thickness, this formula may be transposed to read $t = pD/2S$ where t is the pipe wall thickness in inches, p is the design pressure in psi, D is the outside diameter of the pipe or tubing in inches, and S is the allowable stress for the material in psi. The allowable stress may be read from a table such as Table V on page 43, or it may be determined with respect to a factor of safety F, the seam effi-

ciency E if there is a seam, and the tensile strength of the pipe material from the relation $S = E \times (\text{tensile strength})/F$.

With hydraulic piping it is customary to allow a factor of safety of 5 to 6 in determining S for use in the Barlow formula. The joint efficiencies E commonly used for different kinds of pipe are seamless, 1.00; resistance-welded, 0.85; lap-welded, 0.80; butt-welded, 60. If severe water hammer is to be expected, some manufacturers of hydraulic equipment recommend that the piping be designed for a maximum pressure at least one-third higher than the normal working pressure. With high velocities in large water systems, this still leaves considerable shock to be looked after in the factor of safety, since in a high-pressure system shock pressures under adverse conditions may run from 50 to 100 per cent above the normal working pressure. As mentioned before, the designers of hydraulic systems often consider the computed thickness t as the nominal wall thickness without making any allowance for mill tolerance in manufacture which may run as much as 12½ per cent under the nominal thickness, or for material removed in threading if screwed joints are used.

Where more conservative design is wanted, the formulas given in the American Standard Code for Pressure Piping and the ASME Boiler Code can be used. These formulas, given on pages 42 to 49, distinguish between threaded pipe and "plain-end" pipe, require an allowance for mill tolerance and, with the exception of nonferrous pipe, provide an allowance for mechanical strength and/or corrosion.

The use of copper tubing, usually made up with flared connectors, is limited to pressures of about 1,000 psi. Steel tube and pipe are used for pressures up to 5,000 to 6,000 psi and, for the higher pressures, are of seamless construction and sometimes of alloy steel.

Although the selection of tube or pipe wall thickness should be based primarily on the pressure requirements of the system, heavier walls may be required where pipe or tubing is subject to unusual mechanical abuse, excessive vibration, mechanical loads, and large hydraulic shock or surge pressures. These conditions should be avoided where it is possible to do so, but if this is not practicable, pipe or tubing of heavier wall than would be required by the working pressure alone must be used. The wall thickness required in these cases is left largely to the judgment of the designing engineer to suit the particular conditions encountered.

After the *minimum* wall thickness for pipe or tubing has been arrived at from the foregoing considerations, the next heavier *commercial* thickness should be selected from the dimensional standards and used for the job. Schedules and tables giving the dimensions for regularly manufactured and stocked pipe diameters and thicknesses will be found in Chap. IV. In the absence of standard dimensions for the special kinds of tubing used with flared-type connectors, reference should be made to manufacturers' catalogues or to standard specifications for the particular material, such as ASTM B75 for copper tubing. Recommendations of the Parker Appliance Company for weight of their fittings to use with corresponding thicknesses of the connecting tubing are given in Table V. These dimensional recommendations apply to both ferrous and nonferrous tubes and fittings, subject to the limitation that suitable materials be used together.

TABLE V.—MAXIMUM WALL THICKNESS IN INCHES OF TUBING SUITABLE FOR USE WITH THE PARKER APPLIANCE COMPANY STANDARDS OF FITTINGS

Tube O.D., in.	Fitting designation			
	Standard	Heavy	Extra-heavy	Double-extra-heavy
1/8	0.035			
3/16	0.035			
1/4	0.035	0.050	0.060	0.080
5/16	0.035	0.060	0.075	0.100
3/8	0.035	0.060	0.090	0.120
7/16	0.050	0.065		
1/2	0.050	0.070	0.115	0.160
5/8	0.060	0.080	0.140	0.190
3/4	0.065	0.090	0.165	0.225
7/8	0.070	0.100	0.190	0.260
1	0.070	0.110	0.210	0.300
1 1/8	0.080	0.110		
1 1/4	0.080	0.120		
1 1/2	0.090	0.130		
1 3/4	0.095	0.140		
2	0.100	0.150		

Types of Joints.—Connections between ends of pipe or tubing may be made by use of screwed, flanged, brazed, or welding fittings, or by means of flared-tube connectors. Where it is necessary to disassemble a joint frequently, flared-tube connectors are used

extensively because of their ease of disassembly and reconnection. Where it is desirable to make a permanent leakproof joint, welded or brazed joints may be employed. Dimensions for steel socket-welding fittings which match the ASA B36 schedule series for pipe are given in the proposed American Standard for Steel Socket-welding Fittings, ASA B16.11 (see page 505). Socket-welding fittings to correspond to standard-weight, extra-strong, and double-extra-strong pipe also are available. Butt-welding fittings in

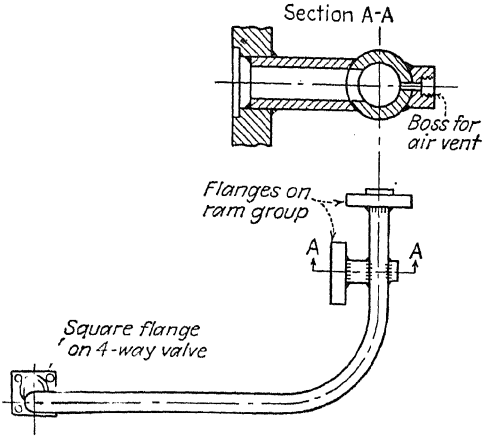


FIG. 35.—Typical piping detail for 1,500-psi hydraulic-steering-gear system on ship-board.

sizes 1 in. and larger are included in the American Standard for Steel Butt-welding Fittings, ASA B16.9. Brazed joints are not widely used for hydraulic piping, but fittings for pressures up to 3,000 psi are available in accordance with the standards of several manufacturers.

Screwed fittings are, in general, restricted for use with steel pipe since the pipe must be of ample wall thickness so that sufficient metal remains after threading to withstand the pressure requirements as well as the additional shock and mechanical loads that may be imposed on the system. Flanged joints, where called for, often are of special design with gaskets retained in recesses against the ends of the pipe, and with square or oval instead of round flanges in the small to medium sizes. Typical piping using slip-on

welding flanges designed for a 1,500-psi hydraulic-steering-gear system on shipboard is shown in Fig. 35.

Strength of Fittings.—Forged-steel screwed fittings are available in *hydraulic* ratings of 2,000, 3,000, and 6,000 psi. Forged-steel socket-welding fittings (see pages 505 to 506) are available for the same pressures as forged-steel screwed fittings, as well as to match the ASA B36 Schedule Number system for pipe. Steel butt-welding fittings (see pages 502 to 505) also are available in weights up to what is used with Schedule 160 pipe. Forged-steel flanged unions are offered to manufacturers' standards for pressures from 500 to 6,000 psi. Flanges usually are provided with tongue-and-groove facings and a gasket or are of the ball-and-socket ground-joint type. Two-bolt oval designs are offered for sizes 1 in. and smaller, whereas a four-bolt square design may be secured for sizes up to about 1½ in. American Standard flanges are available with standard facing designs for pressures from 150 to 2,500 psi and in sizes ½ in. and larger (see ASA B16e for Steel Pipe Flanges and Flanged Fittings, pages 626 to 680).

Flared-tube fittings are available in different weights depending on the make of the fitting but, in general, the weight of the fitting should be selected to match the wall thickness of the tubing used. Recommendations of the Parker Appliance Company for weight of fitting based on the maximum thickness of the connected tubing are given in Table V.

Flared-tube Fittings.—Largely because of their ease of disassembly, flared-tube fittings are widely used in hydraulic piping. In this joint the flared end of thin-wall tubing is compressed by a nut and screw between two surfaces parallel to the flare so that the tubing acts as a gasket. Flared-tube fittings consist of two or three parts. Those having two parts involve a body and a nut. Three-part fittings have, in addition, a sleeve in which the seal is made by squeezing the tube flare between the sleeve and the tapered surface on the fitting. This design avoids the wiping action of the nut on the tubing flare which tends to thin out the latter, or to score the seat between the nut and tube, either of which may result ultimately in a leaky joint. Figure 36 shows several types of flared-tube fittings and the Ermeto fitting in which the seal is made by utilizing a special design of sleeve without flaring the tube ends.

Tubing sometimes is given what is called a "double flare" in which the end of the tubing is bent back upon itself to provide a

double wall thickness at the end of the tube. A flare of this type is shown in Fig. 37. This is intended to eliminate leaky joints that result from thinning of the single-flare thickness caused by tightening nuts too severely or by frequent remaking of the joint.

Fitting size generally is designated by the size of tubing with which it is used, which is normally given in inches or fractions down

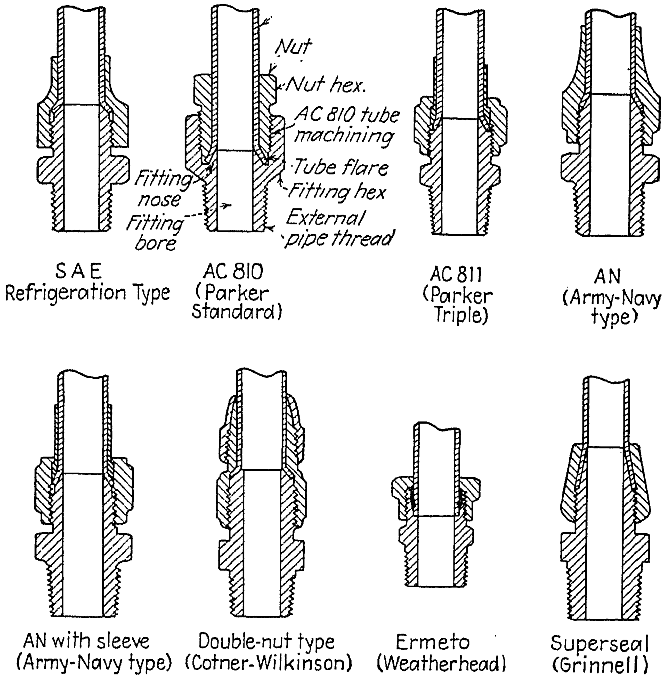


FIG. 36.—Various types of tube fittings.

to sixteenths of an inch corresponding to the outside diameter of the tubing. Fittings are commonly available in sizes from $\frac{1}{8}$ to 2 in. On fittings that have one or more pipe thread outlets, standard pipe threads corresponding to the tubing size are provided.

Figure 36 shows flared-tube fittings of the same shape, *i.e.*, straight connectors with external pipe threads. Other standard shapes are 90-deg elbows, 45-deg elbows, tees, and crosses. There

are numerous special shape fittings also such as side-outlet fittings and pad fittings. In addition, fittings of the same shape may be provided with different outlets having internal or external pipe or machine threads depending on the purpose for which the fitting is to be used. Dimensions for the more commonly used fitting shapes may be obtained by reference to catalogues of the manufacturer whose design of fitting is being used.

Although certain parts of some of the several types of fittings are interchangeable, difference in threads, variation in flare angle, or difference in over-all dimensions renders it impractical to interchange parts promiscuously. It generally is advisable to replace a fitting with an identical fitting, or to replace the entire unit and reflare the tube if necessary.[1] Since the *pipe* threads used on all

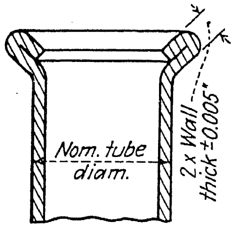


FIG. 37.—Double flare for tubing.

fittings are identical, size for size, a complete assembly may usually be interchanged unless clearance in close quarters will not permit because of interference.

In making up flared-tube connections, it is desirable to use an antiseize compound or lubricant to prevent galling and to facilitate assembly and permit ready disassembly. Aluminum-alloy threads will gall or seize unless a lubricant is used. Screwed joints frequently are fillet-welded or brazed for tightness where the pipe enters the fitting. Where this is not done, the threads sometimes are tinned or copper-plated to prevent leakage. A sealing compound also may be used to ensure a tight joint. Numerous types of lubricants and sealing compounds are available, selection from which should be made to suit the fluid conveyed. In general, it may be stated that the compound should be insoluble in the fluid conveyed except in the case of oil where the reverse is true.

[1] Charts showing equivalent fittings in the AC810, AC811, and AN types are given in "Equivalent Charts for Aircraft Plumbing Fittings," by R. H. Murray, *Aviation*, May, 1944, pp. 165–184.

If oil is the fluid conveyed, the oil itself will provide sufficient lubrication once the joint is made.

Care should be exercised to avoid excessive wrench torque on flared-tube nuts when making a joint using soft tubing, particularly when a thread lubricant is applied since the lubricant tends to eliminate the "feel" when the joint is tight. If nuts are pulled down too tightly with soft tubing, the flare will be damaged.

TABLE VI.—DATA ON REINFORCED SYNTHETIC-RUBBER FLEXIBLE HOSE[1]

Nominal I.D., in.	Max. O.D., in.	Working pressure max., psi	Proof test pressure, psi	Burst pressure, min., psi	Min. bend radii, in.	Weight, lb per ft
Low-pressure type						
1/8	0.358	415	1,250	2,500	3	0.06
3/16	0.420	333	1,000	2,000	4	0.075
1/4	0.490	275	750	1,500	5	0.08
3/8	0.625	165	500	1,000	6	0.11
Medium-pressure type						
1/8	0.500	975	2,900	5,800	5	0.08
3/16	0.625	835	2,500	5,000	6	0.12
1/4	0.720	835	2,500	5,000	6.5	0.13
5/16	0.780	800	2,500	4,800	7	0.19
3/8	0.875	800	2,500	4,500	8	0.22
1/2	1.032	500	1,500	3,000	9	0.35
High-pressure type						
3/16	0.656	3,600	9,000	18,000	6.5	0.28
1/4	0.718	3,600	9,000	18,000	7.5	0.31
5/16	0.843	3,000	7,500	15,000	8	0.40
3/8	0.906	3,000	7,500	15,000	9	0.50
1/2	1.031	2,400	6,000	12,000	10	0.60

[1] From "Hydraulic Lines, Which Material, What Size," by Richard K. Lotz, *Machine Design*, December, 1943, p. 145.

In laying out circuits for hydraulic lines, it is desirable to keep the number of bends to a minimum to avoid excessive fluid pressure drop. However, at least one bend should be present in all lines to absorb expansion and contraction strains resulting from changes in temperature. Otherwise, such strains may cause a failure of the tube flare which is, of course, the weakest part of the line.

Information on bending steel pipe and tubing will be found on pages 772 to 777. For instructions on bending nonferrous tubing, see "Pipe and Tube Bending Handbook," published by the Copper and Brass Research Association, 420 Lexington Ave., New York 17, N.Y., and technical magazine articles.[1]

Flexible Lines.—In aircraft applications as well as for many machines utilizing hydraulic fluids for power, movement is required between units that are interconnected. This may be accomplished by the use of reinforced synthetic-rubber hose which is available in three general pressure classifications: low-pressure, medium-pressure, and high-pressure. Such hose generally is provided with an inner and outer layer of synthetic rubber with one or more reinforcing braids of cotton. High-pressure hose also may have a wire reinforcing braid. Data on the three pressure classifications of such hose are given in Table VI. ASTM, SAE, and Army-Navy specifications are available which give construction details and test requirements for hose used in various applications.

[1] See, for instance, "How to Bend Aluminum Tubes and Shapes," *Iron Age*, Dec. 28, 1944, pp. 42–45.

INDEX

F

G

H

N

P